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Mechano-Mathematical Model and Exprimental Results on the Excitation of The Ship's Hull Oscillations by the Marine Propeller

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Abstract. **This study tackles a critical issue in ship design: propeller-induced vibrations in the hull. We propose a novel mechano-mathematical model that goes beyond previous approaches to elucidate the excitation of these oscillations. Our model incorporates the crucial dynamic connection between the propeller's rotational motion and its oscillations. This refinement allows for a more precise understanding of how propellers, even when mechanically balanced, can excite vibrations through hydrodynamic imbalances.**

The traditional approach relies on simpler models, often leading to qualitative analysis. This work advances the field by introducing a more robust mechano-mathematical framework. This framework considers the complex interaction between the main engine, shaft line, and propeller, including factors like engine torque variations, propeller moment, and elastic properties of the system.

The model is not just theoretical. We present experimental results that validate the spectral components predicted by our model. This successful validation demonstrates the model's accuracy in capturing the real-world dynamics of propeller-induced vibrations.

The practical implications of this work are significant. By pinpointing the excitation mechanisms with greater precision, this research can pave the way for the development of improved propeller designs that minimize vibrations. Reduced vibrations translate to enhanced crew comfort, lower maintenance requirements, and potentially even improved fuel efficiency. Additionally, the model can be a valuable tool for optimizing ship powerplant design to ensure smooth operation and extended lifespan.

Keywords: oscillations, ship's hull, marine propeller

I. INTRODUCTION

It is known that the marine propeller is the exciter of the oscillations of the ship's hull [1, 2, 5, 6, 7, 8, 10, 11].

The oscillations are excited by mechanical imbalance (the center of mass does not coincide with the axis of rotation) and are of reversible frequency. In order not to excite such oscillations, the marine propellers are statically balanced on special stands. A mechanically balanced marine propeller can excite oscillations if the mass center of the water attached during the marine propeller's rotation does not coincide with the axis of rotation. The condition that the mass center of the joined mass of water coincides with the axis of rotation is geometric uniformity of the blades. This imbalance of the marine propeller is called hydrodynamic. A solid-state propeller may have a hydrodynamic imbalance.

There is also hydrodynamic imbalance in the line of hydrodynamic forces generated by the propeller blades. With the same blade geometry, the tangential hydrodynamic forces have a principal vector equal to zero. When the geometry of blades is disturbed, the main vector of the tangential hydrodynamic forces is a vector rotating with the rotational frequency. During the operation of the propeller, due to the inhomogeneous hydrodynamic field around the stern of the ship, hydrodynamic forces are generated with blade frequency ($z\omega$) equal to the product of the number of blades (z) multiplied with frequency of rotation ω of propeller.

The moment of the propeller contains harmonics of order $z, 2z$ and $3z$. The amplitudes of harmonics are determined by Brahms formulas [1] as a function of the mean torque.

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Table 1 Propeller Moment Harmonics

z	M_z	M_{2z}	M_{3z}
3	$0,042M_{cp}$	$0,042M_{cp}$	$0,027M_{cp}$
4	$0,076M_{cp}$	$0,026M_{cp}$	$0,08M_{cp}$
5	$0,09M_{cp}$	$0,015M_{cp}$	0

In [5], [6] the possibility of excitation of oscillations by the propeller along the line of dynamic connection between oscillations and rotational motion is proved. The proof is rather qualitative in nature, as far as the simplest mechano-mathematical model is used. The main purpose of this present work is to prove the possibility of excitation of oscillations by a propeller along the line of the dynamic connection between the oscillations and the rotational motion by a proposed more precise mechano-mathematical model.

II. Materials and Methods

The propeller is a rotating rigid body for which the dynamic connection proved in [3] between the oscillation of an unbalanced rotor and its rotation is valid. On this basis, in [5] the simplest dynamic model is proposed with which it is possible to study the oscillations of the propeller, considering the dynamic relation between the oscillations of the propeller and the torsional oscillations.

Fig. 1 shows a specified dynamic model of the system – main engine, shaft line, and propeller.

