DYNAMIC ANALYSES OF SHIPS SHAFTS LINES

Andrzej Grządziela

Naval University of Gdynia
Śmidowicza Street 69, Gdynia, Poland
tel.: +48 58 6262724
e-mail: a.grzadziela@amw.gdynia.pl

Abstract

Ships' propulsion plant usually works in a hard environment caused by static forces and permanent dynamic loads. Exciding of tolerated values of shaft alignments causes a damage of radial and thrust bearings in relative short time. Modelling of dynamical reactions could bring information to the designer for recognizing the level of hazard for propulsion system. Knowledge of a character of dynamic loading which affects ship shaft line can make it possible to identify potential failures by means of on-line vibration measuring systems. This way elimination of costly and time-consuming overhauls on dock leads to lowering operational costs and increasing ship fighting merits. A paper presents a proposal of identification of a degree of hazard to ship shaft line due to forces of shafts misalignment. A theoretical analysis was made of influence of changes in co - axiality of shafts resulting from elastic deformations of hull structure in vicinity of shaft bearing foundations. The main problem of naval vessels is a lack of dynamical requirements of stiffness of the hull. Modelled signals were recognized within sensitive symptoms of two sub models: model of propulsion system and model of shaft's misalignment. Both sub models allow testing forces and their responses in vibration spectrum using SIMULINK software.

Keywords: *transport, ship propulsions, dynamic analyse, propulsion system, SIMULINK software*

1. Introduction

Ship propulsion systems are subjected to specific sea loads due to waving and dynamical loads associated with mission of a given ship. Sea waving can be sufficiently exactly modelled by means of statistical methods. In operation of contemporary technical objects including naval ships greater and greater attention is paid to such notions as: time of serviceability, repair time, maintenance and diagnosing costs [1]. Diagnosing process has become now a standard procedure performed for technical maintenance. Out of the above mentioned the notions of time of serviceability and maintenance costs seem to be crucial for the diagnosing process of ship power plant. Knowledge of a character of dynamic loading which affects ship shaft line can make it possible to identify potential failures by means of on - line vibration measuring systems. This way elimination of costly and time - consuming overhauls on dock leads to lowering operational costs and increasing ship fighting merits.

2. Analysis of reactions forcing on shaft - line bearings

Ship shaft lines are subjected to loads in the form of forces and torques which generate bending, torsional and axial vibrations. In most cases strength calculations of driving shafts are carried out by using a static method as required by majority of ship classification institutions. Moreover, they require calculations of torsional vibrations which have to comply with permissible values, to be performed. Calculation procedures of ship shaft lines generally amount to determination of reduced stresses and safety factor related to tensile yield strength of material. The above mentioned methods do not model real conditions of shaft-line operation, which is confirmed by the character of ship hull response, i.e. its deformations under static and dynamic loads. Much more reliable would be to relate results of the calculations to fatigue strength of material instead of its yield strength [4]. In static calculation procedures no analysis of dynamic excitations, except torsional vibrations, is taken into consideration. In certain circumstances the adoption of static load criterion may be disastrous especially in the case of resonance between natural vibration frequencies of shafts line and those of external forces due to dynamic loads. To analyze the dynamic interaction a simplified model of shaft line is presented below, Fig. 1.

