Mechano-Mathematical Model And Experimental Results On The Dynamic Relation Between Marine Propeller Oscillation And Main Ship Engine

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Abstract. **Ship propellers, essential for propulsion, can cause unwanted hull vibrations. These vibrations affect crew comfort, increase maintenance costs, and potentially reduce fuel efficiency. This study addresses this issue with a novel mechano-mathematical model that analyses the propeller contribution to vibrations. Previous models have focused primarily on the engine as the vibration source. This work bridges the gap by considering the dynamic interplay between different vibration modes within the entire propulsion system: engine, shaft line and propeller. It considers key elements such as engine torque variations, propeller moment and system elasticity. The model represents these components with the propeller as a weightless elastic element. It meticulously accounts for the complex motion of the propeller, allowing its absolute speed to be calculated. The model recognises the flexibility of the shaft line and represents the engine using simplified point masses based on established principles. A reference system tracks the position of each component, capturing engine vibration, stern tube bearing motion and shaft line torsional stiffness. A set of differential equations governs the oscillations of the system, incorporating relevant factors such as engine inertia and torque, propeller moment, and inertial and hydrodynamic forces. The model captures the dynamic relationship between different modes of vibration, including propeller, engine, shaft precession and torsional vibrations from both engine and propeller. While detailed equations and experimental results are omitted, a qualitative analysis demonstrates the model's ability to predict the frequencies of real-world vibration spectra. This successful validation highlights its potential to capture propeller-induced vibration dynamics. By pinpointing excitation mechanisms with greater accuracy, this research can pave the way for improved propeller designs that minimise vibration, leading to improved crew comfort, reduced maintenance costs and potentially even improved fuel efficiency.**

Keywords: oscillations, marine propeller, ship's engine

I. INTRODUCTION

The marine propeller is an element of the shaft line of the ship, which performs torsional oscillations, [1], [2]. The main exciter of torsional oscillations is the main ship's engine. The calculation of the torsional oscillations and their standardization is regulated by the classification societies. The marine propeller is also the exciter of torsional oscillations [1], [2] due to the inhomogeneity of the hydrodynamic field around the stern of the ship. In the presence of an imbalance, the marine propeller excites oscillations of the shaft line and the ship's hull. The oscillations of a rigid unbalanced rotor, considering the dynamic relation between the oscillations of the rotor and rotational motion, were first studied in [3]. The presence of torsional vibration exciter in ship's shaft line and dynamic relation between the oscillations of an unbalanced rotor and its rotation proved in [3] define the need to study the oscillations of propellers considering dynamic relation between propeller oscillations and torsional oscillations excited from ship's engine and a screw. In [5], [6] a solution of this problem is given based on elementary dynamic models. In [11] a more precise mechano-mathematical model was proposed, which, however, cannot explain some experimentally observed vibration phenomena. Therefore, in the present work, based on [9], a mechano-mathematical model is proposed in which the dynamic relation between the oscillations of

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the processing elastic shaft, the oscillations of the supports and torsional oscillations is considered.

II. MATERIALS AND METHODS

The propeller is modeled as a rotating rigid body with a dynamic connection to its rotation, like unbalanced rotors [3]. Building upon this concept, a simple dynamic model is introduced in [5] to study propeller oscillations while considering the dynamic relationship between propeller and torsional oscillations.

Fig. 1 shows a dynamic model of the system main engine, shaft line, and marine propeller. The rotor is weightless elastic (Fig. 1-b) with concentrated mass m.

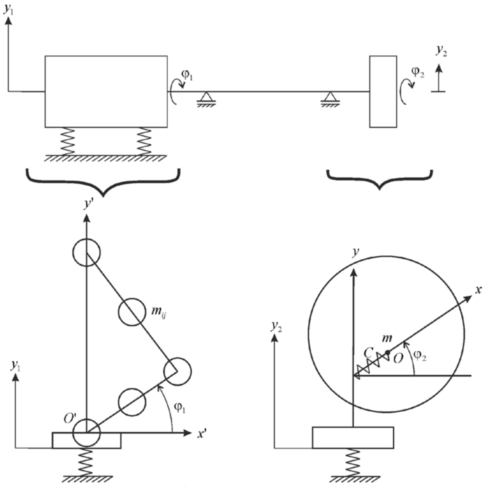

Fig. 1 Dynamic Model of the main engine, shaft line and marine propeller.