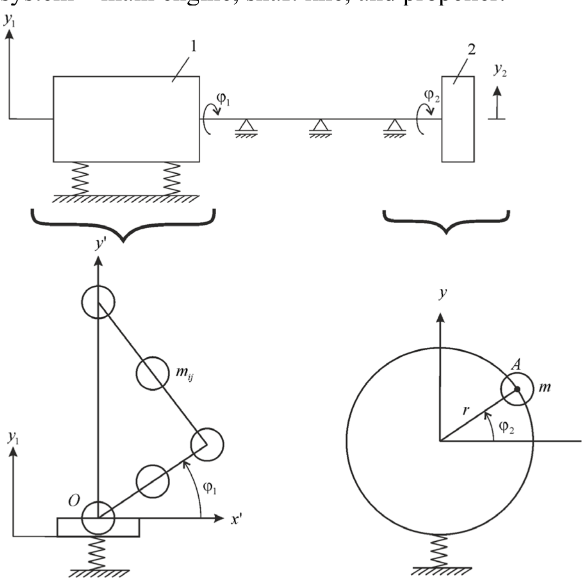

Fig.1 Dynamic Model of the Main Engine, Shaft Line, and Propeller

The main motor 1 is mounted elastically. It performs two oscillating processes simultaneously:

1. General oscillation along an y axis, excited by the unbalanced inertial forces.
2. Unequal rotation due to variable engine torque $M_д$ and propeller torque $M_в$.

These two vibrational processes interact [1, 2, 4, 7, 8]. General oscillation creates additional inertial forces of all moving parts of the unit, which creates an additional moment influencing the law of rotational motion. On the other hand, unequal rotation creates additional inertial forces that excite the oscillations of the unit. In this regard, a mechano-mathematical model [1, 2, 4, 7, 8] has been created, which describes these two vibrational processes in their natural interaction. This mechano-mathematical model will be used in the present work. The propeller is modeled as a rotating mass with mass moment of inertia. The imbalance of the screw is modeled by a mass m of a distance r from the axis of rotation.

The engine consists of crank shaft mechanisms (Fig. 1б) with n number of relatively mobile units.

Each i unit is modeled through three material particles m_{ij} $(i = 1, 2, ..., n; j = 1, 2, 3)$ of the dynamic equivalence condition [4]. We will introduce a coordinate system $x'oy'$ fixed to the stand. The position of each point mass m_{ij} in the movable in the movable coordinate system is determined by the coordinates x_{ij}, y_{ij}, that are function of position φ_1 of the crank shaft mechanism. The position of the movable coordinate system is determined by the coordinate y_1, that defines the oscillation of the engine. The oscillations of the screw are defined by y_2, and its rotation by φ_2. The shaft line has torsional stiffness c_0. The oscillations of the system are described by the differential equations:

$$\begin{cases} I_1\ddot{\varphi}_1 + M\ddot{y}_1 + \frac{1}{2}\dot{\varphi}_1^2\frac{dI_1}{d\varphi_1} + c_0(\varphi_1 - \varphi_2) = M_1 \\ m_1\ddot{y}_1 + M\ddot{\varphi}_1 + \dot{\varphi}_1^2\frac{dM}{d\varphi_1} + c_{11}y_1 + c_{12}y_2 = F_1 \\ (m + m_2)\ddot{y}_2 + c_{21}y_1 + c_{22}y_2 = -mr(\ddot{\varphi}_2\cos\varphi_2 - \\ -\dot{\varphi}_2^2\sin\varphi_2) + F_2 \\ (I_2 + mr^2)\ddot{\varphi}_2 - c_0(\varphi_1 - \varphi_2) = -mr\ddot{y}_2\cos\varphi_2 + M_2 \end{cases} \quad (1)$$

Here I_1 - mass moment of inertia of the engine, which is generally a function of φ_1:

$$I_1 = I_1(\varphi_1) = I_{10} + \Delta I_1(\varphi_1)$$

$$M = \sum_{i=1}^{n}\sum_{j=1}^{3} m_{ij}\frac{dy_{ij}}{d\varphi_1} = M(\varphi_1)$$

M_1 - engine torque

M_2 - moment of propeller.

F_1 - main vector of the inertial forces of the engine.

F_2 - hydrodynamic force.

Elastic constants $c_{ij}(i, j = 1, 2)$ define the transverse stiffness of the shaft line, including the stiffness of the elastic suspension of the engine:

$$c_{11} = \frac{\delta_{22}}{\Delta}, \qquad c_{12} = -\frac{\delta_{12}}{\Delta}$$

$$c_{21} = -\frac{\delta_{21}}{\Delta}, \qquad c_{22} = \frac{\delta_{11}}{\Delta}$$

$$\Delta = \delta_{11}\delta_{22} - \delta_{12}^2$$

With δ_{ij} are signed coefficients of influence – deformation of i section under the action of unit force applied in j section. Differential equations (1) describe propeller's oscillations considering the dynamic connection with the torsional oscillations excited by the motor and the propeller.

