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VIBRATIONS IN MARINE POWER TRANSMISSION SYSTEM

DRGANIA OKRĘTOWYCH UKŁADÓW PRZENIESIENIA NAPĘDU

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Abstract: Vibration analyses of marine machines and structures are one of the most important during the design process as well as during exploitation. Vibrations of ship hull (including superstructure and main engine body) are separately analysed from the vibrations of power transmission system. Vibrations of propulsion system include three types: lateral vibration, coupled axial vibration and torsional vibration. Among them, torsional vibrations are usually the most dangerous for the shaft line and the crankshaft. These vibrations may cause the increasing failure of the engine crankshaft as broken and bent shaft. Therefore, this article focuses on the study of torsional vibration of ship propulsion system. Calculation of torsional vibration of propulsion system with a medium-speed main engine is presented. The analysis is based on finite element method, with the code written in Matlab software. The result of this paper is applied for the tugboat with the engine of power 350 HP.

Keywords: marine propulsion ship vibrations, torsional vibration, marine propulsion system, finite element method.

Streszczenie: Analizy drgań okrętowych maszyn i konstrukcji są jednymi z najważniejszych podczas procesu projektowania oraz ich eksploatacji. Drgania kadłuba statku (z nadbudówką i korpusem silnika głównego włącznie) są analizowane oddzielnie od drgań układu przeniesienia napędu. Wyróżnia się trzy typy drgań układu napędowego: drgania giętne, sprzężone wzdłużne oraz skrętne. Wśród nich drgania skrętne są zwykle najgroźniejsze dla linii wałów wału korbowego. Mogą one zwiększyć prawdopodobieństwo uszkodzenia wału korbowego poprzez jego złamanie lub wygięcie. Z tego powodu w artykule skupiono się na analizie drgań skrętnych okrętowych układów napędowych. Zaprezentowano obliczenia drgań skrętnych układu napędowego wyposażonego w średnioobrotowy silnik główny. Analizę przeprowadzono metodą elementów skończonych, której procedura została napisana w programie Matlab. Zastosowano ją do obliczeń holownika wyposażonego w silnik o mocy 350 HP.

Słowa kluczowe: drgania okrętowych układów napędowych, drgania skrętne, okrętowe układy napędowe, metoda elementów skończonych.

1. INTRODUCTION

Economy of shipping forced a trend to increasing the ships capacity and speed. Marine internal combustion engine must be more efficient and strenuous. It is lead to many failures in the engine. Especially, the phenomenon of broken, cracked crankshaft is very dangerous. The cause of that phenomenon is not only due to overload, but also due to the cyclical mode of vibration occurring in the crankshaft - connecting rod structure [Thomson 1982; Geradin and Rixen 1994; Rao 1995].

The propulsion system of a ship is used to propel the ship and control its maneuvering. The system is consisting of main engine, gearbox, propulsion shaft line, propeller and pertinent auxiliary systems. The power transmission system is the essential part of a ship propulsion system. The excitation forces originated from the shaft line can greatly affect the dynamic response of the whole ship structure [Murawski 2003]. A reliable FE model of the shaft line is an assistance in dynamic response analysis of the ship vibration. There have been many studies on the crankshaft torsional vibration. Based on these studies, engineers and operators can offer technical solutions to improve the reliability and safety of internal combustion engines while working under different exploitations' conditions [Iijima, Yao and Moon 2008]. For example, the application of twisted rubber dampers and hydraulic simple for the internal combustion engine. There are many methods of calculating torsional vibration of ship propulsion shaft lines [Lin et al. 2009], in which torsion vibration calculation method based on finite element method [Zienkiewicz and Taylor 2005] is the simplest and most effective.

In the presented paper, vibration analysis of a ship propulsion power transmission system is considered. By applying the finite element method, model of the shaft line and the crankshaft is simplified as a one degree of freedom (torsion vibration). Natural vibrations frequencies, vibration modes, response of the forced vibrations and amplifier functions are identified. The analysis can make accurate conclusions about the state of a ship's power transmission system, and the resonance positions (barred speed range) may occur.

2. OVERVIEW OF SHIP'S POWER TRANSMISSION SYSTEM

Propulsion system tasked ships transmit torque from the engine to the propeller and get thrust from the propeller passed on to the hull makes the ship forward or backward. A shaft propulsion system consists of multiple segments axis interconnecting together and placed on a straight line. Depending on use, and features of each type of ship, the ship may have one or more axis.