The elasticity of the shaft line is denoted by C, and elastic deformation of the shaft in cross section of the screw with x. Such a model of a processing rotor is proposed in [9]. The elastically deformed shaft rotates with ω_2, and its position is indicated by φ_2, (fig.1-b). The point O, at which the mass of the screw is concentrated performs a complex motion – transferred translation with velocity $\dot{y}_2$, relative rotation φ_2, accompanied by relative displacement x. Taking these factors into account for the absolute velocity of the point mass m we obtain:

$$v^2 = (\dot{y}_2 + x\dot{\varphi}_2 \cos\varphi_2 + \dot{x}\sin\varphi_2)^2 + \\ +(\dot{x}\cos\varphi_2 - x\dot{\varphi}_2 \sin\varphi_2)$$

The shaft line performs bending oscillations. The engine consists of crank shaft mechanisms with n number relatively mobile units (Fig 1-b). Each unit i is modeled through three-point masses m_{ij} $(i = 1, 2, ..., n, j = 1, 2, 3)$ of the dynamic equivalence condition [4]. We will introduce a coordinate system $x'o'y'$ fixed to the stand.

The position of each point mass m_{ij} in movable coordinate system is determined by to coordinates $x_{ij}(\varphi_1)$, $y_{ij}(\varphi_1)$ which are function of the position φ_1 of the crank shaft mechanism. The position of movable coordinate system is defined by the coordinate y_1, that defines the oscillation of the engine. The oscillation of stern tube bearing are determined by y_2. The shaft line has torsional stiffness c_o.

The oscillations of the system are described by the differential equations:

$$\begin{aligned}
&I_1\ddot{\varphi}_1 + M\ddot{y}_1 + \frac{1}{2}\dot{\varphi}_1^2 \frac{dI_1}{d\varphi_1} + c_0(\varphi_1 - \varphi_2) = M_1 \\
&m_1\ddot{y}_1 + M\ddot{\varphi}_1 + \dot{\varphi}_1^2 \frac{dM}{d\varphi_1} + c_{11}y_1 + c_{12}y_2 = F_1 \\
&(m_0 + m)\ddot{y}_2 + c_{21}y_1 + c_{22}y_2 = -2m\dot{x}\dot{\varphi}_2\cos\varphi_2 - \\
&-mx(\ddot{\varphi}_2\cos\varphi_2 - \dot{\varphi}_2^2\sin\varphi_2) - m\ddot{x}\sin\varphi_2 + F_2 \\
&(I_2 + mx^2)\ddot{\varphi}_2 - c_0(\varphi_1 - \varphi_2) = -m\ddot{y}_2 x\cos\varphi_2 - \\
&-2m\dot{x}\dot{\varphi}_2 x - M_2 \\
&m\ddot{x} + cx = -m\ddot{y}_2\sin\varphi_2 + mx\dot{\varphi}_2^2
\end{aligned} \tag{1}$$

Here

I_1 - reduced mass moment of inertia of the engine, which is 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 - reduced torque of the gas forces of the engine.

M_2 - moment of hydrodynamic forces of the propeller.

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

F_2 - main vector of hydrodynamic forces.

Differential equations (1) describe the oscillations of the marine propeller and the motor, considering the dynamic relation between the oscillations of the supports, the oscillations of the precession elastic shaft and the torsional oscillations excited by the motor and the propeller.

If accepted.

$$\begin{aligned}
&\dot{\varphi}_1^2 = \omega_1^2 = const \\
&\sin\varphi_2 \approx \sin\omega_2 t, \qquad \cos\varphi_2 \approx \cos\omega_2 t
\end{aligned} \tag{2}$$

The equations are greatly simplified. This assumption is traditional for machine dynamics.