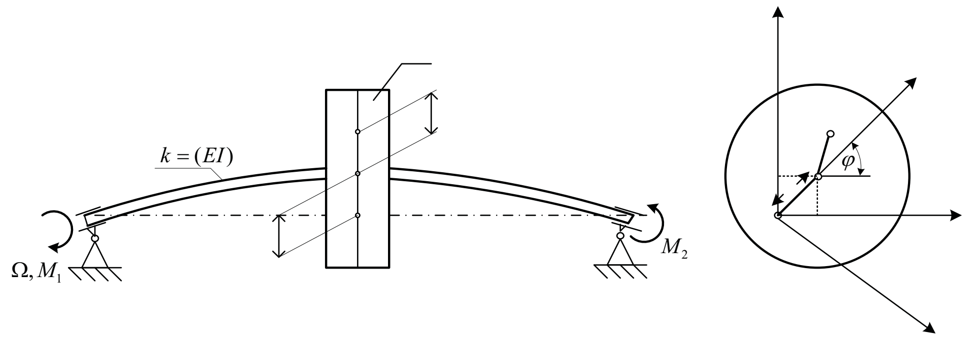

Fig. 1. A simplified shaft-line model for critical speed calculation [3] where: M_1 - torque, M_2 - anti-torque

The system can be represented by the following set of equations:

$$
\begin{aligned}
&m\ddot{h} + kh = me(\ddot{\varphi}\sin\varphi + \dot{\varphi}^2\cos\varphi),\\
&m\ddot{v} + kh = me(-\ddot{\varphi}\cos\varphi + \dot{\varphi}^2\sin\varphi),\\
&(J + me^2)\ddot{\varphi} = me(\ddot{h}\sin\varphi - \ddot{v}\cos\varphi) + M_1 - M_2.
\end{aligned}
\tag{1}
$$

The presented form of the equations is non - linear. Considering the third of the equations Eq. 1, one can observe that the variables h, v and φ are mutually coupled. It means that any bending vibration would disturb rotational motion of the shaft. The third block of the Eqs. 1 can be written also in the equivalent form as follows:

$$J\ddot{\varphi} = ke(v\cos\varphi - h\sin\varphi) + M_1 - M_2. \tag{2}$$

To obtain the shaft angular speed Ω_w constant to use time - variable torque is necessary:

$$M = M_1 - M_2 = ke(h\sin\varphi - v\cos\varphi). \tag{3}$$

Theoretical analysis indicates that shaft bending deformation continuously accumulates a part of shaft torque. However the quantity of torque non - uniformity is rather low since shaft-line eccentricity is low; it results from manufacturing tolerance, non - homogeneity of material, propeller weight and permissible assembling clearances of bearing foundations. Occurrence of such kind vibrations is conditioned by non-zero value of e, which - in the case of ship shaft line - appears just after dislocation of a weight along ship, a change of ship displacement or even due to sunshine operation on one of ship sides. A similar situation will happen when e varies due to dynamic excitations resulting from e.g. sea waving or underwater explosion. In this case the critical speed will vary depending on instantaneous value of e and damping.

Theoretical analysis of operational conditions of intermediate and propeller shafts indicates that static and dynamic loads appear. In a more detailed analysis of dynamic excitations of all

kinds the following factors should be additionally taken into consideration:
- disturbances coming from ship propeller,
- disturbances from propulsion engine (torsional and compressive stresses),
- disturbances from reduction gear (torsional stresses),
- disturbances from other sources characteristic for a given propulsion system or ship mission.

3. Theoretical background of shafts misalignment simulation

The dynamic calculation procedure needs analysis of excitations using theoretical model of ships propeller shaft - Fig. 2. Kinetic energy of presented system can be expressed as follow:

$$E_k = \frac{1}{2} I_N \dot{\varphi}_N^2 + \frac{1}{2} I_{SR} \dot{\varphi}_{SR}^2 + \frac{1}{2} m_I (\dot{v}_I^2 + \dot{h}_I^2) + \frac{1}{2} m_{II} (\dot{v}_{II}^2 + \dot{h}_{II}^2) + \\ + \frac{1}{2} m_{SR} (\dot{v}_{SR}^2 + \dot{h}_{SR}^2) + \frac{1}{2} m_N (\dot{v}_N^2 + \dot{h}_N^2) . \quad (4)$$