The propulsion system is working in very complex conditions. One end of the power transmission system is connected to the main engine (directly affected by torque from the main engine), the other end is brought to the propeller (directly affected by the propeller torque resistance). Also, remaining axes affected by the thrust of the propeller, under the effect of weight shaft itself. Therefore, the determination of the working mode of the axis is of major importance and necessity. The propulsion system of a ship structure which consists of several elements include: main engine, couplings, gearbox, bearings, shaft line, brackets and propeller. A typical marine propulsion system is presented in Fig. 1.

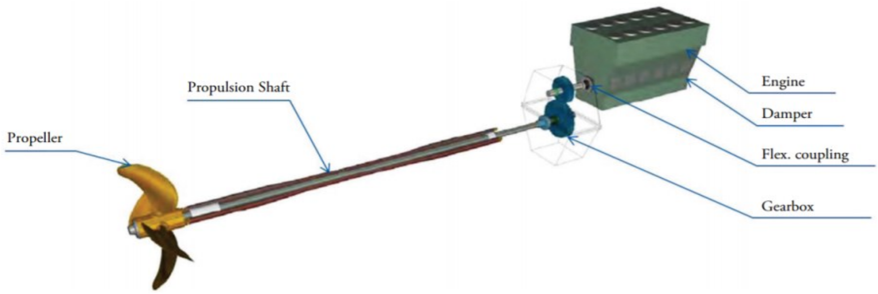


Fig. 1. *Typical marine propulsion system*

Rys. 1. Typowy okrętowy układ napędowy

3. VIBRATIONS OF THE POWER TRANSMISSION SYSTEM

Three main types of vibration of the ship shafting can be featured: lateral vibration, axial vibration and torsional vibration. We in turn consider each case vibration and its impact on ship shafting system [Murawski 2004, 2005; Murawski and Charchalis 2014].

Lateral (transverse, bending, whirling) vibration of a typical marine power transmission system is not usually dangerous for a slow-speed propulsion systems. Dynamic, bending stresses, shear forces and lateral vibration amplitudes of the shaft line are negligible in comparison to a shaft line torsional vibration. Usually, the lateral vibration amplitudes are very small and do not exceed 0.3 mm on the propeller (where the amplitudes are maximum). Nevertheless, the shaft line lateral vibration analysis are quite often performed, especially for a long, elastic shaft line or when a ship hull is elastic compared to the propulsion system.

Axial (longitudinal) vibrations are a result of the pulsing hydrodynamic forces inducted on the propeller and a dynamic longitudinal deformations of the crankshaft. When crankshaft throw is loaded by gas pressure and mass forces through a connecting rod mechanism, arms of the crank throw deflect in the axial direction of the crankshaft, exciting axial vibrations. These vibrations may cause the increasing failure frequency of the engine crankshaft. The propulsion system is connected to the ship hull through a thrust bearing. Therefore, the axial vibration is transferred to different regions of the ship hull structure through a thrust bearing (and possibly axial detuner or damper) and the sip double bottom. Excessive superstructure vibrations worsen the ship's crew working conditions and may detrimentally influence maritime safety. Generally, axial vibrations are only dangerous for propulsion system with slow-speed, two-stroke engines and directly driven propellers by shaft lines. Nowadays, all slow-speed engines are equipped with an axial damper (detuner). If the damper is well regulated axial vibration are not dangerous for the main engine.

Torsional vibrations are the result of the pulsing torque of the reciprocating combustion engine as well as reciprocating propeller's power output, and the torsional elasticity of the power transmission system. All system components like the crankshaft, intermediate shaft, propeller shaft and optional couplings and gears have to transmit the static and additionally dynamic torque. Torsional vibrations of the marine power transmission system are usually the most dangerous for the shaft line and the crankshaft. Vibratory stresses, torques and/or angular amplitudes have to be analysed by calculations and very often by measurements for all marine propulsion systems. In the exploitation process, fractures of ship's shaft due to torsional vibrations are very dangerous for propulsion system. Torsional vibrations can lead to the shafts to suffer – a huge external cycle force that lead to fatigue of the material and damage the shaft. Given the importance of torsional vibrations, this paper will focus primarily on analyzing and assessing their impact on the propulsion system for ships.

