

## DYNAMIC PROBLEMS OF SHAFTLINES

Andrzej GRZĄDZIELA

Naval University of Gdynia, Mechanical – Electrical Faculty  
ul. Śmidowicza 69, 81-103 Gdynia, Poland, e-mail: [agrza@amw.gdynia.pl](mailto:agrza@amw.gdynia.pl)

### Summary

Ships' propulsion plant usually works in a hard environment caused by static forces and permanent dynamic loads. Elastic strains from machine vibration can cause resonance of plastic strain of shell plating and foundation of shafting elements. Exceeding of tolerated values of shaft alignments causes a damage of radial and thrust bearings in relative short time. The alignment deviations in the construction of ships propulsion shaft line has been an effect of tensile forces, compressive forces, bending moments and transverse vibration from disturbances of rotation movement. Modeling of dynamical reactions could bring information to the project data base for recognizing the level of hazard for propulsion system of the naval vessels. Recorded signals were recognized within sensitive symptoms of two models: model of propulsion system and model of underwater explosion.

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

### PROBLEMY DYNAMIKI LINII WAŁÓW OKRĘTOWYCH

#### Streszczenie

W artykule przedstawiono propozycję identyfikacji stopnia zagrożenia od obciążenia impulsowego wywołanego przez wybuch podwodny na okrętową linię wałów. Dokonano analizy teoretycznej wpływu zmian współosiowości wałów na prędkości krytyczne drugiego rodzaju wynikające z odkształceń sprężystych kadłuba w rejonie fundamentów łożysk nośnych. Przedstawiono wyniki pilotażowych badań na poligonie morskim z wykorzystaniem eksplozji podwodnej. Zaproponowano wstępny model matematyczny opisu eksplozji podwodnej uwzględniający masę ładunku oraz odległość od obiektu. We wnioskach przedstawiono sposób identyfikacji zagrożenia dla linii wałów przy wykorzystaniu analizy widmowej przebiegów czasowych sygnałów drganiowych rejestrowanych na łożyskach nośnych.

Słowa kluczowe: linia wałów okrętowych, diagnostyka techniczna, modelowanie, drgania, wybuch podwodny.

## 1. INTRODUCTION

Ship propulsion systems are subjected to specific sea loads due to waving and dynamical impacts associated with mission of a given ship. Sea waving can be sufficiently exactly modeled by means of statistical methods. Much more problems arise from modeling impacts due to underwater explosion. 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 during every 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 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 – Fig. 1.

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. 2.

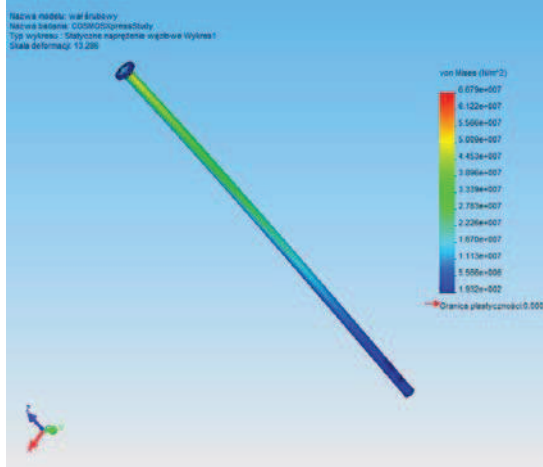


Fig. 1. Simulated static bending stresses in propeller shaft due to weight of propeller

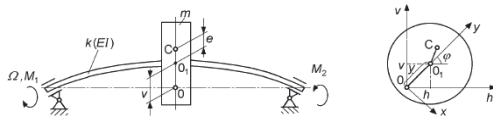


Fig. 2. A simplified shaft-line model for critical speed calculation [4]

Let us note :  $M_1$  - torque,  $M_2$  - anti-torque. The system can be represented by the following set of equations:

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

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  $\varphi$  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:

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

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

$$M = M_1 - M_2 = ke(h \sin \varphi - v \cos \varphi) \quad (3)$$

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.

For ship 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.

