Journal of POLISH CIMAC

Faculty of Ocean Engineering & Ship Technology
GDAŃSK UNIVERSITY OF TECHNOLOGY

EFFECTS OF UNDERWATER EXPLOSION ON MINEHUNTERS SHAFTS LINES

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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. Modeling of dynamical reactions could bring information to the designer for recognizing the level of hazard for propulsion system. A paper presents a proposal of identification of a degree of hazard to ship shaft line due to forces of underwater explosion. 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 shafts misalignment. Both sub models allow testing forces and their responses in vibration spectrum using SIMULINK software.

***Keywords**: ship shaft lines, technical diagnostics, modelling, vibrations, underwater explosion*

1. Introduction

Minehunters propulsion systems are subjected to specific sea loads due to waving and dynamical impacts associated with underwater explosion. Sea waving can be sufficiently exactly modeled by means of statistical methods. Much more problems arise from modeling impacts due to underwater explosion. Knowledge of a character of impulse loading which affects ship shaft line can make it possible to identify potential failures by means of on-line vibration measuring systems.

2. Analysis of forces acting on shaft-line bearings

Ship shaft lines are subjected to loads in the form of forces and moments 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 dynamic loads. Much more reliable would be to relate results of the calculations to fatigue strength of material instead of its yield strength [5].

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 and those of external forces due to dynamic impacts.

To analyze the dynamic interaction a simplified model of shaft line is presented below, Fig. 1.

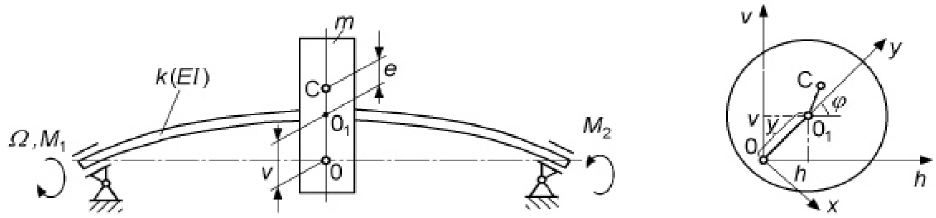

Fig. 1. A simplified shaft-line model for critical speed calculation [4]

Let us note: 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 (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 of the equations (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:

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

Presented 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. For minehunters propulsion system the torque pulsation expressed by means of Fourier series is much more complex. It additionally contains components resulting from number of propeller blades, kinematical features of reduction gear as well as disturbances from main engine and neighbouring devices. In general case occurrence of only one harmonic does not change reasoning logic.

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 (torsional, bending and compressive stresses);
- 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. Underwater explosion

Knowledge of loads determined during simulative explosions is helpful in dimensioning ship's hull scantlings [3]. Another issue is possible quantification of explosion energy as well as current potential hazard to whole ship and its moving system. From the point of view of shock wave impact on shaft line, underwater and over-water explosions should be considered in two situations:

- when shock wave (or its component) impacts screw propeller axially,
- when shock wave (or its component) impacts screw propeller perpendicularly to its rotation axis.

The axial shock-wave component affects thrust bearing and due to its stepwise character it may completely damage sliding thrust bearing. Rolling thrust bearings are more resistant to stepwise loading hence they are commonly used on naval ships [3]. The shock wave component perpendicular to shaft rotation axis is much more endangering. Shock wave can cause: damage of stern tube, brittle cracks in bearing covers and tracks, plastic displacement of shaft supporting elements including transmission gear and main engine, and even permanent deformation of propeller shaft. The problem of influence of sea mine explosion on hull structure is complex and belongs to more difficult issues of ship dynamics. Underwater explosion is meant as a violent upset of balance of a given system due to detonation of explosives in water environment. The process is accompanied with emission of large quantity of energy within a short time, fast running chemical and physical reactions, emission of heat and gas products. The influence of underwater explosion does not constitute a single impulse but a few (2 to 4) large energy pulsations of gas bubbles [2,8,9]. The pulsation process is repeated several times till the instant when the gas bubble surfaces. Hence the number of pulsations depends a.o. on immersion depth of the explosive charge. The character of changes of pressure values in a motionless point of the considered area is shown in Fig. 2. In the subject-matter literature can be found many formulae for determining maximum pressure value, based on results of experiments; however data on a character of pulsation and its impact on ship structures are lacking. To identify underwater explosion parameters a pilotage test was performed with the use of the explosive charge having the mass m = 37,5 kg. The schematic diagram of the experiment is shown in Fig. 3.