Based on (2) the system of differential equations (1) takes the form:

$$\begin{cases} I_1\ddot{\varphi}_1 + M\ddot{y}_1 + c_0(\varphi_1 - \varphi_2) = M_1 - \frac{1}{2}\omega_1^2 \frac{dI_1}{d\varphi_1} \\ m_1\ddot{y}_1 + M\ddot{\varphi}_1 + c_{11}y_1 + c_{12}y_2 = F_1 - \omega_1^2 \frac{dM}{d\varphi_1} \\ (m_0 - m)\ddot{y}_2 + c_{21}y_1 + c_{22}y_2 = -2m\dot{x}\omega_2 \cos\omega_2 t - \\ -mx(\ddot{\varphi}_2 \cos\omega_2 t - \omega_2^2 \sin\omega_2 t) - m\ddot{x}\sin\omega_2 t + F_2 \quad (3) \\ (I_2 + mx^2)\ddot{\varphi}_2 - c_0(\varphi_1 - \varphi_2) = \\ = -m\ddot{y}_2 x\cos\omega_2 t - 2m\dot{x}\omega_2 x - M_2 \\ m\ddot{x} + cx = -m\ddot{y}_2 \sin\omega_2 t + 2mx\omega_2^2 \end{cases}$$

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

We will present the moment of the engine 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:

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

The screw excites oscillation with a blade frequency:

$$y_2 = B \sin z\omega_2 t \qquad (7)$$

These oscillations generate an inertial force.

$$\Phi_x = -m\ddot{y}_2 \sin\omega_2 t \qquad (8)$$

After replacing (7) in (8) it is obtained

$$\Phi_x = \frac{1}{2} m(z\omega_2^2) B[\cos(z\omega_2 - \omega_2)t - \cos(z\omega_2 + \omega_2)t] \; (9)$$

This inertial force excites the oscillations of the elastic rotor of the form:

$$X = a\cos(z\omega_2 - \omega_2)t + b\cos(z\omega_2 + \omega_2)t \quad (10)$$

The torsional oscillations of the elastic rotor generate inertial force.

$$\Phi_y = -mx\ddot{\varphi}_2 \cos\omega_2 t \qquad (11)$$

After substituting (6) and (10) in (11), the spectrum of inertial force (11) is obtained. The main component is frequency.

$$k\omega_1 \pm z\omega_2 \qquad (12)$$

In addition, there are components with frequencies:

$$k\omega_1 \pm z\omega_2 \pm 2\omega_2 \quad (13)$$

The inertial force (11) will excite oscillations with frequencies (12) and (13).

III. Results and discussion

The results of experimental studies are presented at Fig. 2 till Fig. 10.

The object of experimental investigation is shaft line of port tugs. The main power plant consists of two Caterpillar 3508B engines. The relation to the shaft line is made with a reducer with a gear ratio $i = 5.05$. The oscillations of the stern tube bearing and intermediate bearing in three directions (horizontal, vertical, axial) were measured for three modes of the main engine corresponding to $900rpm, 1300rpm, 1500rpm$.

The fourth harmonic of the engine's 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 figures clearly show oscillations with frequencies $k\omega_1 \pm z\omega_2$

Table 1 Oscillation Spectrums description.

Place of measurement	**Direc-tion**	**Mode** rpm	**Characteristic frequencies of spectrum**	**Figure**
Stern tube bearing right shaft line	H	900	$4\omega_1 - z\omega_2 =$ $= 3600 - 712 =$ $= 2888$	Fig. 2
Stern tube bearing right shaft line	V	900	$4\omega_1 - z\omega_2 =$ $= 3600 - 712 =$ $= 2888$	Fig. 3
Stern tube bearing right shaft line	A	900	$4\omega_1 - z\omega_2 =$ $= 3600 - 712 =$ $= 2888$	Fig. 4
Stern tube bearing left	H	900	$4\omega_1 - z\omega_2 =$ $= 3600 - 712 =$ $= 2888$	Fig. 5
Stern tube bearing left shaft line	V	900	$4\omega_1 - z\omega_2 =$ $= 3600 - 712 =$ $= 2888$ $z\omega_2 = 712$	Fig. 6
Stern tube bearing left shaft line	A	900	$4\omega_1 - z\omega_2 =$ $= 3600 - 712 =$ $= 2888$	Fig. 7
Stern tube bearing right shaft line	V	1300	$3\omega_1 - z\omega_2 =$ $= 3900 - 1029 =$ $= 2871$ $z\omega_2 = 1029$	Fig. 8
Stern tube bearing right shaft line	A	1300	$3\omega_1 - z\omega_2 =$ $= 3900 - 1029 =$ $= 2871$ $z\omega_2 = 1029$	Fig. 9
Stern tube bearing left shaft line	V	1300	$3\omega_1 - z\omega_2 =$ $= 3900 - 1029 =$ $= 2871$ $z\omega_2 = 1029$	Fig. 10