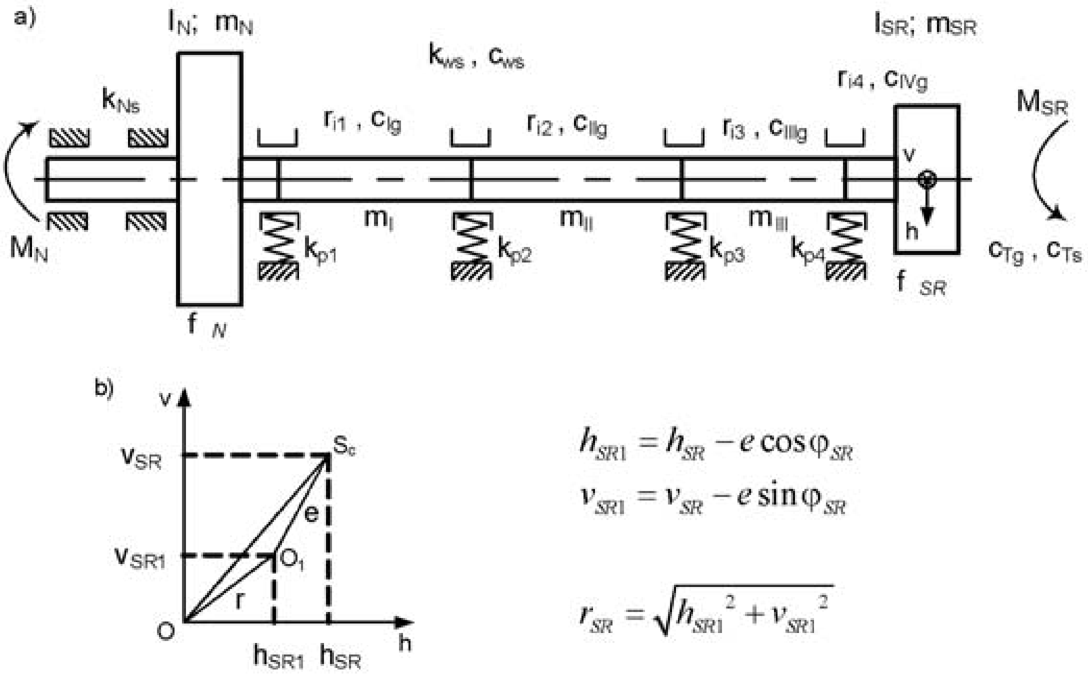

Fig. 2. Model of excitation of propeller shaft for simulation process where: I - moments of inertia, m – masses k - coefficients of stiffness, c - coefficients of damping, N - drive, SR - propeller, h - horizontal axis, v - vertical axis r - deflection of support, e - eccentricity of rotational shafts axis

For further analysis the following module of Eq. (4) was disregard:

$$(\frac{1}{2} m_N (\dot{v}_N^2 + \dot{h}_N^2)) . \quad (5)$$

The potential energy of torque transmission system can be present in the form (without taking into consideration Maxwell's coefficients) [3]:

$$E_p = \frac{1}{2} k_{NS} \varphi_N^2 + \frac{1}{2} k_{ws} (\varphi_{SR} - \varphi_N)^2 + \frac{1}{2} k_{Ig} (h_I^2 + v_I^2) + \frac{1}{2} k_{IIg} (h_{II}^2 + v_{II}^2) + \\ + \frac{1}{2} k_{IIIg} (h_{III}^2 + v_{III}^2) + \frac{1}{2} k_{IVg} (h_{SR1}^2 + v_{SR1}^2) . \quad (6)$$

$$E_R = \frac{1}{2}c_{ws}(\dot{\varphi}_{SR} - \dot{\varphi}_N)^2 + \frac{1}{2}c_{Ts}\dot{\varphi}_{SR}{}^2 + \frac{1}{2}c_{Ig}(\dot{h}_I^2 + \dot{v}_I^2) + \frac{1}{2}c_{IIg}(\dot{h}_{II}^2 + \dot{v}_{II}^2) + \\ + \frac{1}{2}c_{IIIg}(\dot{h}_{III}^2 + \dot{v}_{III}^2) + \frac{1}{2}c_{IVg}(\dot{h}_{SR1}^2 + \dot{v}_{SR1}^2) \quad . \tag{7}$$

