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THE IMPACT ESTIMATION OF DAMPING FOUNDATIONS IN DYNAMICS OF THE ROTOR SYSTEM

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Abstract

The paper presents results of dynamic researches conducted on test bed of rotor system. The aim of research was to identify the impact of damping washers to natural frequency of the system. The first part of paper treats with measurements of rotor system in non-stationary states and the second one is about vibration performances parameters in stationary states. Article presents an example of approach to the problem conducted in laboratory conditions, the results of which can be easily transferred on board of a ship or vessel.

Keywords: rotor system, deceleration, resonance, damping

OCENA WPŁYWU TŁUMIENIA FUNDAMENTÓW NA CHARAKTERYSTYKI DYNAMICZNE UKŁADÓW WIRNIKOWYCH

Streszczenie

W artykule przedstawiono wyniki badań przeprowadzonych na stanowisku modelowym układu wirującego. Celem badań było zidentyfikowanie wpływu podkładek tłumiących na częstotliwość rezonansową systemu. W pierwszej część pracy zaprezentowano wyniki uzyskane podczas pomiarów parametrów drganiowych w czasie pracy układu wirnikowego w stanach nieustalonych, natomiast druga dotyczy parametrów drgań w stanie ustalonym. Artykuł przedstawia przykład podejścia do problemu prowadzonego w warunkach laboratoryjnych, którego wyniki można łatwo zaimplementować podczas pomiarów na okrętach lub innych jednostkach pływających.

Słowa kluczowe: układ wirujący, wybieg, rezonans, tłumienie

1. INTRODUCTION

High angular speed of unbalanced rotating masses causes forces transferred through the bearing to the foundation of machine. The amplitudes of vibrations connected with that forces become higher as far as the damping factor of foundation is getting worst [3,8,10,12]. The paper contains the description of the active-passive experiment. The research stand consists of a foundation, spring washers, dampers, rotor system with two bearings and two masses and an electric motor with a rotational speed controller.

Due to utilization of power inverter a smooth adjustment of acceleration and deceleration of rotor system (within 0 to 1480 rpm) was possible. The main aim of this work was to create dynamic characteristics which made possible the selection and evaluation of damping elements. Basing on the results of measurements some resonant frequencies in the vibration spectra was determined. Another objective of the research was the methodology of selection damping pads for the object of research. Not all the data necessary to determine the dynamic characteristics was known. Such situation often occurs during the repairs and maintenance process of rotating parts in the shipbuilding industry. The study was performed using the procedure of order tracking. The adopted methodology allows recognition of resonance frequencies in the process of acceleration and deceleration of the machine. It is essential for the proper calculation and selection of damping pads for rotating machines.

Rotating machines produce repetitive vibrations and acoustic signals connected with rotational speed. These relationships are not always obvious with standard dynamic signal analysis (for example FFT) and especially in this cases the order analysis is useful tool. Order analysis becoming a commonly used technique for analysis of vibrations generated in machines, where many vibrations are related to machine RPMs [1,2,3,4,5,7,9,11,13,14].

The FFT process transforms time domain data to the frequency domain, creating a spectrum. Periodic signals in the time domain appear as peaks in the frequency domain. In order analysis the FFT transforms the revolution domain data into an order spectrum. Signals that are periodic in the revolution domain appear as peaks in the order domain [4]. For example, in rotor with 10 blades without any unbalance and misalignment the peaks connected with blade passing frequency will appear as 10 order in the order spectrum.

In order analysis normally a signal from tachometer probe is used as a tracking reference. It allows a measurement to be related to the revolutions of a rotating element in the machinery. In the order analysis spectral elements that are constant with frequency, for example resonance peaks are well visualized, this is the reason why order analysis is often first step in a trouble-shooting scheme, in order to investigate whether a vibration problem is resonances or other reasons [11].

1.1 Marine machinery foundations

Actually almost all machinery working in ship's power plants are located on the rigid foundations. In order to minimize the propagation of vibration generated by working devices on the hull of the vessel, they are mounted to the foundations by the properly selected vibration isolators. Figure 1 presents an example of marine foundation with vibroisolators potential locations.