4. METHOD OF TORSIONAL VIBRATION CALCULATIONS

The most popular is calculation of the torsional vibration by finite element method [Reddy 1993, Zienkiewicz and Taylor 2005]. The authors program codes were written in Matlab. It can distinguish the following steps during the torsional vibrations analysis:

- Modeling ship's shafts.
- Discretization of the structure.
- Determination the stiffness matrix, the mass (inertia) matrix, torque of the elements, and assembly matrix for structural (boundary conditions).
- Solve equations of torsional vibration and the matrix of inertia torque of the whole structure.

In the presented procedure, the following contents are implemented:

- Solve problems and finding eigenvalues and modes (natural vibration) for separate vectors.
- Finding response system (forced vibration); the force (torque of the crankshaft) is put into as discrete value; the Newmark method was implemented; the damping matrix is determined as linear combination of stiffness and mass matrix ($C = aM + bK$, where: $a = 10^{-7}$, $b = 10^{-5}$).
- Finding amplified functional form.

During the torsional vibration analyses, real axes have to be replaced by simple system consisting of an elastic cylinder shaft and round plates attached to the shaft. This alternative system must meet the following conditions:

- The torsion angle of the real axis must coincide with the torsion angle of the equivalent axis, for any frequency value; the modelling method of real axes to the equivalent axis is presented in Fig. 2.
- Moment of inertia of equivalent mass must be equal to the moment of inertia of real mass.

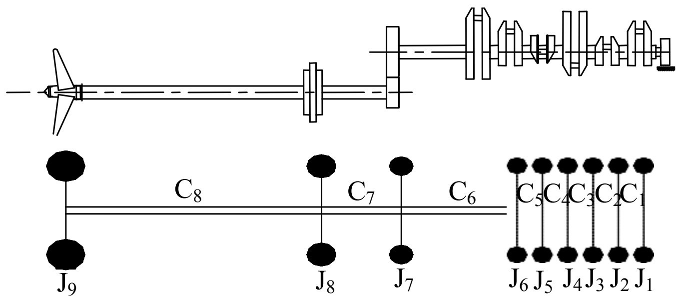


***Fig. 2.** Real axes and equivalent axis in torsion vibration*

Rys. 2. Rzeczywiste i ekwiwalentne osi w obliczeniach drgań skrętnych

Usually, the reciprocating and rotating masses of the engine including crankshaft, intermediate shaft(s), propeller shaft and propeller are modeled as a system of rotating masses (inertias) connected by the torsional spring (see Fig. 2). The power transmission system model with one degree of freedom in each node is usually sufficient [Nestorides 1958; Geveci, Osburn and Franchek 2005]. There is no problem with any boundary conditions. Therefore, a more detailed model of the power transmission system is not required for typical analysis. The detailed 3-D FEM model of the crankshaft is used by one of the authors, only for determining the coupling dependencies between torsional and longitudinal vibration [Murawski 2004] or some special case of shaft line bending vibration [Murawski 2005]. In general, the multi-node, unbounded torsional vibration modes are interesting.

The first question is where the main natural frequency of a system should be situated. This can be achieved by changing the masses and/or the stiffness of the system so as to give a much higher, or much lower natural frequency, called undercritical or overcritical running, respectively. In the undercritical case one-node resonance vibration with the main critical order should occur about 35÷45% above the nominal engine speed. Such undercritical conditions can be realised by choosing a rigid shaft system, leading to a relatively high natural frequency. The characteristics of an undercritical propulsion system are normally: a relatively short shafting system, probably with no tuning wheel, a turning wheel with relatively low inertia and large diameters of shafting. The main advantage of undercritical propulsion is that the system does not have a barred speed range. But, the highest torsional stress level in the nominal main engine speed is a disadvantage. When running undercritical, significant varying torque at nominal conditions of about 100÷150% of the mean torque is expected. This torque (propeller torsional amplitude) induces a significant varying propeller thrust. Changed propeller thrust might be a source of high level of longitudinal vibrations on the power transmission system and then double bottom and ship hull and deckhouse. For those reasons the undercritical propulsion system is quite rarely applied. In the overcritical case one-node natural vibration frequency is placed about 30÷70% below the nominal engine speed. Such overcritical conditions can be realised by choosing an elastic shaft system, leading to a relatively low natural frequency. The characteristics of an undercritical propulsion system are a tuning wheel necessary on the crankshaft fore end, a turning wheel with relatively high inertia and shafts with relatively small diameters (requiring shafting material with a relatively high ultimate tensile strength). A barred speed range is expected in this propulsion system. Excessive torsional vibrations in overcritical conditions may have to be eliminated by the use of a torsional vibration damper.