Dissipation energy can be expressed as follow:
And external torques as follow:

$$Q = M_N - M_{SR} \,. \tag{8}$$

The equation of oscillation movement of propellers shaft was written using the Lagrange's II kind equations. All blocks of the equation set are depended from r_{ij} coefficients. Presented model of blocks are resolved by using SIMULINK software - Eq. (9).

$$\begin{aligned}
&I_N\ddot{\varphi}_N + c_{ws}(\dot{\varphi}_N - \dot{\varphi}_{SR}) + k_{NS}\varphi_N + k_{ws}(\varphi_N - \varphi_{SR}) = M_N \\
&m_I\ddot{h}_I + c_{Ig}\dot{h}_I + r_{11}h_I + r_{12}h_{II} + r_{13}h_{III} + r_{14}h_{SR} = 0 \\
&m_I\ddot{v}_I + c_{Ig}\dot{v}_I + r_{11}v_I + r_{12}v_{II} + r_{13}v_{III} + r_{14}v_{SR} = 0 \\
&m_{II}\ddot{h}_{II} + c_{IIg}\dot{h}_{II} + r_{21}h_I + r_{22}h_{II} + r_{23}h_{III} + r_{24}h_{SR} = 0 \\
&m_{II}\ddot{v}_{II} + c_{IIg}\dot{v}_{II} + r_{21}v_I + r_{22}v_{II} + r_{23}v_{III} + r_{24}v_{SR} = 0 \\
&m_{III}\ddot{h}_{III} + c_{IIIg}\dot{h}_{III} + r_{31}h_I + r_{32}h_{II} + r_{33}h_{III} + r_{34}h_{SR} = 0 \\
&m_{III}\ddot{v}_{III} + c_{IIIg}\dot{v}_{III} + r_{31}v_I + r_{32}v_{II} + r_{33}v_{III} + r_{34}v_{SR} = 0 \\
&I_{SR}\ddot{\varphi}_{SR} + c_{ws}(\dot{\varphi}_{SR} - \dot{\varphi}_N) + c_{Ts}\dot{\varphi}_{SR} + c_{IIIg}(\dot{h}_{SR}e\sin\varphi_{SR} - \\
&\dot{v}_{SR}e\cos\varphi_{SR} + e^2\dot{\varphi}_{SR}) + k_{ws}(\varphi_{SR} - \varphi_N) + \\
&r_{44}(h_{SR}e\sin\varphi_{SR} - v_{SR}e\cos\varphi_{SR}) = M_{SR} \\
&m_{SR}\ddot{h}_{SR} + c_{IVg}(\dot{h}_{SR} + \dot{\varphi}_{SR}e\sin\varphi_{SR}) + r_{41}h_I + r_{42}h_{II} + r_{43}h_{III} + \\
&r_{44}(h_{SR} - e\cos\varphi_{SR}) = 0 \\
&m_{SR}\ddot{v}_{SR} + c_{IVg}(\dot{v}_{SR} - \dot{\varphi}_{SR}e\cos\varphi_{SR}) + r_{41}v_I + r_{42}v_{II} + \\
&r_{43}v_{III} + r_{44}(v_{SR} - e\sin\varphi_{SR}) = 0 \quad .
\end{aligned} \tag{9}$$

4. Results of simulations

To identify vibration parameters coming from misalignment, in the point $N^0 1$ of shafts line model - Fig. 2, simulation tests were performed with the introducing different axis slopes and different rotational speeds of shaft. Important parts of simulation model are blocks of disturbances, drive and propeller. The driving torque and ani - torque of propeller, represented as harmonic functions, can contain disturbances coming from changes of technical state of engine and propeller. Simulations of time waveforms of both torques are illustrated in Fig. 3, 4.
A model of vibration in different points of shaft lines makes it possible to predict critical speeds and changes of technical states of shaft in wide spectrum of rotational speeds. Additionally, it can analyze such situation assumed different misalignment parameters. It is important task during process of dynamical testing of shafts construction. Comparison between two typical analyzed spectra of acceleration of vibration in the analyzed point of shafts line (point N^0 1 - the centre of gravity of mass m_I , Fig. 2) is presented in the Fig. 5, 6.