If we accept:

$$\dot{\varphi}_1^2 = \omega_1^2 = const \qquad (2)\ (a)$$

$$\cos\varphi_2 \approx \cos\omega_2 t, \qquad \sin\varphi_2 \approx \sin\omega_2 t \qquad (2)\ (b)$$

equation (1) is significantly simplified.

It should be noted that the assumption (2-b) is traditional for machine dynamics.

Based on (2) the system of equations takes the form:

$$\begin{cases} I_1\ddot{\varphi}_1 + M\ddot{y}_1 + c_0(\varphi_0 - \varphi_2) = M_1 - \frac{1}{2}\omega_1^2 \frac{dI_1}{d\varphi_1} \\ m_1\ddot{y}_1 + M\ddot{\varphi}_1 + c_{11}y_1 + c_{12}y_2 = F_1 - \omega_1^2 \frac{dM}{d\varphi_1} \\ (m+m_2)\ddot{y}_2 + c_{21}y_1 + c_{22}y_2 = -mr(\ddot{\varphi}_2\cos\omega_2 t - \\ -\omega_2^2\sin\omega_2 t) + F_2 \\ (I_2 + mr^2)\ddot{\varphi}_2 - c_0(\varphi_1 - \varphi_2) = -mr\ddot{y}_2\cos\omega_2 t + M_2 \end{cases} \qquad (3)$$

III. Results and Discussion

We will perform a qualitative analysis on the influence of the moments of the engine and the propeller on the vibration state of ship's power plant.

We will present the moment of engine only with the main harmonic.

$$M_1 = M_{1k}\sin k\omega_1 t \qquad (4)$$

We will present the torque of the screw with the main harmonic of the order of the number of blades z :

$$M_2 = M_{2z}\sin z\omega_2 t \qquad (5)$$

Moments (4) and (5) excite torsional oscillations:

$$\varphi_i = A_{ik}\sin k\omega_1 t + A_{iz}\sin z\omega_2 t, \quad i = 1,2 \qquad (6)$$

Torsional oscillations generate inertial force:

$$\Phi_2 = -mr\ddot{\varphi}_2\cos\omega_2 t \qquad (7)$$

After substituting (6) in (7) we get:

$$\Phi_2 = \Phi_{21}[\sin(z\omega_2 - \omega_2)t + \sin(z\omega_2 + \omega_2)t] + \\ +\Phi_{22}[\sin(k\omega_1 - \omega_2)t + \sin(k\omega_1 + \omega_2)t] \qquad (8)$$

Here

$$\Phi_{21} = \frac{1}{2}A_{2z}mr(z\omega_2)^2$$

$$\Phi_{22} = \frac{1}{2}A_{2k}mr(k\omega_1)^2$$

The inertial force (8) excites the oscillations of the propeller with frequencies:

$$z\omega_2 \pm \omega_2 = (z \pm 1)\omega_2, \qquad k\omega_1 \pm \omega_2 \qquad (9)$$

Table 2 Deadwood & Intermediate Bearing Oscillations (Engine Modes)

Place of measurement	directions	Mode *rpm*	Characteristic frequencies of the spectrum	figure
Intermediate bearing left shaft line	H	900	$4\omega_1 \pm \omega_2 = 3600 \pm 178$	Fig. 2
Intermediate bearing left shaft line	V	900	$4\omega_1 \pm \omega_2 = 3600 \pm 178$	Fig. 3
Intermediate bearing left shaft line	A	900	$4\omega_1 \pm \omega_2 = 3600 \pm 178$	Fig. 4
Deadwood bearing left shaft line	H	900	$4\omega_1 + \omega_2 = 3600 + 178$	Fig. 5
Deadwood bearing left shaft line	V	900	$z\omega_2 = 712$; $5\omega_1 - \omega_2 = 4500 - 178$	Fig. 6
Deadwood bearing left shaft line	H	1300	$z\omega_2 = 1029$	Fig. 7
Intermediate bearing left shaft line	H	1300	$4\omega_1 \pm \omega_2 = 5200 \pm 257$	Fig. 8
Intermediate bearing left shaft line	V	1300	$4\omega_1 \pm \omega_2 = 5200 \pm 257$	Fig. 9
Deadwood bearing left shaft line	V	1300	$z\omega_2 = 1029$	Fig. 10
Deadwood bearing right shaft line	H	1500	$\omega_2 = 297$; $z\omega_2 = 1188$; $z\omega_2 + \omega_2 = 1485$	Fig.11
Deadwood bearing right shaft line	V	1500	$z\omega_2 = 1188$; $z\omega_2 \pm \omega_2 = 1188 \pm 297$	Fig. 12
Deadwood bearing right shaft line	A	1500	$z\omega_2 = 1188$; $z\omega_2 + \omega_2 = 1485$	Fig. 13
Intermediate bearing left shaft line	V	1500	$3\omega_1 + \omega_2 = 4500 + 297$	Fig. 14
Deadwood bearing left shaft line	H	1500	$\omega_2 = 297$; $z\omega_2 = 1188$; $z\omega_2 + \omega_2 = 1485$	Fig. 15
Deadwood bearing left shaft line	V	1500	$z\omega_2 \pm \omega_2 = 1188 \pm 297$	Fig. 16
Deadwood bearing left shaft line	A	1500	$z\omega_2 \pm \omega_2 = 1188 \pm 297$	Fig. 17