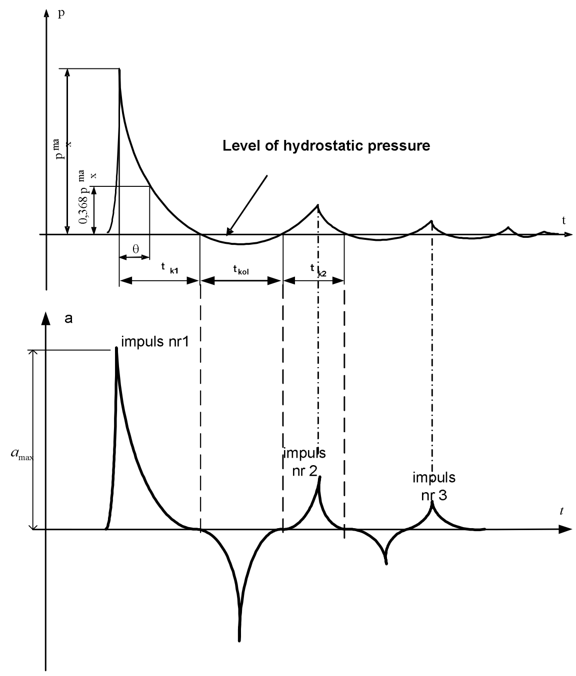

Fig. 2. Run of changes of shock wave pressure and ship hull acceleration measured on hull surface during underwater explosion.

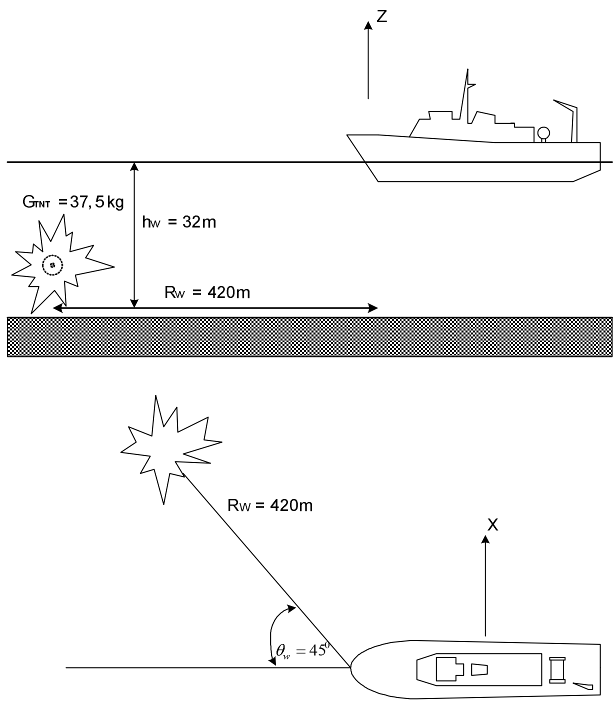

Fig. 3 Schematic diagram of the performed experimental test

During the sea-test were measured vibration accelerations of casings of intermediate and thrust bearings in the thrust direction and that perpendicular to shaft rotation axis. The ship course angle relative to the explosion epicentre was 45^0 and the shaft line rotated with the speed $n_{LW} = 500$ rpm. Ship's distance from the mine and its immersion depth was determined by using a hydro-location station and ROV underwater vehicle – figure 4. The vibration gauges were fixed over the reduction gear bearing as well as on the intermediate shaft bearing.

Fig. 4. ROV vehicle with TNT charge

The typical measurement record, in the vertical directions, is presented in Fig. 5. The time waveform of acceleration of the recorded signals were the same in all measurement points.