5. Model identification

The important part of machine modelling is identification. The process of models identification is made in time and frequency domains. Acceptable relative errors were less than 5% in the frequency domain and 10% in the time domain during models verification. Presented model is dedicated for diagnosing procedures so the main goal is to verify its sensitivity. First step is analyse monitored symptoms of technical state in different operational parameters of shaft line. The second harmonic of velocity of vibration which represented shafts misalignment should change their frequency attribute with rotational speed of shaft. Next step is analysing of interdependence between value of symptom of missalignment and axis slope of shaft. Compare between modelled and measured spectra of velocity vibration in the point No. 1 are presented in Fig. 6, 7. It is clear that both steps confirm good sensitivity of shaft model.

Final step of models identification is verification of its insensible for the environment disturbances. The Fig. 8, 9 present effects of disturbances coming from sea waves. The hull contacts reaction has their specific characteristic frequency. There is not influence of this signal for values of misalignment vibration symptom.

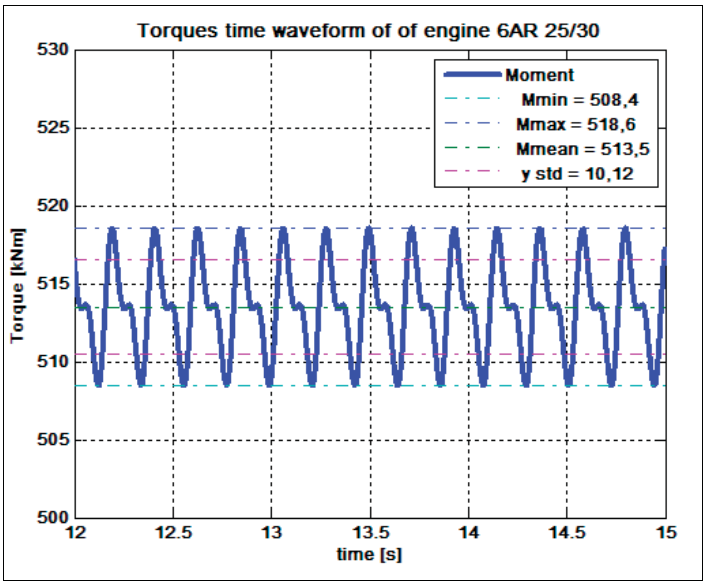

Fig. 4. Simulations of time waveform of driving torque, M = 100% load

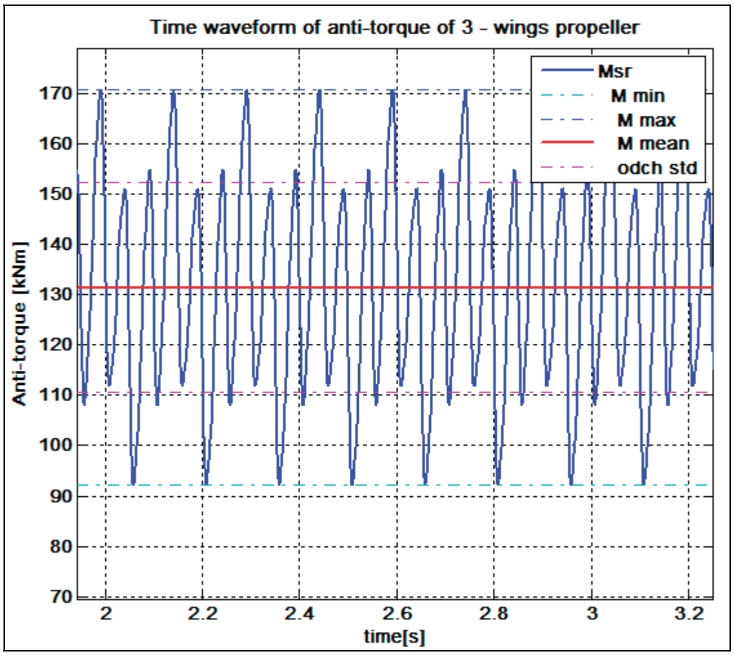

Fig. 5. Simulations of time waveform of anti - torque, M = 35% load

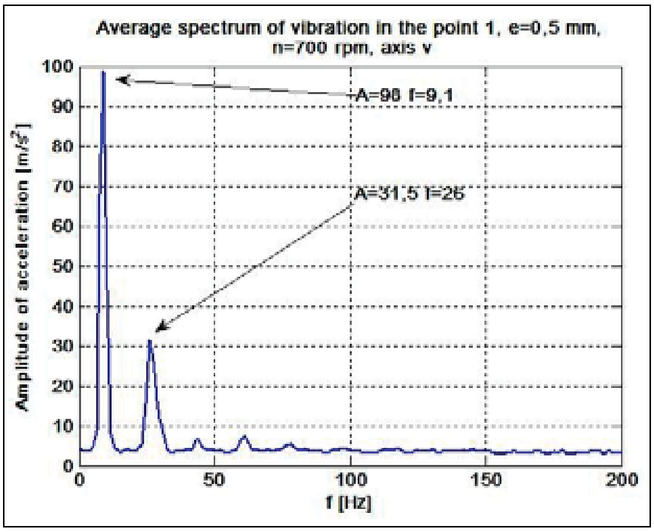

Fig. 6. Average spectrum of acceleration, n = 460 rpm, e = 0.5 mm

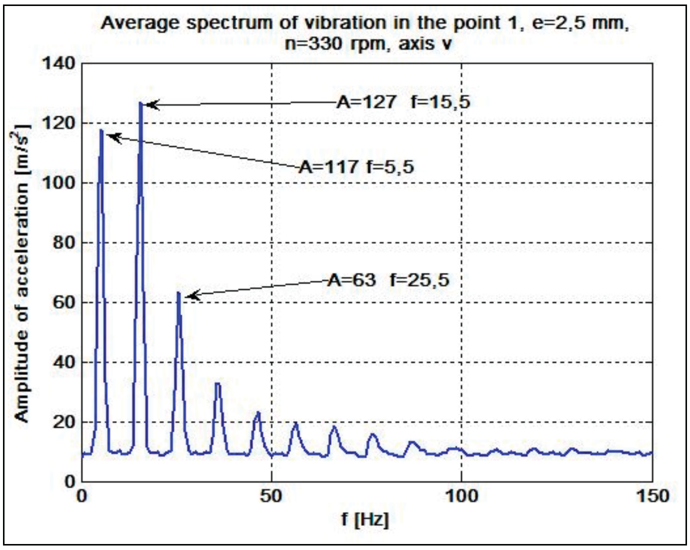

Fig. 7. Average spectrum of acceleration, n = 330 rpm, e = 2.5 mm

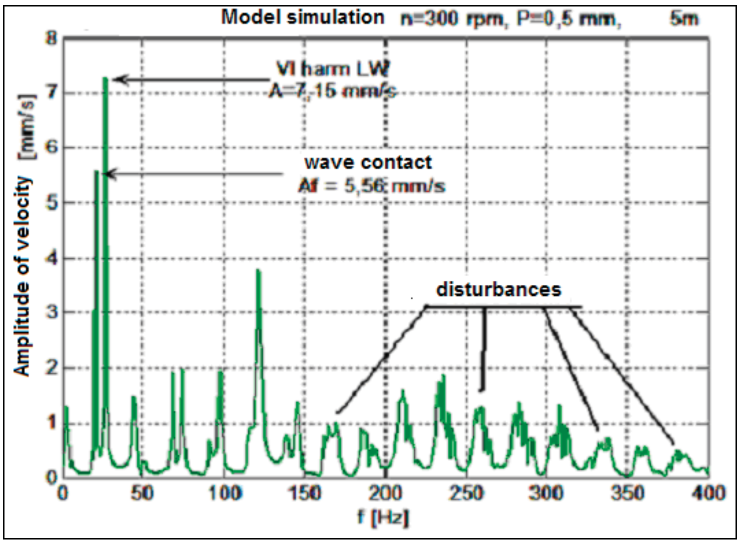

Fig. 8. Average spectrum of velocity, n = 300 rpm, e = 0.5 mm, wave amplitude 5 m (model)

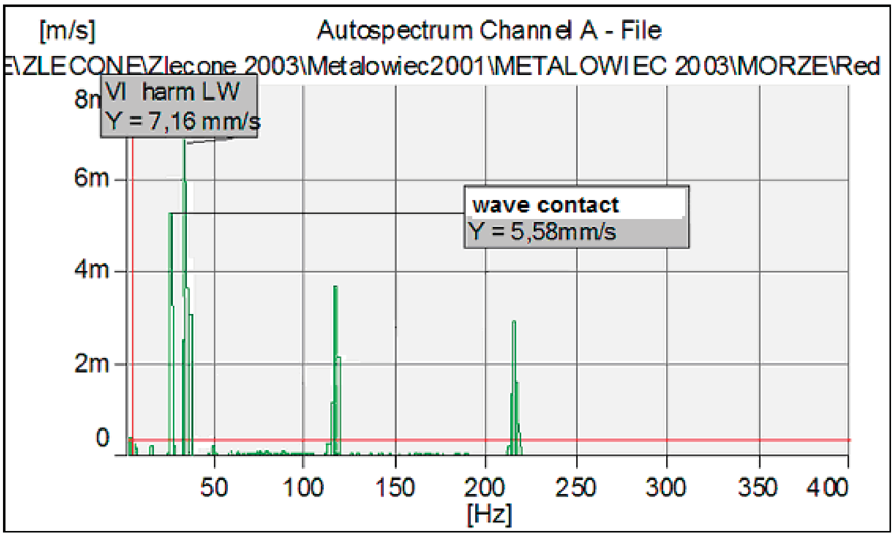

Fig. 9. Average spectrum of velocity, n = 300 rpm, e = 0.5 mm, wave amplitude 5 m (real object)

Final step of models identification is verification of its insensible for the environment disturbances. The Fig. 8, 9 present effects of disturbances coming from sea waves. The hull contacts reaction has their specific characteristic frequency. There is not influence of this signal for values of misalignment vibration symptom.

6. Final conclusions

It is common knowledge that failure frequency is the most hazardous factor in marine industry, just after aeronautics. Dynamic reactions which occur on ships hull in service at sea are rarely able to produce wear sufficient to cause a failure. The possible application of an on-line monitoring system of vibration parameters of the propulsion system of ships makes it possible to perform the typical technical diagnostic tests of torque transmission system and to identify shafts vibration as a result of misalignment. The modelling of periodic excitation form and next its identification in practice makes it possible:

- to identify values of vibrations coming from misalignment parameters during construction procedures,
- to identify cause of misalignment - periodic hull deformation, plastic deformation etc,
- to select dynamic characteristics of a measuring system which has to comply with requirements for typical technical diagnostics and for a hazard identification system,
- to identify elastic or plastic deformation of shaft line by using spectral assessment of its characteristic features in different ships displacements.

The presented results of modelling make it possible - due to strongly non-linear character of interactions occurring in sea environment - to assign unambiguously the modelled signal features to those of the recorded ones during the real test.

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