General equation of torsional vibration can be presented in the following form:

$$\boldsymbol{I\ddot{\varphi} + C\dot{\varphi} + K\varphi = M_E(t)} \tag{1}$$

where:

- φ – the rotational angle,
- $\boldsymbol{I}$ – the matrix of masses moments of inertia,
- $\boldsymbol{C}$ – the matrix of torsional dampings,
- $\boldsymbol{K}$ – the matrix of torsional stiffnesses,
- $\boldsymbol{M_E}$ – the excitation moment.

Dampings have an insignificant influence on natural vibration frequency and mode. Natural vibrations are defined as motion without excitations. Therefore, equation 1 may be simplified to:

$$\boldsymbol{I}\ddot{\varphi} + \boldsymbol{K}\varphi = \boldsymbol{0} \tag{2}$$

Moments of inertas and stiffnesses of typical elements (shafts) can be determined on the base of well-known simple formulas given by general mechanics. Determination of added water mass is important and difficult problem [Senjanovic et al. 2014]. There are several formulas describing propeller inertia of added water mass [Nestorides 1958]. The best one, in the authors' opinion, has been derived on the basis of Parson’s theory (the equation No. 3 and Tab. 1).

$$J_H = D^5\rho\left[CJ_1 + CJ_2\frac{A_e}{A_0} + CJ_3\frac{P}{D} + CJ_4\left(\frac{A_e}{A_0}\right)^2 + CJ_5\left(\frac{P}{D}\right)^2 + CJ_6\frac{A_e}{A_0}\frac{P}{D}\right] \tag{3}$$

where:

J_H – inertia of entrained water [kgm^2],
D – propeller diameter [m],
ρ – specific mass of sea water (usually 1025 kg/m^3),
CJ_i – coefficients given in Table 1,
A_e/A_0 – expanded area blade ratio,
P/D – propeller pitch ratio.

Table 1. *Coefficients for propeller inertia of entrained water*

Tabela 1. Współczynniki do wyznaczania masowych momentów bezwładności wody towarzyszącej śruby napędowej

No. of blades	CJ_1	CJ_2	CJ_3	CJ_4	CJ_5	CJ_6
4	3.0315E-3	-8.0782E-3	-4.0731E-3	3.4170E-3	4.3437E-4	9.9715E-3
5	2.7835E-3	-7.1650E-3	-3.7301E-3	3.0526E-3	4.6275E-4	8.5327E-3
6	2.3732E-3	-6.2877E-3	-3.0606E-3	2.7478E-3	2.9060E-4	7.3650E-3

Damping characteristics are also determined. What is more, there are several different methods of characterising damping phenomena. On the other hand damping has no real influence on natural frequencies. Forced vibration (especially in resonance range) is strongly depended on damping. Damping can be described by a vibration magnifier on the basis of measurements. These magnifiers may be used on a similar mechanism. In the ship construction a typical vibration magnifier is between 20-25, but for torsional vibrations the dampings have higher values because of the water damping. A typical vibration magnifier for torsional vibrations is between 12-15. Some manufacturers (mostly engine factories) give us damping factors corresponding to their products.

Global stiffness matrix of the structure can be presented in the following form:

$$K = \begin{bmatrix} k_1 & -k_1 & 0 & 0 \\ -k_1 & k_1 + k_2 & -k_2 & \vdots \\ 0 & -k_2 & \ddots & -k_n \\ 0 & \cdots & -k_n & k_n \end{bmatrix} \tag{4}$$

Global moment of inertia matrix of the structure can be presented by the following equation:

$$M = \begin{bmatrix} J_1 & 0 & \cdots & 0 \\ 0 & J_2 & \ddots & 0 \\ 0 & 0 & \ddots & \vdots \\ 0 & 0 & \cdots & J_n \end{bmatrix} \tag{5}$$

Cylindrical gas forces and the assembly of the crankshaft, the piston and the connecting-rod mass forces are only one significant excitation of the torsional vibration. Therefore excitation forces have to be determined by the producers for each separate engine type. These forces are periodic but not harmonic so based on experimental data, after decomposition on the harmonic components, described by the Fourier series. Sixteen up to twenty-four first components of the gas forces are given for two-stroke engines. In the case of four-stroke engines there are half-harmonic components (one working period is equal to two engine revolutions). Decomposition of the mass forces to five harmonics is sufficient.