the existence of which was proved theoretically because of qualitative research on the system of differential equations.

The presented experimental results show that in addition to the traditional spectral components with frequencies:

$$\omega_2, z\omega_2, k\omega_1 \qquad (14)$$

oscillations with frequencies are observed:

$$k\omega_1 \pm z\omega_2$$

The presence of these spectral components cannot be predicted by existing theory.

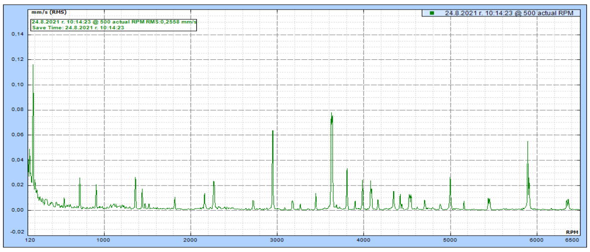

Figure 2 Spectrum of oscillations of Stern tube bearing right shaft line.

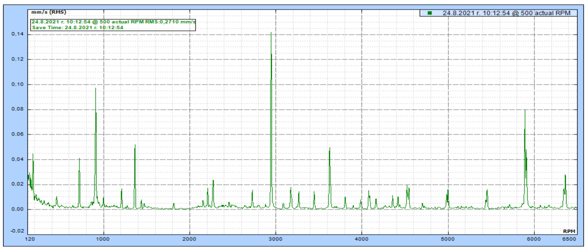

Figure 3 Spectrum of oscillations of Stern tube bearing right shaft line.

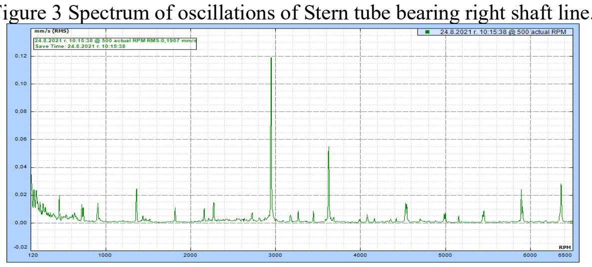

Figure 4 Spectrum of oscillations of Stern tube bearing right shaft line.

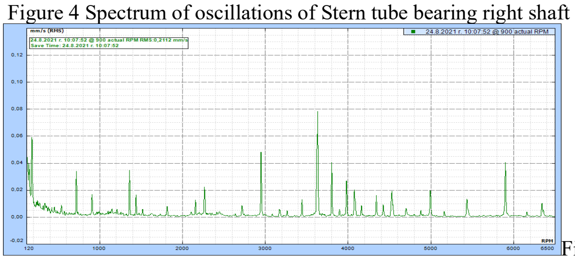

Figure 5 Spectrum of oscillations of Stern tube bearing left shaft line.

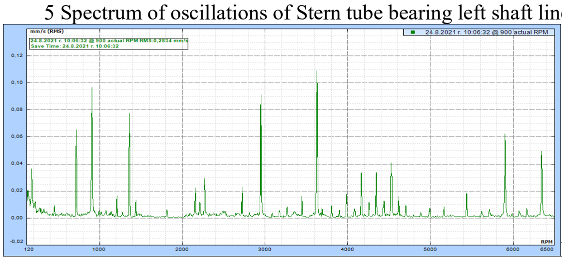

Figure 6 Spectrum of oscillations of Stern tube bearing LEFT shaft line.