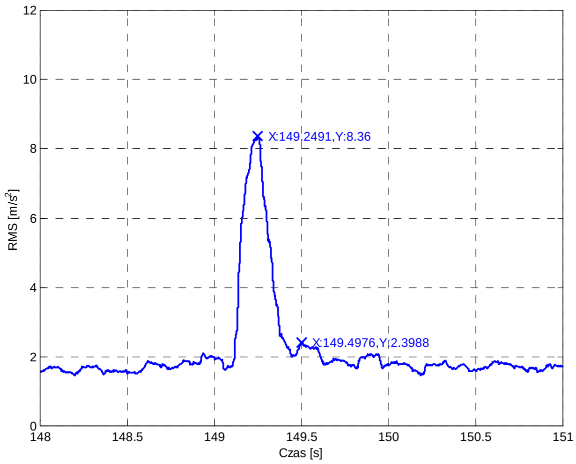

Fig. 5 Explosion, Port side (LB), Thrust bearing, V axis

The performed tests were aimed at achieving information dealing with:

- character of shock wave impact on shaft-line bearings , in the form of recorded vibration parameters;
- assessment of time-run of vibration accelerations with taking into account dynamic features of the signals in set measurement points;
- assessment of possible identification of influence of pulsation of successive gas bubbles during the time-run of vibration accelerations;
- identification of features of the signals by means of spectral analysis.

Since the mass of the explosive charge was small, to reliably identify the effect of only first and second pulsation was possible during the test.

4. Models of excitation due to underwater explosion

Analysis of dynamic impacts including impulse ones should take into account basic parameters which influence character of time-run of a given signal as well as its spectrum. The basic parameters which identify impulse impact resulting from explosion are the following:

- form of impulse which identifies kind of impulse;
- impulse duration time t_I at the ratio A/t_I maintained constant , which identifies explosive charge power (time of propagation of gas bubble);
- influence of damping on spectrum form , which identifies distance from explosion and - simultaneously - epicentre depth
- number of excitation impulses, which informs on distance from explosion, combined with explosive charge mass;
- time between successive impulses , which characterizes explosive charge mass;

The possible recording of measured shock wave pressure and accelerations on intermediate and propeller shaft bearings enables to identify some explosion parameters hence also hazards to power transmission system. Analysing the run of underwater shock wave pressure one is able to assume its time-dependent function (Fig. 6 and Eq. 4):

$$A = at^{kb} \cdot e^{kct} \tag{4}$$

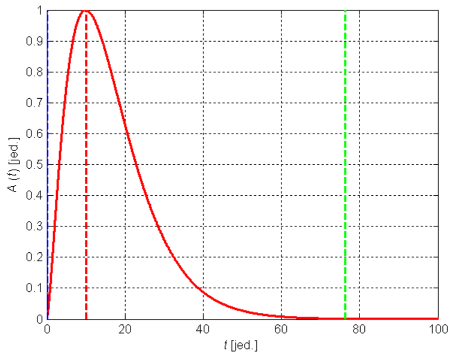

Fig. 6 Example of the function form for b=1,5 , c=-0,15 and k=1

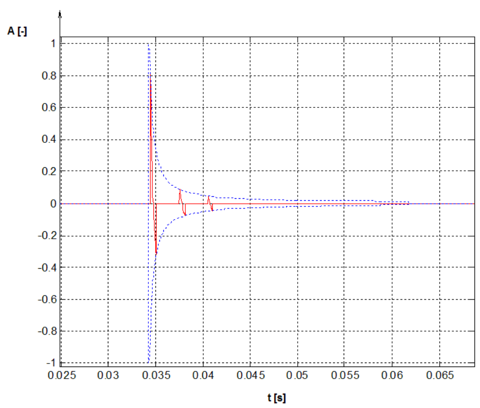

Fig. 7. Run of the assumed unit vibration acceleration model

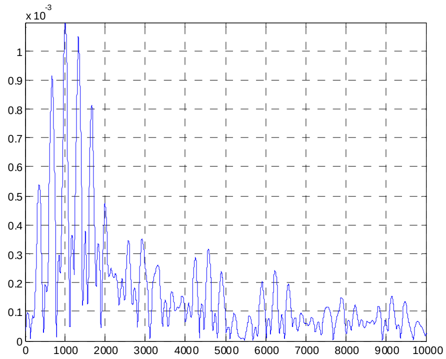

Fig. 8. Spectrum of the assumed vibration acceleration model

For the assumed mathematical model of the first shock wave impulse the run of vibration accelerations recorded on ship hull - for the example function given in Eq. 4 - can be presented as shown in Fig. 7 and 8.