The gas and mass forces acting on the crank pin are decomposed to radial and tangential components. The radial forces are the source of crank bending and coupled longitudinal vibrations. The tangential forces cause dynamic torsion moments and are the source of the torsional vibrations. The decomposition of gas forces is presented in Fig. 3; the layout of mass forces is presented in Fig. 4. An example of radial gas forces with decomposition on its harmonic components are presented in Fig. 5.

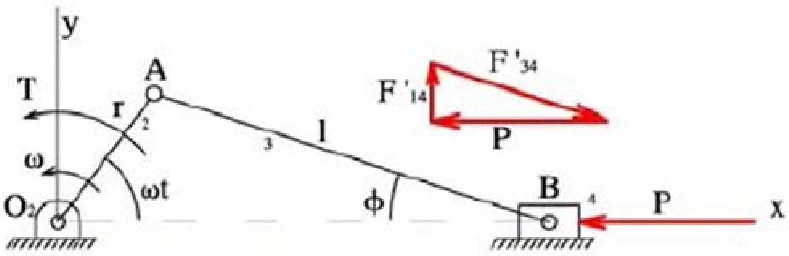


Fig. 3. *The layout of gas forces*
Rys. 3. Rozkład sił gazowych

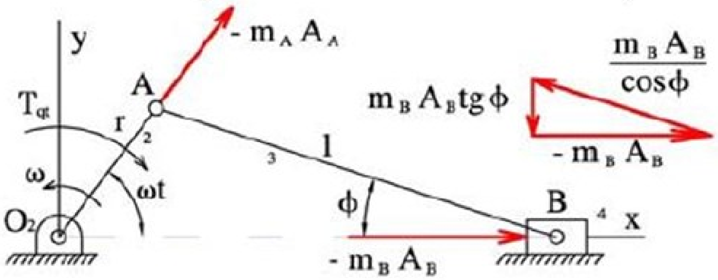
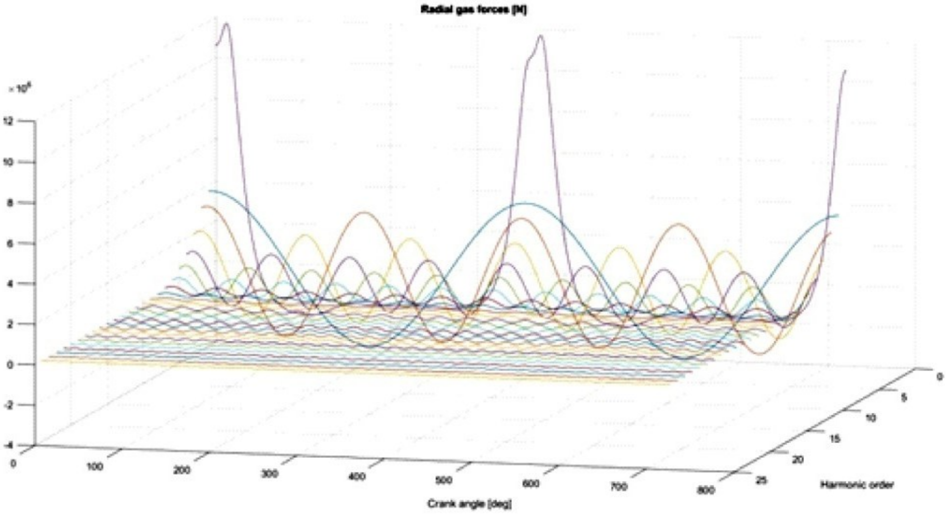


Fig. 4. *The layout of mass forces*
Rys. 4. Rozkład sił masowych



Rys. 5. Przykład składowych harmonicznych gazowych sił promieniowych

Fig. 5. *Example of harmonic components of radial gas forces*

5. EXAMPLE OF TORSIONAL VIBRATION CALCULATIONS

Analysis of the power transmission system torsional vibrations was performed by the authors' specialised FEM software. Propulsion system of the tugboat (32.5 m length, 280 DWT) was analysed. The propulsion system is based on a high-speed, four-stroke, six-cylinder main engine Volvo TAMD122A type. The engine main parameters are as follows: power – 380 HP and nominal speed – 2100 rpm. The propulsion system was equipped in five-blade propeller: diameter 1.2 m and mass in air – 280 kg.