A. Experimental investigations

The object of experimental investigations are shaft lines of port tugs. The main power plant consists of two Caterpillar engines 35008B. The connection to the shaft line is made with gearboxes with a gear ratio $i = 5.05$. The oscillation of the deadwood bearing and the intermediate bearing of each shaft line for three modes of main engine corresponding to $900rpm, 1300rpm, 1500rpm$.

The fourth harmonic of the gas forces is the lowest frequency harmonic that has a dominant influence.

Some of the oscillation spectrums are given in the figures described in Table 1.

The oscillations with frequencies are clearly visible in the figures:

$$z\omega_2 \pm \omega_2 = (z \pm 1)\omega_2$$
$$k\omega_1 \pm \omega_2 \qquad (9)$$

the existence of which has been proven theoretically.

According to the existing notations of excitation of oscillations from the propeller, we should observe in the spectrum of oscillations frequencies.

$$z\omega_2, \quad k\omega_1, \omega_2 \qquad (10)$$

The presented experimental results show that in addition to the traditional spectral components (10), the frequency oscillations are also observed (9).

Precise identification of the spectral components is required in connection with the evaluation of the operation of the main engine (evaluation of the fuel adjustment) according to the spectrum of oscillations, [3].

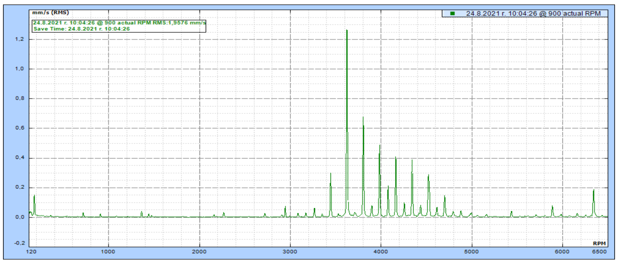

Figure 2 Spectrum of oscillations of Intermediate bearing left shaft line.

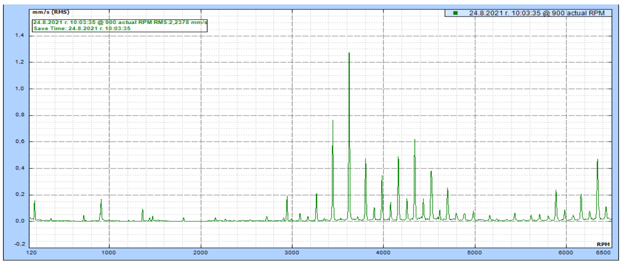

Figure 3 Spectrum of oscillations of Intermediate bearing left shaft line.

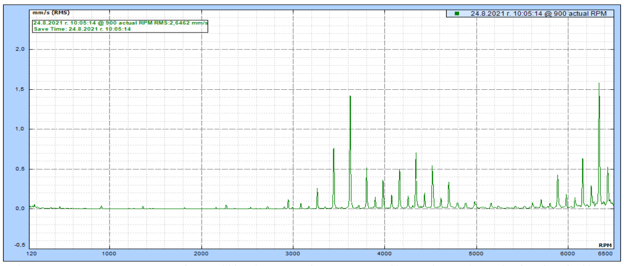

Figure 4 Spectrum of oscillations of Intermediate bearing left shaft line.