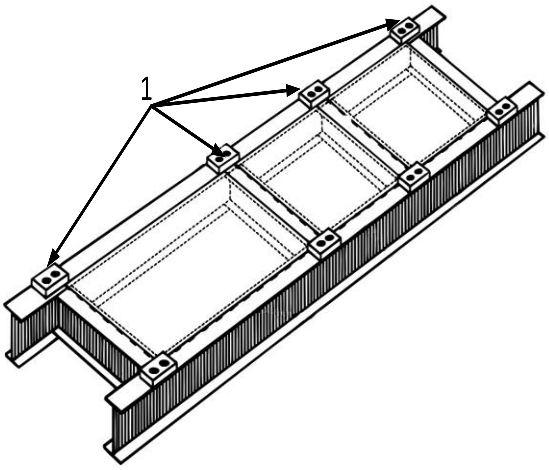

Fig. 1. Example of marine foundation. 1 – vibroisolators mounting place

The most commonly used solution to decrees influence of machine vibrations to ship's hull is use of rubber metal shock absorbers between the insulated machine and the foundation. There is also a solution of the so-called floating foundation frame where additional vibroisolation is applied at the point where the frame is joined to the hull. Particular attention should be paid to the variations in the characteristics of the rubber elements used as the vibration isolators. In shipbuilding, there are extremely difficult conditions for the use of rubber materials, which are primarily due to the high temperature and ubiquitous vapors of petroleum products and salt. Under such conditions, the aging process of gum and the changes in its characteristics are quite fast. Regardless of the type of vibration insulation used, vibration isolation characteristics change over time, resulting in changes in the dynamic characteristics of the properly working machine. It is possible to calculate this influence during computer simulations but better solution is to conduct tests on working machines. There is one condition: measurements for proper working machine had to be known.

2. OBJECT AND METHODOLOGY OF RESEARCHES

Researches was conducted on the Schenck rotor system of the laboratory stand. It is combined with electrical motor coupled with set of two discs mounted on a solid shaft placed on two bearings – Figure 2. Entire construction is attached to steel frame. Metal discs have premade holes to be able to install additional masses to simulate unbalance.

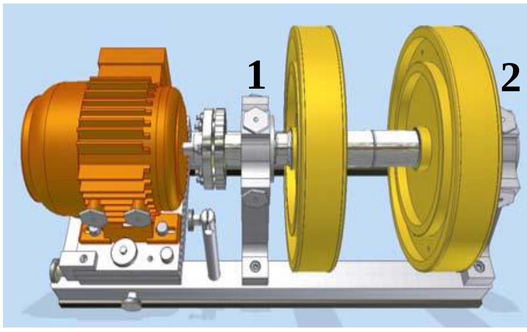

Fig. 2. Schenck laboratory station, upper – the photography and lower - spatial model with number of bearings

Schenck rotor system is propelled with asynchronous three-phase motor AEG AM 56KY4. The power inlet was based on block of assembled capacitive-inductive reactance creating 3-phase electrical connection from 1-phase electrical line (230 V/50 Hz) by allowing phase shift for two additional phases (120 and 240 degrees). The negative outcome of such array was lack of possibility of any adjustments nor control.

After proper consideration of AEG motor it was replaced by existing power supply with ABB power inverter type ACS150-01E-02A4-2 (0,37 kW).

The study used the process of deceleration of the rotor system, which in both variants of the experiment (with/without dampers) allowed unforced (free) vibration analysis. The adopted model of research has provided no interactions unexpected, external forces and torques, the effects of which could affect the experiment results. This approach to research, along with the possibility of variable speed has allowed to define the experiment as passive-active type.

2.1 Diagnostics Methodology

The aim of passive-active experiment is to observe the input signals with simultaneous measurement of the quality of state without the possibility of interference in their values (Fig 3). It is possible to conduct the passive-active experiment during normal operation of machine [15].

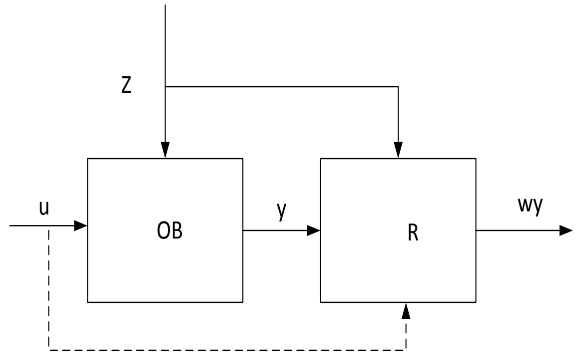

Fig. 3. Passive-active experiment S - control, u - forced control inputs OB – research object, ZE - external inputs, y - outputs, R - processing, wy – experiment result [15]