The FEM model of the power transmission system is presented in Fig. 2. Based on the finite element method we obtain: moment inertia of equivalent mass $J_1 = J_2 = J_3 = J_4 = J_5 = J_6 = 0.32$ [kg×m^2], $J_7 = 11.79$ [kg×m^2], $J_8 = 205$ [kg×m^2], $J_9 = 276$ [kg×m^2], equivalent stiffness $C_1 = C_2 = C_3 = C_4 = C_5 = 6.08$ [MNm/rad], $C_6 = 24.4$ [MNm/rad], $C_7 = 17.6$ [MNm/rad], $C_8 = 0.42$ [MNm/rad], equivalent length $l_1 = l_2 = l_3 = l_4 = l_5 = 0.15$ [m], $l_6 = 0.126$ [m], $l_7 = 0.175$ [m], $l_8 = 12.5$ [m]. Nine modes and response of the natural vibrations were determined.

The modes are presented in Fig. 6 and 7.

The responses of force vibration and amplifier function are presented in Fig. 8 and 9.

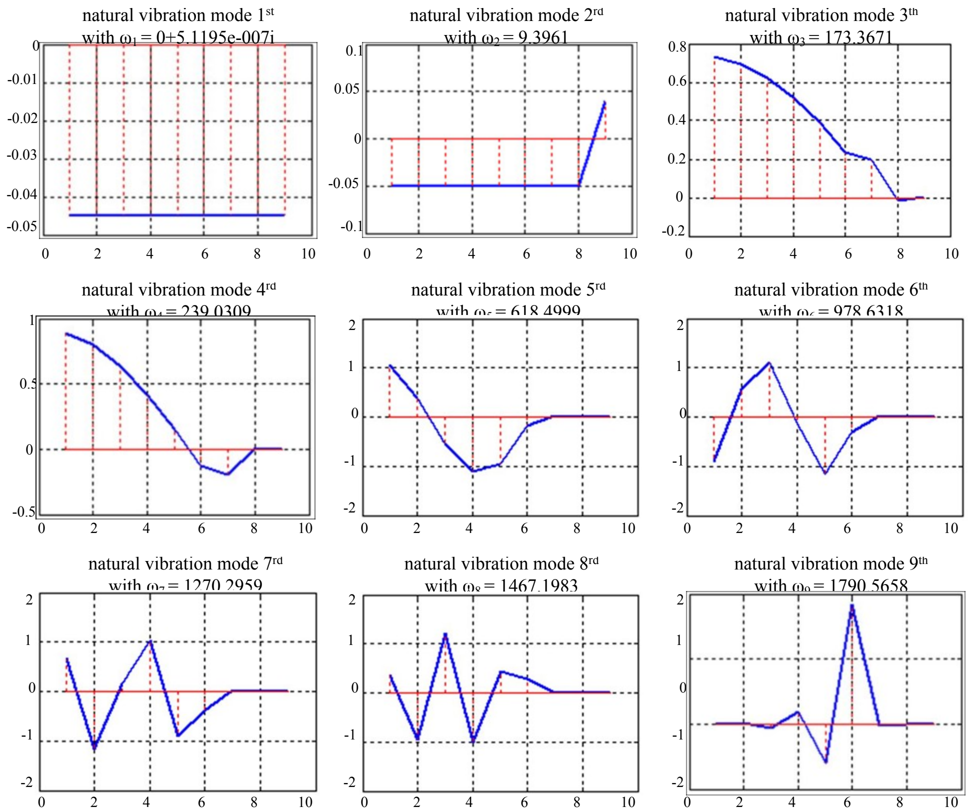


Fig. 6. *Natural vibrations modes*

Rys. 6. Postacie drgań własnych

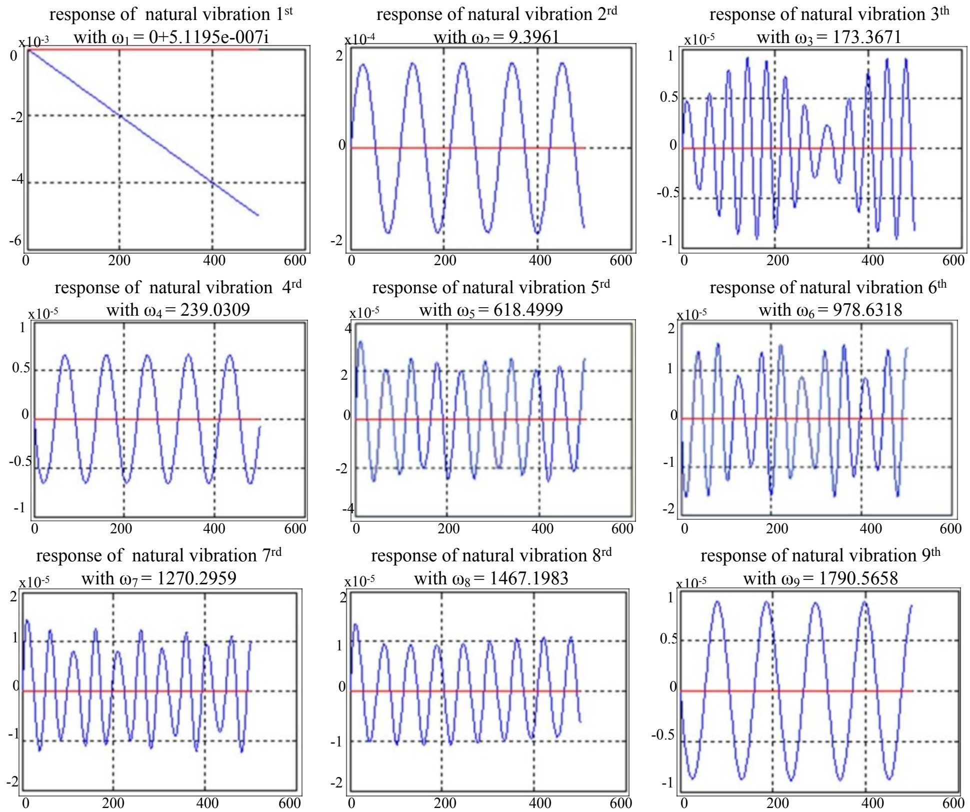