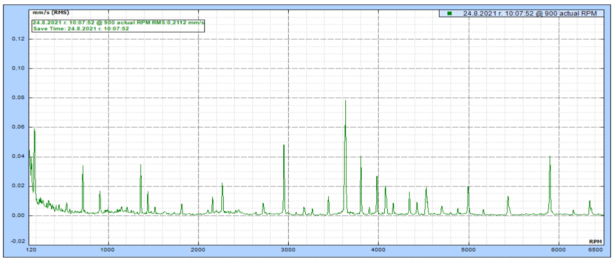

Figure 5 Spectrum of oscillations of Deadwood bearing left shaft line.

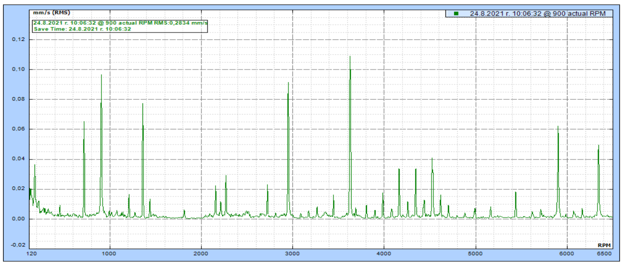

Figure 6 Spectrum of oscillations of Deadwood bearing left shaft line.

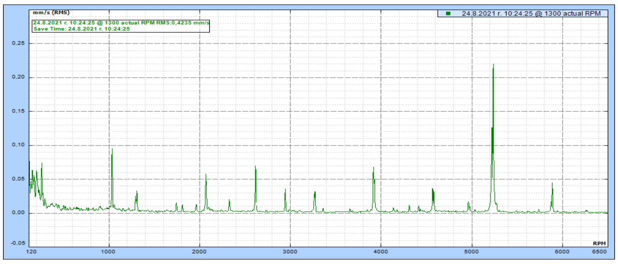

Figure 7 Spectrum of oscillations of Deadwood bearing left shaft line.

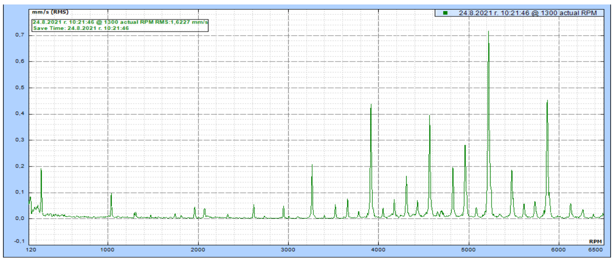

Figure 8 Spectrum of Intermediate bearing left shaft line.

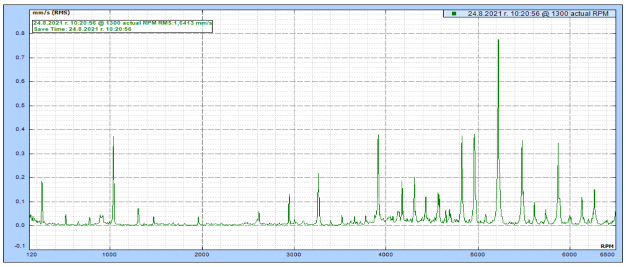

Figure 9 Spectrum of Intermediate bearing left shaft line.

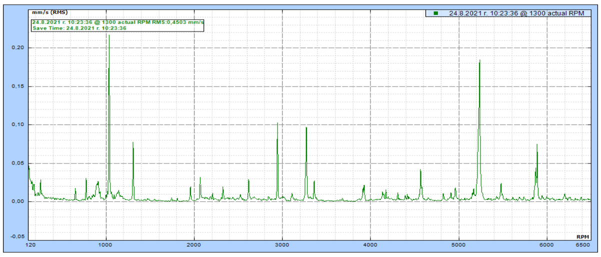

Figure 10 Spectrum of oscillations of Deadwood bearing right shaft line.

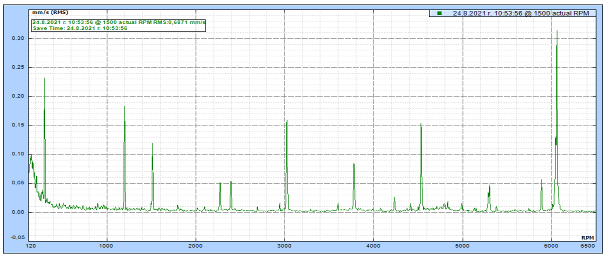

Figure 11 Spectrum of oscillations of Deadwood bearing right shaft line.