5. Model of shaftline

The proposed model of minehunter shaft line is nonlinear and it is presented in Fig. 9.

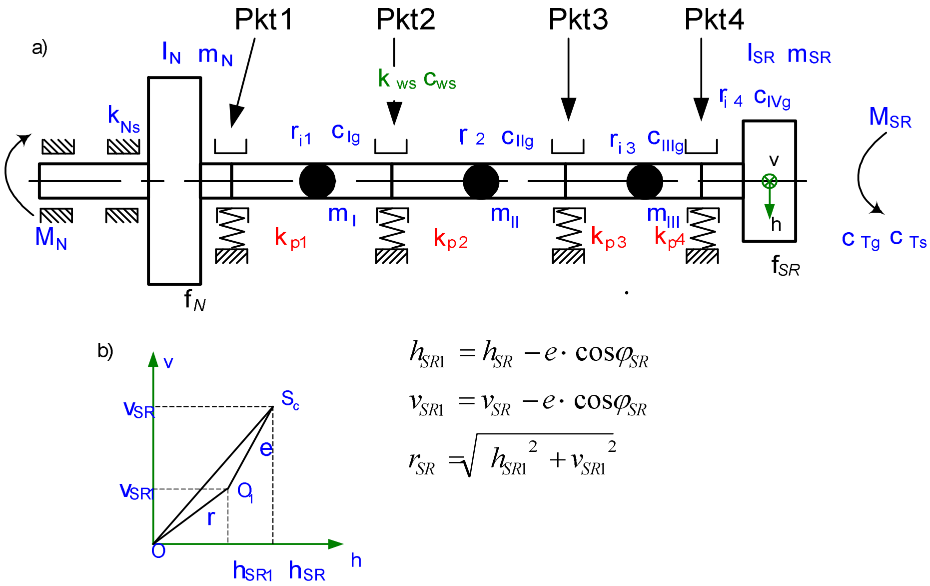

Fig. 9 Scheme of shaft line of minehunter, where:

I_N; m_N - reduced moment of inertia of the propulsion part, I_{SR}; m_{SR} - reduced moment of inertia of the propelled part, M_N; M_{SR} – torque and anti torque - propeller, m_I; m_{II}; m_{III} – reduced mass of shaft between bearings, k_{Ns} - torsional stiffness of propulsion system, r_{i1}; r_{i2}; r_{i3}; r_{i4};- coefficients of bending stiffness elements of shaft between bearings, c_{Ig}; c_{IIg}; c_{IIIg}; c_{IVg} - coefficients of bending damping elements of shaft between bearings, k_{ws}; c_{ws} - coefficients of torsional stiffness and damping of the shaft, φ_N; φ_{SR} – angles of rotation propulsion system and propeller, h_{SR}; v_{SR} – horizontal and vertical coordinates of centre of gravity S_c (after the shaft deflection), h_{SR1}; v_{SR1} - horizontal and vertical coordinates of centre of rotation O_1, h_N; h_I; h_{II}; v_N; v_I; v_{II} - horizontal and vertical coordinates for masses m_N; m_I; m_{II}, c_{Tg}; c_{Ts} - resistance of water bending and torsional damping

The model fulfill following requirements:

- it allows to insert the environmental loads,
- it allows to insert to the model the impulse of underwater explosion,
- the model reacts to changes of rotational speed of shaft line,
- the model reacts for axes misalignment,
- it keeps concordance in spectral and time domains with real object.

The kinetic energy of the model of shaft line can be represented by the following equation:

$$\begin{aligned}E_k &= \frac{1}{2} I_N \dot{\varphi}_N^2 + \frac{1}{2} I_{SR} \dot{\varphi}_{SR}^2 + \frac{1}{2} m_I (\dot{v}_I^2 + \dot{h}_I^2) + \frac{1}{2} m_{II} (\dot{v}_{II}^2 + \dot{h}_{II}^2) + \\ &+ \frac{1}{2} m_{SR} (\dot{v}_{SR}^2 + \dot{h}_{SR}^2) + \frac{1}{2} m_N (\dot{v}_N^2 + \dot{h}_N^2)\end{aligned} \tag{5}$$

The potential energy is written as follows:

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

and the dissipated energy in form:

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

It allows obtaining the second kind of Lagrange’s set of equations which can be represented as follow:

$$\begin{aligned}
& I_N \ddot{\varphi}_N + c_{ws} (\dot{\varphi}_N - \dot{\varphi}_{SR}) + k_{NS} \varphi_N + k_{ws} (\varphi_N - \varphi_{SR}) = M_N \\
& m_I \ddot{h}_I + c_{Ig} \dot{h}_I + r_{11} h_I + r_{12} h_{II} + r_{13} h_{III} + r_{14} h_{SR} = 0 \\
& m_I \ddot{v}_I + c_{Ig} \dot{v}_I + r_{11} v_I + r_{12} v_{II} + r_{13} v_{III} + r_{14} v_{SR} = 0 \\
& m_{II} \ddot{h}_{II} + c_{IIg} \dot{h}_{II} + r_{21} h_I + r_{22} h_{II} + r_{23} h_{III} + r_{24} h_{SR} = 0 \\
& m_{II} \ddot{v}_{II} + c_{IIg} \dot{v}_{II} + r_{21} v_I + r_{22} v_{II} + r_{23} v_{III} + r_{24} v_{SR} = 0 \\
& m_{III} \ddot{h}_{III} + c_{IIIg} \dot{h}_{III} + r_{31} h_I + r_{32} h_{II} + r_{33} h_{III} + r_{34} h_{SR} = 0 \\
& m_{III} \ddot{v}_{III} + c_{IIIg} \dot{v}_{III} + r_{31} v_I + r_{32} v_{II} + r_{33} v_{III} + r_{34} v_{SR} = 0 \\
& I_{SR} \ddot{\varphi}_{SR} + c_{ws} (\dot{\varphi}_{SR} - \dot{\varphi}_N) + c_{Ts} \dot{\varphi}_{SR} + c_{IIIg} (\dot{h}_{SR} e \sin \varphi_{SR} - \\
& \dot{v}_{SR} e \cos \varphi_{SR} + e^2 \dot{\varphi}_{SR}) + k_{ws} (\varphi_{SR} - \varphi_N) + \\
& r_{44} (h_{SR} e \sin \varphi_{SR} - v_{SR} e \cos \varphi_{SR}) = M_{SR} \\
& m_{SR} \ddot{h}_{SR} + c_{IVg} (\dot{h}_{SR} + \dot{\varphi}_{SR} e \sin \varphi_{SR}) + r_{41} h_I + r_{42} h_{II} + r_{43} h_{III} + \\
& r_{44} (h_{SR} - e \cos \varphi_{SR}) = 0 \\
& m_{SR} \ddot{v}_{SR} + c_{IVg} (\dot{v}_{SR} - \dot{\varphi}_{SR} e \cos \varphi_{SR}) + r_{41} v_I + r_{42} v_{II} + \\
& r_{43} v_{III} + r_{44} (v_{SR} - e \sin \varphi_{SR}) = 0
\end{aligned} \tag{8}$$

The model of shaft line is implemented to the MATLAB SIMULINK software with accordance of mentioned requirements – Fig. 10.