For long shaft lines of ships the influence of gravity forces on critical speeds should be taken into consideration [7]. According to Eq. (4) the generated vibrations will be then performed respective to static deflection axis of the shaft.

$$\begin{aligned} m\ddot{h} + kh &= me\Omega^2 \cos \varphi \\ m\ddot{v} + kv &= me\Omega^2 \sin \varphi \\ \ddot{\varphi} &= 0 \end{aligned} \quad (4)$$

Hence the equations obtain the following form:

$$\begin{aligned} \ddot{h} + \omega_0^2 h &= e \cdot \Omega^2 \cos \varphi \\ \ddot{v} + \omega_0^2 v &= e \cdot \Omega^2 \sin \varphi - mg, \\ J\ddot{\varphi} &= -mge \cos \varphi. \end{aligned} \quad (5)$$

Since in the third equation of the set (1), i.e. that for  $\Omega_{3KR}$ , appears the exciting torque of the frequency/angular speed ratio  $\beta=1$  it means that one has to do with the critical state of 2<sup>nd</sup> kind for  $\beta=1$ , namely:

$$\Omega_{KR(2)} = \frac{1}{2} \omega_0 \quad (6)$$

Occurrence of such kind vibrations is conditioned by non-zero value of  $e$ , which – in the case of ship shaft line – appear 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 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 (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. PROBLEM OF UNDERWATER EXPLOSION

Information on potential hazard resulting from underwater explosion is crucial not only for ship's commander during warfare but also for ship structure designers. 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. 3.

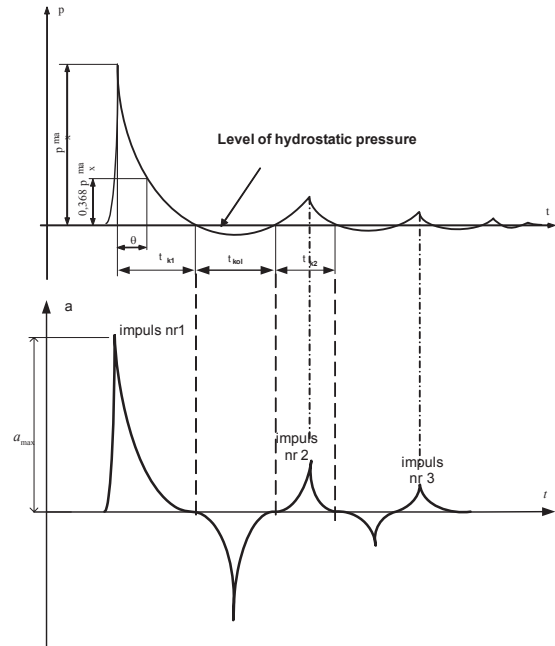


Fig. 3. Run of changes of shock wave pressure and ship hull acceleration measured on hull surface during underwater explosion

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. 4.

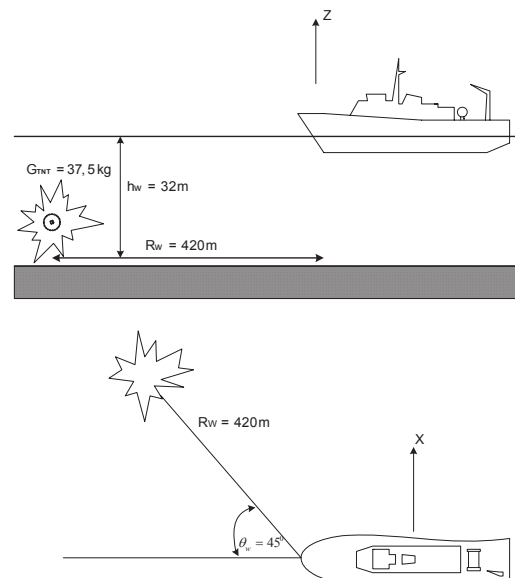


Fig. 4. Schematic diagram of the performed experimental test

During the 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  $45^\circ$  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 5. The vibration gauges were fixed over the reduction gear bearing as well as on the intermediate shaft bearing.



Fig. 5. ROV vehicle with TNT charge

The measurement directions (X and Z axes) are presented in Fig. 6 as well.

The time lag of the recorded signals was the same in all measurement points, as shown in Fig. 6 and 7.

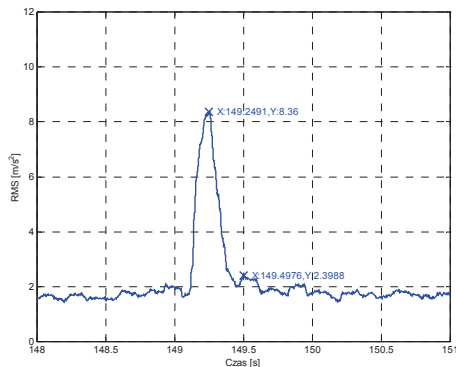


Fig. 6. Explosion, Port side (LB), Thrust bearing, X axis

The performed test was 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.