Fig. 7. *Responses of natural vibrasion*

Rys. 7. Odpowiedź drgań własnych

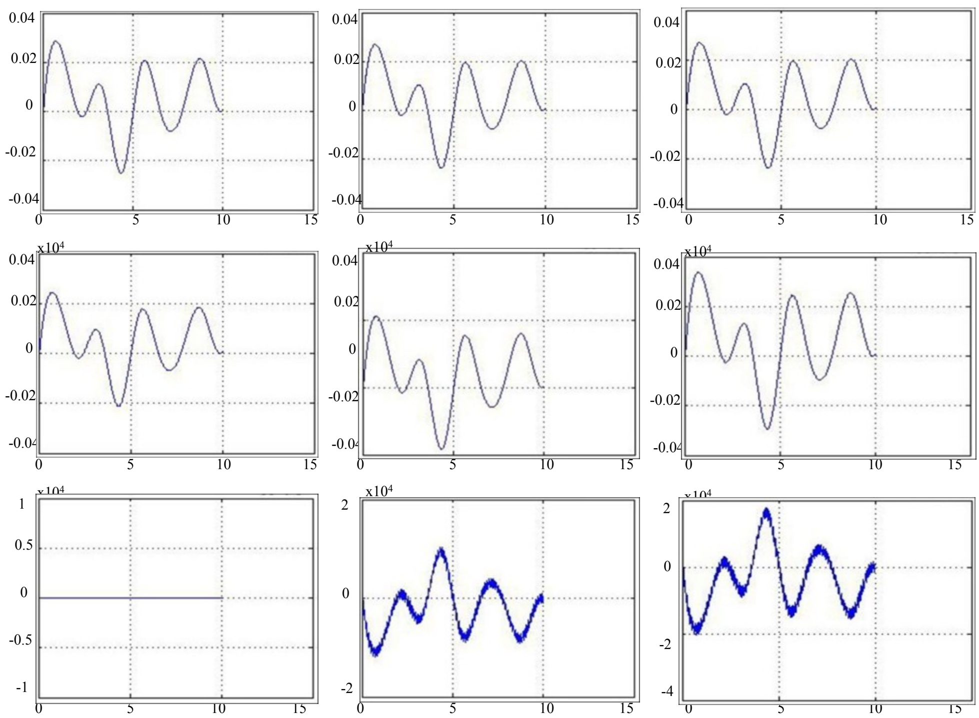


Fig. 8. *Response of force vibration*
Rys. 8. Odpowiedź drgań wymuszonych

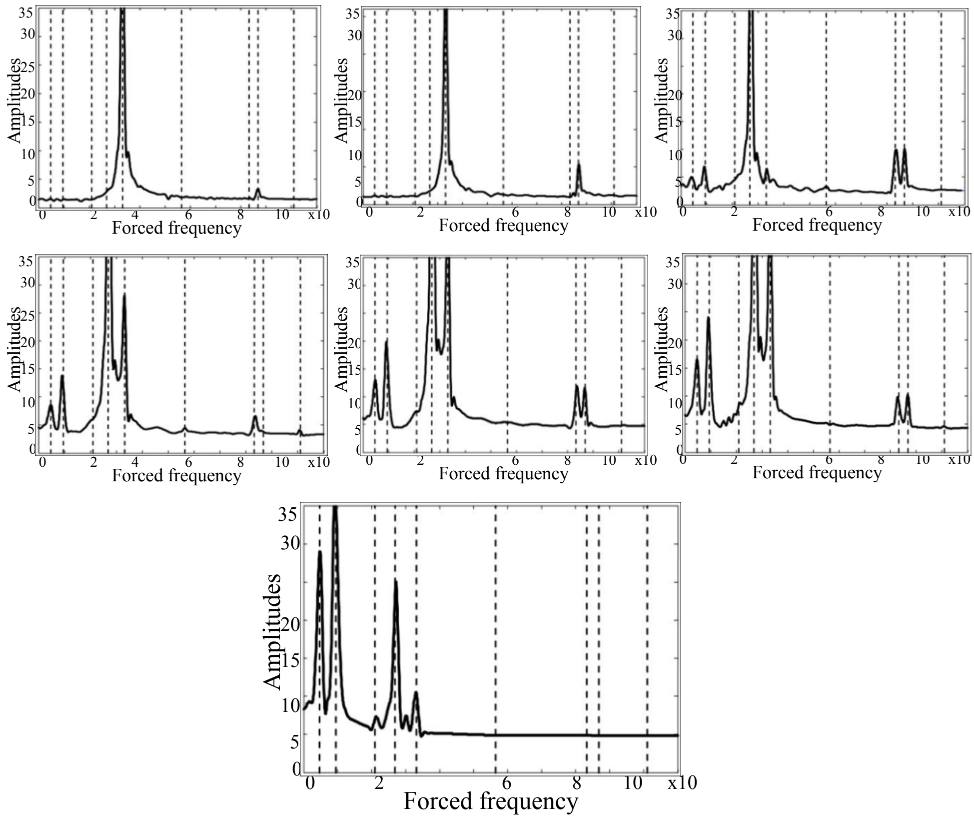


***Fig. 9.** Amplifier function form of the model*

Rys. 9. Funkcja powiększenia drgań modelu

6. CONCLUSIONS

Natural vibration modes show the possible vibration modes of the system, by which, we can predict where to measure the torsion oscillations, when using specialized equipment. Vibration form has many nodes which might be a cause of stresses changes along the shaft. Through amplification graph it can be seen that when the frequency of the force (torque generated by the engine) is close to the natural frequency, the vibration amplitude will increase very quickly (resonance). Especially at the degrees of freedom have to stimulate moments, then there will be very big vibration amplitude. Also from the results it can be found that when the frequency falls into the area from w1 to w4, the vibration amplitudes of degrees of freedom increases very quickly. The working range of the machine is also for the other frequency; it is not within the scope of this activity.

In conclusion through analysis of a specific case, we can know the issues related to vibration analysis for ship propulsion system. This result is also suitable to the requirements of the registry agency when need to appraise torsion vibration of ship propulsion system (natural frequency, amplifier function). That was previously only methods like used tole table or convert the system have multiple degrees of freedom become system only have a degree of freedom to find equivalent frequency.

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