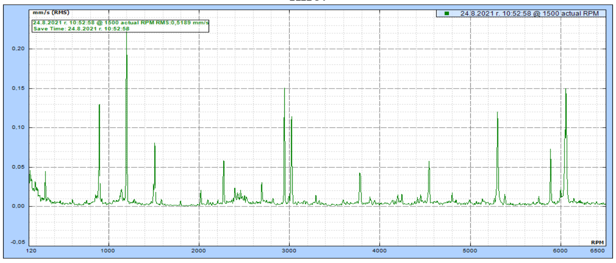

Figure 12 Spectrum of oscillations of Deadwood bearing right shaft line.

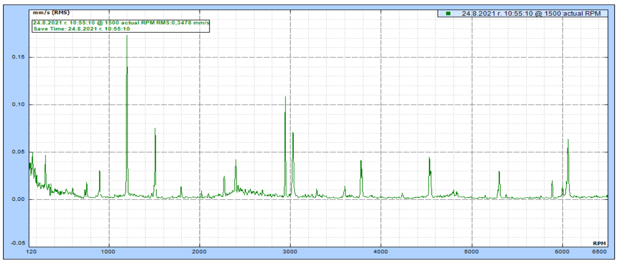

Figure 13 Spectrum of oscillations of Deadwood bearing right shaft line.

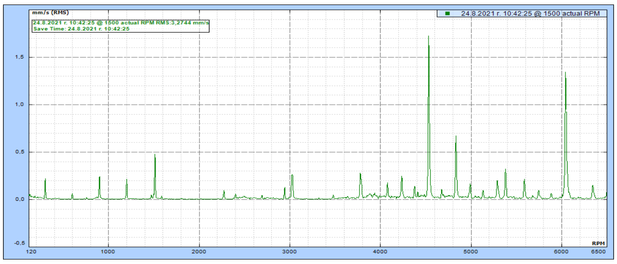

Figure 14 Spectrum of oscillations of Intermediate bearing left shaft line.

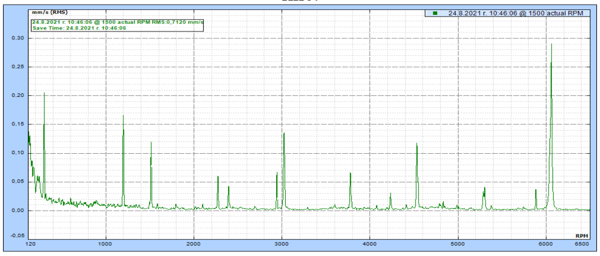

Figure 15 Spectrum of oscillations of Deadwood bearing left shaft line.

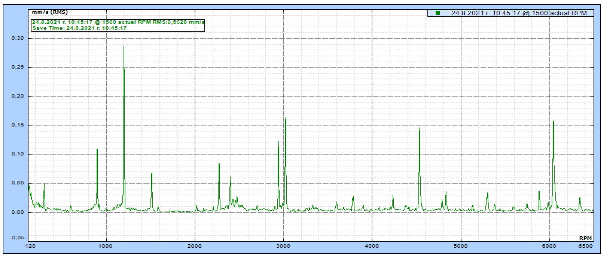

Figure 16 Spectrum of oscillations of Deadwood bearing left shaft line.

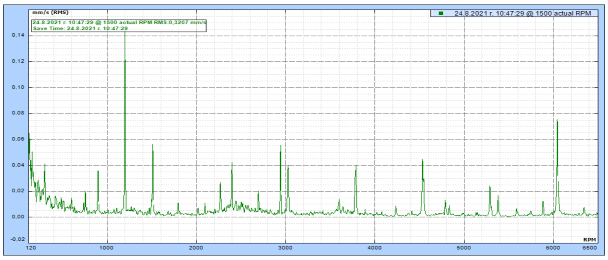

Figure 17 Spectrum of oscillations of Deadwood bearing left shaft line.

IV. CONCLUSION

The presented experimental results confirm the theoretically obtained spectral components (9), which is a result of the dynamic relation between the torsional and general oscillations of the propeller. In conclusion, this study presents a significant advancement in understanding propeller-induced hull oscillations. The proposed mechano-mathematical model, along with its experimental validation, offers a powerful tool for designing propellers and ship powerplants that minimize vibrations, leading to a range of practical benefits.

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