The identification of dynamic parameters in mechanical systems is important for improving model-based control as well as for performing realistic dynamic simulations. Generally, when identification techniques are applied only a subset of so-called base parameters can be identified. In order to evaluate the forward dynamics response, an approach for obtaining the forward dynamics in terms of the relevant parameters is also proposed. To assess the impact on unbalance and damping value of measured vibrations, a series of measurements was performed, containing measurements in non-stationary states:

- without any additional mass on cylinders and without any additional damping,
- without additional mass and with damping unit,
- with additional mass (causing unbalance), and without any additional damping,
- with additional mass (causing unbalance) and with damping unit.

The algorithm of the researches in non-stationary states is presented on the Figure 4.

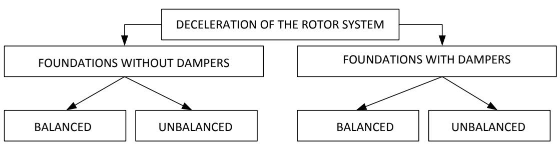

Fig. 4 The algorithm of the researches

Measurements was conducted during the run down of rotor system. Such approach allows to measure only the resonance of the rotor system without influence of current frequency and impact of power torque. At the end measurements in stationary state of rotor system also had been conducted.

Exceeding acceptable unbalanced on rotors increases vibration energy. The consequence may be the excitation of resonant vibration and dynamic load growth. During operation rotors are subject to the occurrence of vibration in planes of bent, torsional and longitudinal. Working rotor should therefore be considered as a system of vibrating twist-bend-split.

In practice, most industrial and laboratory measured values are change in size over time. This variation is the base of criterion for the distribution of measurements: for static and dynamic measurements. In static measurements value do not change with time or variation of the measured value accuracy, but it does not affect the result. Dynamic measurement is made when the aim of the measurement is quantitative illustration of the time variation of the measured value. The result of a dynamic measurement is representation of the time course of the measured value. The outcome may be a plot of the measured value as a function of time, the so-called drawn directly by an analog recorder or within pairs of numbers *{[t,x·(t)]·i}*, where *i* is the number of time point, *t* is time and *x·(t)* is the instantaneous value of the measurement.

During measurements on laboratory test bed there were used two accelerometers type B&K 4514 B, tacho probe type MM024 and measurement frontend type 3650-B-120. Sampling frequency of 8192 Hz has been used.

3. RESULTS OF RESEARCHES

As a result of measurements time signals of accelerations had been obtained. With use of Pulse software platform signals were analyzed in frequency and order domain. At first stage frequency spectrum of accelerations has been calculated and then the order spectrum with resolution of 30 orders. Sample of obtained results has been presented on figures 5 – 12. All results were collected in table 1. Location of bearing in rotor system was presented on figure 1. On presented pictures it is clearly visible that in frequency spectrum it is possible to find resonance but it is no so obvious as on order spectrum.

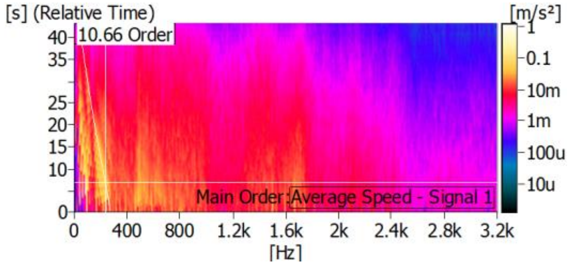

Fig. 5. Spectrogram of acceleration obtained on object without additional mass without dampers – bearing no. 1

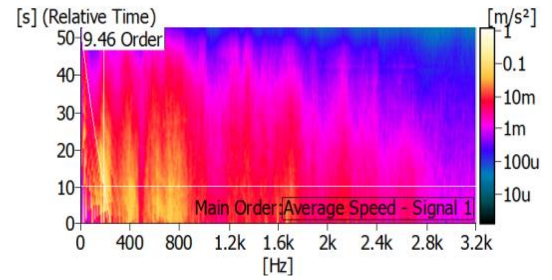

Fig. 6. Spectrogram of acceleration obtained on object without additional mass with dampers – bearing no. 1