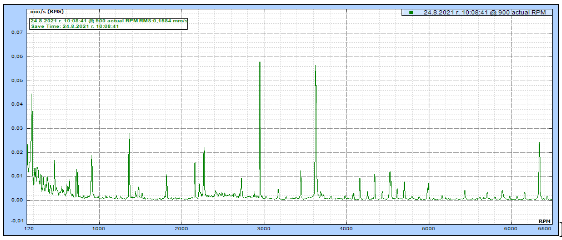

Figure 7 Spectrum of oscillations of Stern tube bearing LEFT shaft line.

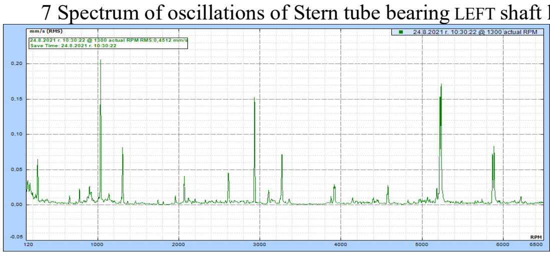

Figure 8 Spectrum of oscillations of Stern tube bearing right shaft line.

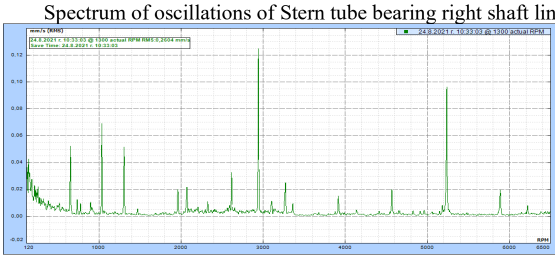

Figure 9 Spectrum of oscillations of Stern tube bearing right shaft line.

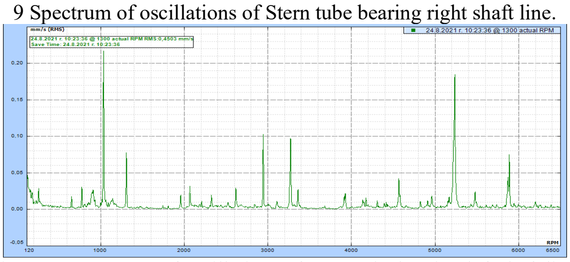

Figure 10 Spectrum of oscillations of Stern tube bearing left shaft line.

IV. Conclusion

The experimental results presented provide a strong validation of the spectral components predicted by the proposed mechano-mathematical model. This validation implies that the model is capable of accurately capturing the complex interplay between different modes of vibration within a ship's propulsion system. Critically, the model incorporates the crucial dynamic relationship between the vibrations of the propeller shaft (considered as a process elastic element), the vibrations of the supporting elements (such as the stern tube bearing) and the torsional vibrations arising from both the engine and the propeller itself. This focus on the dynamic interactions between these components represents a significant advance over previous models.

By successfully predicting real-world vibration spectra, this research provides a powerful tool for marine engineers. The model can be used to

- Optimize propeller design: By accurately identifying the excitation mechanisms responsible for vibration, the model can guide the design of propellers with minimized vibration signatures. This results in propellers that contribute to smoother operation, potentially leading to improved fuel efficiency and reduced underwater noise.

- Improve marine engine design: The model can be used as a valuable tool during the design phase of marine propulsion systems. By simulating the dynamic interactions between different components, engineers can identify potential vibration problems early and make design changes to mitigate them. This proactive approach can result in engines with superior operating characteristics, reduced maintenance requirements and, ultimately, extended service life.

-Improve vibration diagnosis and troubleshooting: The model can be used as a diagnostic tool for existing vessels experiencing vibration problems. By analyzing the measured vibration spectra and comparing them with the model predictions, engineers can pinpoint the cause of vibration with greater accuracy. This targeted approach allows for more effective troubleshooting and corrective action, minimizing downtime and maintenance costs.

In conclusion, the validated mechano-mathematical model represents a significant leap forward in the understanding of propeller-induced hull vibrations. The ability to predict real-world vibration spectra paves the way for significant advances in marine engineering, leading to the development of more efficient, reliable, and crew-friendly vessels.

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