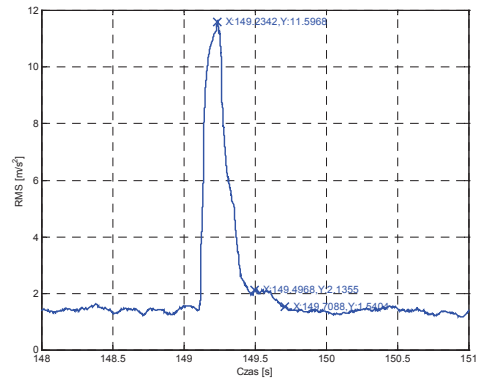


Fig. 7. Explosion, Starboard (PB), Intermediate bearing, Z axis

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. 8 and 9).

$$A = at^{kb} \cdot e^{kct} \quad (7)$$

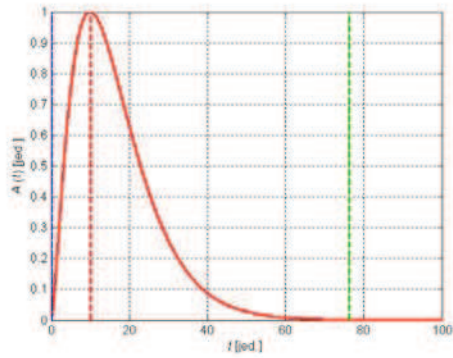


Fig. 8. Example of the function form for  $b=1,5$ ;  $c=-0,15$  and  $k=1$

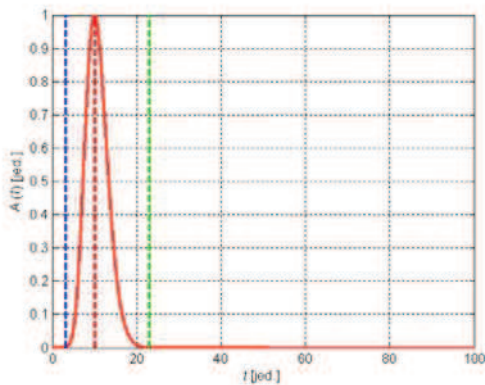


Fig. 9. Example of the function form for  $b=1,5$ ;  $c=-0,15$  and  $k=10$

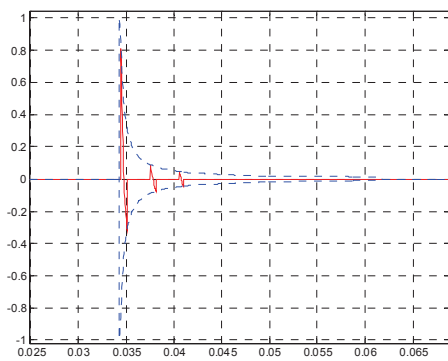


Fig. 10. Run of the assumed vibration acceleration model

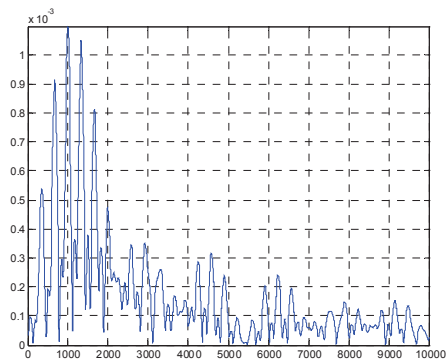


Fig. 11. Spectrum of the assumed vibration acceleration model

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 Fig. 8 - can be presented as shown in Fig. 10 and 11.

## 5. 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 mine hunter 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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Dr inż. **Andrzej  
GRZĄDZIELA** jest  
Kierownikiem Zakładu  
Napędów Okrętowych  
w Instytucie Konstrukcji  
i Eksploatacji Okrętów  
Wydziału Mechaniczno –  
Elektrycznego Akademii

Marynarki Wojennej w Gdyni. W swojej działalności zawodowej zajmuje się problemami oceny niewyważenia i oceny współosiowości w okrętowych układach napędowych a także projektowaniem okrętów i doborem układów napędowych. Członek Polskiego Towarzystwa Diagnostyki Technicznej oraz Polskiego Towarzystwa Naukowego Silników Spalinowych