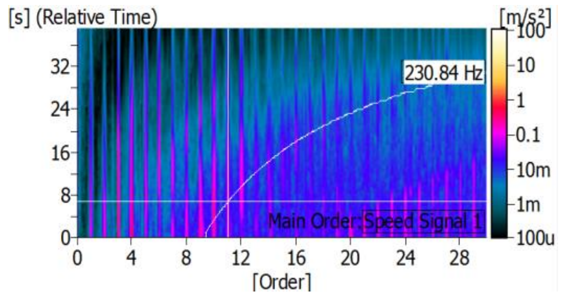

Fig. 7. Order spectrum of acceleration obtained on object without additional mass without dampers – bearing no. 1

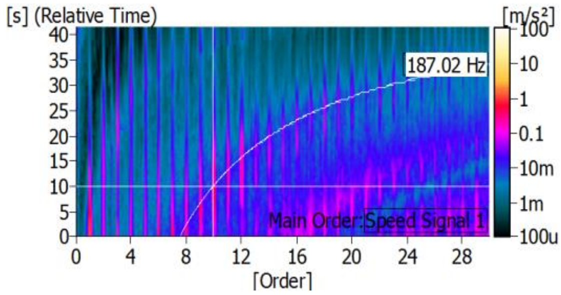

Fig. 8. Order spectrum of acceleration obtained on object without additional mass without dampers – bearing no. 1

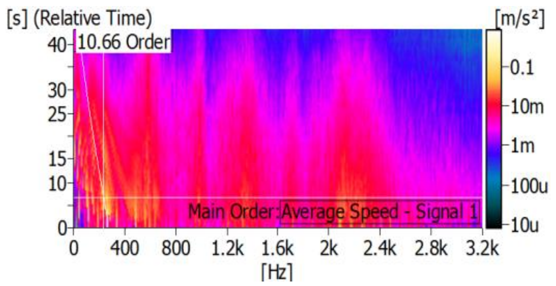

Fig. 9. Spectrogram of acceleration obtained on object without additional mass without dampers – bearing no. 2

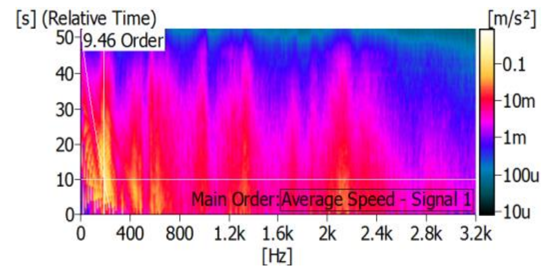

Fig. 10. Spectrogram of acceleration obtained on object without additional mass with dampers – bearing no. 2

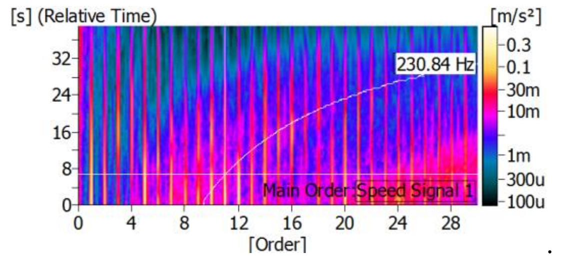

Fig. 11. Order spectrum of acceleration obtained on object without additional mass without dampers – bearing no. 2

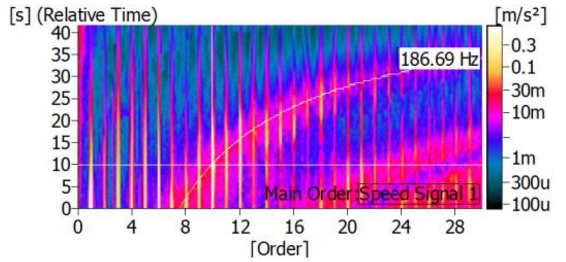

Fig. 12. Order spectrum of acceleration obtained on object without additional mass without dampers – bearing no. 2

Table 1. Results of vibration analysis

	Bearing 1		Bearing 2		Δf Hz
	Frequency Hz	Amplitude mm/s^2	Frequency Hz	Amplitude mm/s^2	
Without additional mass and without dampers	230	172	230	112	43
Without additional mass and with dampers	187	86	187	95	
With additional mass and without dampers	233	180	233	168	57
With additional mass and with dampers	196	67	196	77	

Calculated resonant frequency of a part of rotor system located between the bearings is 282 Hz. Estimated resonant frequency of rotor system without any additional mas and extra dampers as presented in table 1 is 230 Hz. Analyzing the results of researches it is evident that use of damping washers change the vibration characteristic of rotor system. First resonant frequency of rotor system has decrease with frequency and amplitude. More over range of resonance had become more narrow. On the other hand at frequency around 400 Hz new resonance has occurred. It also visible that in domain of orders it is easier to find a resonance than in frequency domain.