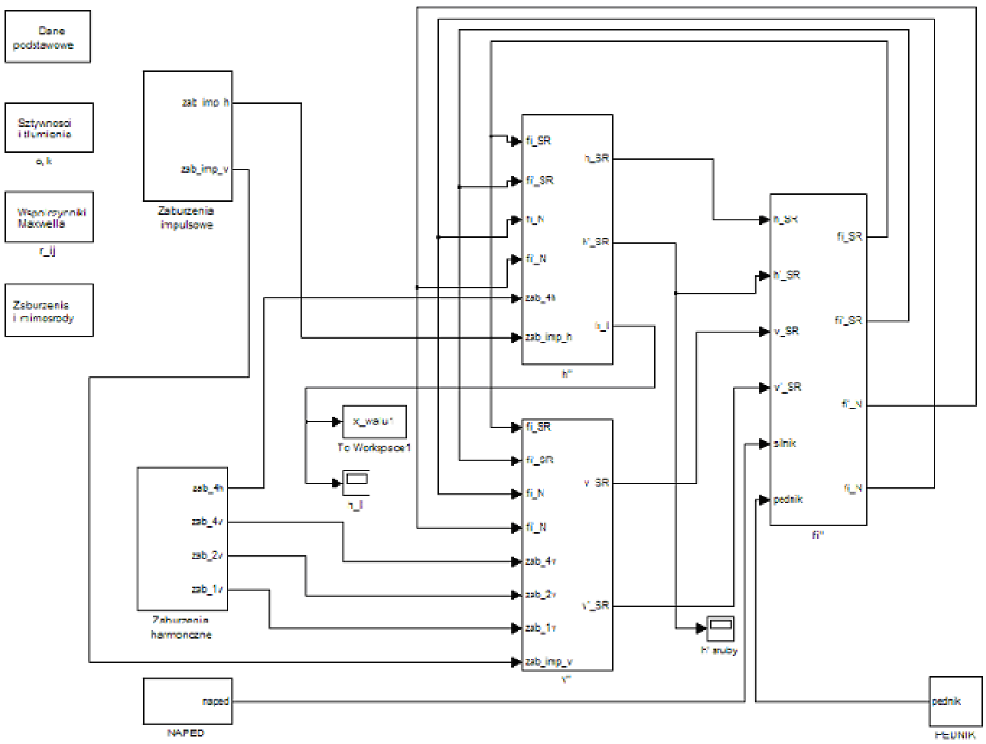

Fig. 10 Scheme of structure shaft line model in the MATLAB SIMULINK

Typical dangers for minhunter propulsion system are stresses coming from underwater explosion. Presented model allows calculating reactions in time and frequency domain. The example of spectrum of vibration acceleration in the point 3 of model (Fig. 9) during the simulated explosion is presented in the Fig. 11.

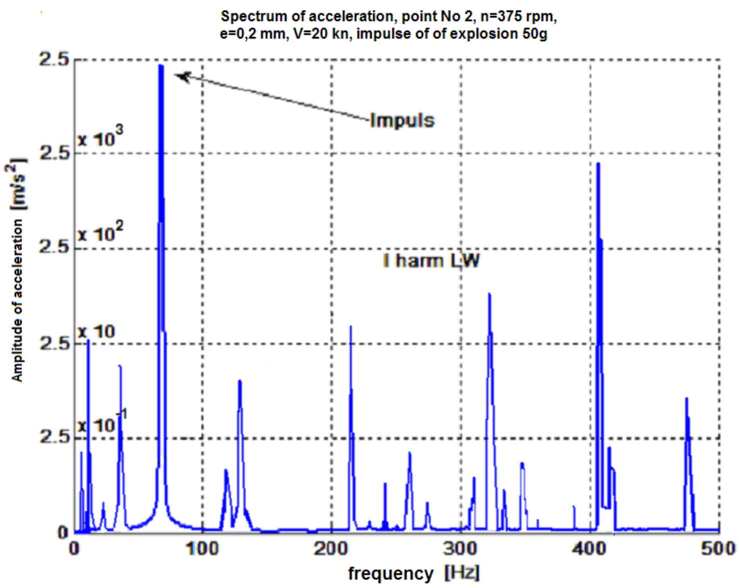

Fig. 11. The spectrum of acceleration with the inserted simulation of underwater explosion

6. Final conclusions

It’s common knowledge that failure frequency is the most hazardous factor in marine industry, just after aeronautics. Dynamic reactions which occur on ships 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 minehunter makes it possible to perform the typical technical diagnostic tests of torque transmission system and to identify possible plastic deformations of hull plating as a result of underwater explosion.

The modelling of impulse impact form and next its identification makes it possible additionally: to identify explosion power by using an analysis of the first vibration impulse amplitude and its duration time, to identify distance from explosion epicentre (hence a degree of hazard) by analysing signal’s damping, to identify a kind of explosion and even characteristic features of type of used mine, 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 from before and after underwater explosion.

The presented results of modelling related to the performed experimental test do not 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.

Successive experimental tests will make it possible to verify features of the signals assumed for the analysis, to be able to build reliable models.

The wide range of stochastic dynamic loads acting on ships during its life-time makes that in the nearest future the application of on-line diagnostic techniques to ship propulsion systems, based on analysing vibration signals, will constitute an obvious tactical and technical necessity.

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