During the next step of researches damping coefficient of damping elements used during measurements was calculated. According to this it was necessary to weight the rotor system and damping elements (Vibrochoc units).
Rotor system weight: $M_{RS} = 11{,}130$ kg
Double Vibrochoc unit weight: $M_{VIBRO} = 1{,}288$ kg
Next a series of various masses was placed on double Vibrochoc unit and its displacement was measured.
Further results calculated from the measured values using formulas for:

- displacement

$$\Delta x_i = |x_i - x_{i+1}| \qquad (1)$$

where index is number of next measurements $i=1,2,3\ldots\ldots12$, number of conducted measurements was 13

- static force:

$$F_i = m_i \cdot g \qquad (2)$$

where g is gravitational acceleration

- spring constant:

$$k_i = \frac{F_i}{\Delta x_i} \qquad (3)$$

Finally with calculated values it was possible to create spring characteristic chart - Fig 13.

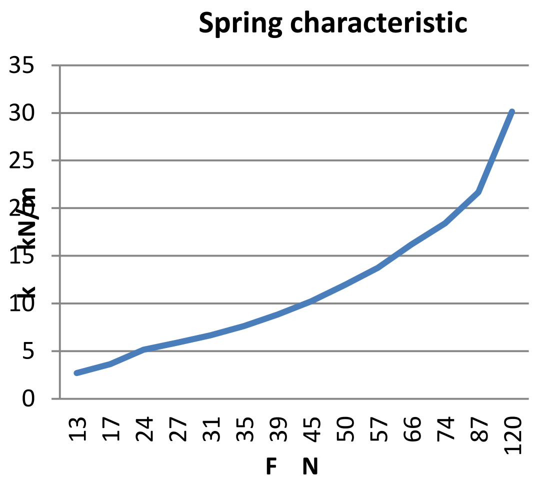

Fig. 13. Vibrochoc damping module characteristic

To calculate damping factor of Vibrochoc which is combination of spring and damping elements some equation transformation had been conducted. The resonant frequency of rotor system without damping is described by the equation (4):

$$f_k = \frac{\sqrt{\frac{k}{m}}}{2\pi} \qquad (4)$$

where: f_k – the resonant frequency of the stiffness, without damping, k – stiffness, m – mass.
If the resonant frequency of system with stiffness and damping f_{kc} is (5)

$$f_{kc} = \frac{\omega_n\sqrt{1-\frac{c^2}{4m^2\omega_n^2}}}{2\pi} \qquad (5)$$

where: $\omega_n = 2\pi f_k$ – natural frequency (without damping), c – damping factor;
then it could be written:

$$\Delta f = f_k - f_{kc} = \frac{\sqrt{\frac{k}{m}}}{2\pi} - \frac{\omega_n\sqrt{1-\frac{c^2}{4m^2\omega_n^2}}}{2\pi} = \sqrt{\frac{k}{m}} - \omega_n\sqrt{1-\frac{c^2}{4m^2\omega_n^2}} = \sqrt{\frac{k}{m}} - \sqrt{\omega_n^2 - \frac{c^2}{4m^2}} \qquad (6)$$

where: Δf – the frequency difference
Comparing the equation parties were obtained:

$$\omega_n^2 - \frac{c^2}{4m^2} > 0 \Leftrightarrow 4m^2\omega_n^2 - c^2 > 0 \qquad (7)$$

$$\sqrt{\omega_n^2 - \frac{c^2}{4m^2}} = \sqrt{\frac{k}{m}} - \Delta f \Leftrightarrow \omega_n^2 - \frac{c^2}{4m^2} = \frac{k}{m} - 2\sqrt{\frac{k}{m}}\Delta f - (\Delta f)^2 \qquad (8)$$

Thus:

$$c^2 = 4m^2\omega_n^2 - 4mk + 8\Delta f\sqrt{k} - 4m^2(\Delta f)^2 \qquad (9)$$

3.1 The impact of vibration isolation parameters on the dynamic state of rotating system

Correct choice of anti-vibration system for rotating machines is an extremely important issue. It affects not only the work of machines but also for its neighborhoods, including the comfort of people working in the vicinity of machine. Unfortunately, this aspect is often neglected. In order to illustrate the effect of the foundation construction on the vibration characteristics some experiment had been performed. Change in the stiffness of the machine foundation shown in Figure 2 had been implemented. During the experiment dynamic parameters of the machine bearings was carried out. Also vibration on ground in the vicinity of the working machine was measured. Figure 14 presents the frequency-amplitude spectra of accelerations recorded during machine operation with correct anti-vibration system.

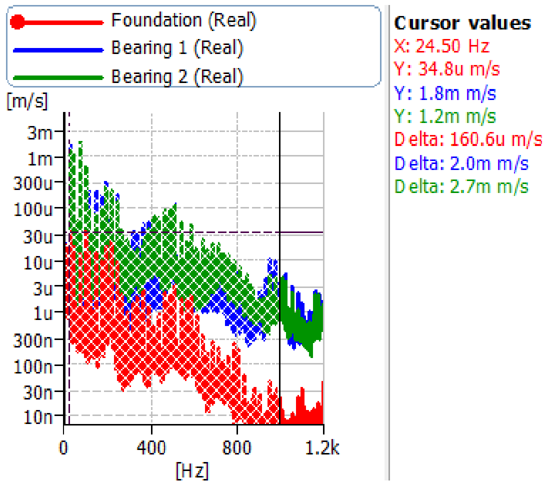

Fig. 14. Acceleration of vibration obtained during machine working with nominal stiffness of foundation

During the second stage of measurements foundation substitutive stiffness was changed to more stiff. Unbalance state of rotating system on all measurements was the same. Increasing of substitutive stiffness was reached by gain in initial tension of anti-vibration spring system – Fig 15.

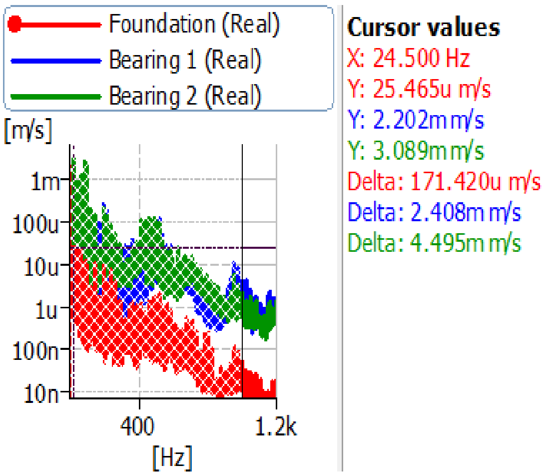

Fig. 15. Acceleration of vibration obtained during machine working with to stiff foundation

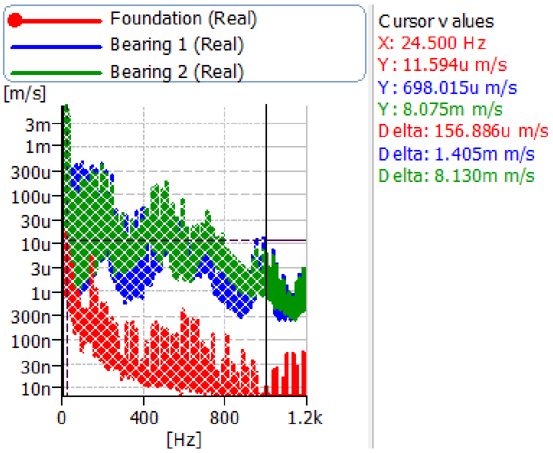

Fig. 16. Acceleration of vibration obtained during machine working with to soft foundation

Analyzing the spectra presented in Figures 14-16 it can be seen an impact of stiffness anti-vibration isolation on the dynamic characteristics of rotating system. The red spectra indicates acceleration of vibration amplitude recorded in the vicinity of the machine under test, blue spectra are vibrations recorded on bearing No. 1 and the green on bearing 2 (bearings markings are presented on Figure 2). Cursor values corresponds to the amplitudes of the rotating speed first harmonic (rotor had rotated at the speed of 1460 rev/min). Cursor delta values determine the average v_{rms} [mm/s] values of each spectra in the frequency range of 2 up to 1000 Hz. The lowest values of the vibration on the ground near the machine was recorded when the system worked with anti-vibration system characterized by high flexibility - Figure 16. However, in such situation dynamic state of the machine is the worst - the highest amplitude of vibration acceleration and the largest average value of v_{rms}. More preferred amplitudes was recorded on the bearings of the system during work with higher than nominal rigidity of vibration isolation system. Nevertheless, the vibration around machine are on the highest level - Fig. 15. Optimally matched anti-vibration system, is a compromise between satisfactory dynamic state of machines and low vibroactivity around it - Fig. 14. Amplitudes recorded on machine working with correct chosen vibration isolation system are lowest. On the other hand in this situation accelerations of vibration recorded in the vicinity of machine are larger than those presented on fig. 16.

4. CONCLUSION

The presented results of researches show a significant influence of damping to the resonance of rotor system. That influence is connected both with frequency of resonance as well as the maximum amplitude. It should be emphasized that this is the changing of the parameters of resonance of the entire system by introducing additional damping. The resonant frequency of the rotor remains unchanged, however, recorded resonance throughout whole system differs considerably. Changes in antivibration system has also significant influence on the neighbourhood of the machine. It was also proven the usefulness of order tracking in determining the resonance frequencies. Presented algorithm of calculations and measurements can be used in engine compartments of marine vessels. A significant number of adjacent rotating machines often causes local resonances in the ship's power plant. Applications of the proper damping pads may be one of the effective ways of detuning working devices in the engine compartment and solve the problem of resonance. On the other hand engineer have to be aware that the minimization of vibration parameters around the working machine might have negative influence on dynamic condition of isolated machine.

REFERENCES

1. Carswell W, Johansson J, Løvholt F, Arwade SR, Madshus C, DeGroot DJ, Myers AT. Foundation damping and the dynamics of offshore wind turbine monopoles, Renewable Energy; 2015; 80:724-736;
2. Dutkiewicz M, Radulski W, Zielek G. Analiza dynamiki turbozespołu w ujęciu przemysłowym [Dynamic analysis of steam turbines in the industrial point of view]; Tribologia; 2002; 1(181); In Polish
3. Harsha SP, Sandeep K, Prakash R. The effect of speed of balanced rotor on nonlinear vibrations associated with ball bearings. International Journal of Mechanical Sciences, 2003; 45:725-740.
4. Grządziela A. Vibroacoustic method of shafting coaxiality assessment of COGAG propulsion system of a vessel. Polish Maritime Researches, 1999; 3: 29-30.
5. Grządziela A. Vibration analysis of unbalancing of marine gas turbines rotors. Mechanika, 2004; 23(2): 187-194.
6. Grządziela A, Kluczyk M. An application of order tracking procedure for diagnosis technical state of rotor system in shut-down process. Journal of KONES Powertrain and Transport, 2013; 20(1): 111-117.

7. Karaś K. Analiza modalna – użyteczna metoda rozwiązywania problemów drganiowych [Modal analysis - a useful method for solving vibration problems], Nowa Energia, 2012; 4(28); In Polish
8. Kiciński J, Drozdowski R, Materny P. The nonlinear analysis of the effect of support construction properties on the dynamic properties of multi-support rotor Journal of Sound and Vibration, 1997; 206(4): 523–539.
9. Korbiel T. Order analysis in the diagnostic nonstationary vibroacustic process. Diagnostyka; 2007; 3 (43):9-103.
10. Krzyworzeka P, Adamczyk J, Cioch W, Jamro E. Monitoring of nonstationary states informatation machinery, Biblioteka Problemów Eksploatacji, Wydawnictwo ITeE, 2006.
11. Pedersen TF, Gade S, Harlufsen H, Konstantin-Hansen H. Order tracking in Vibro-acoustic Measurements: A Novel Approach Eliminating the Tacho Probe, „Technical Review”, 2006;1:15–28.
12. Randall RB. Vibration-based condition monitoring. A John Wiley and Sons, Ltd., Publication; 2011.
13. Randall RB, Coats MD, Smith WA. Repressing the effects of variable speed harmonic orders in operational modal analysis. Mechanical Systems and Signal Processing, 2016; 79: 3-15. doi:/10.1016/j.ymssp.2016.02.042.
14. Witoś M. Zwiększenie żywotności silników turbinowych poprzez aktywne diagnozowanie i sterowanie [Increasing the durability of turbine engines through active diagnostics and control], Research Works of Air Force Institute of Technology; 2011; 29; doi: 10.13140/RG.2.1.4341.4560. In Polish
15. Żółtowski B. Basics in technical diagnostics Bydgoszcz 1996. In Polish.
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