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SOME ASPECTS OF TORSIONAL VIBRATION ANALYSIS METHODS OF MARINE POWER TRANSMISSION SYSTEMS

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Abstract

In the paper, the torsional vibrations of marine power transmission system's nonlinear method have been presented. Short presentation of marine propulsion system evolution (and its influence on ship's vibration level) in the last 30 years was included in the introduction. Some aspects of the modeling method of the elements of propulsion system have been shown. Comparison between one-degree model and 3-D Finite Element Method model was discussed. Short description of advantages and disadvantages of the undercritical and overcritical propulsion system was presented. Modeling method of propeller's mass characteristics and damping recalculation method have been shown as an example. Specialised software, for the marine power transmission system torsional vibration's analysis, made by the author, has been performed as an iterative process. Also example of torsional vibration analysis, for tanker ship, was presented in the paper. A discussion about calculation results was included in the final part of the paper. Overcritical power transmission system is better for typical ship (with slow-speed main engine and directly driven propeller).

Keywords: *torsional vibration, marine propulsion system, FEM procedure, damping, added water mass*

1. Introduction

Two-stroke, slow speed main engines are mostly installed on merchant ship since the late 70-ties (oil crisis). The engines are connected to a directly driven propeller by a relatively short shaft line. A typical marine propulsion system is presented on Fig. 1.

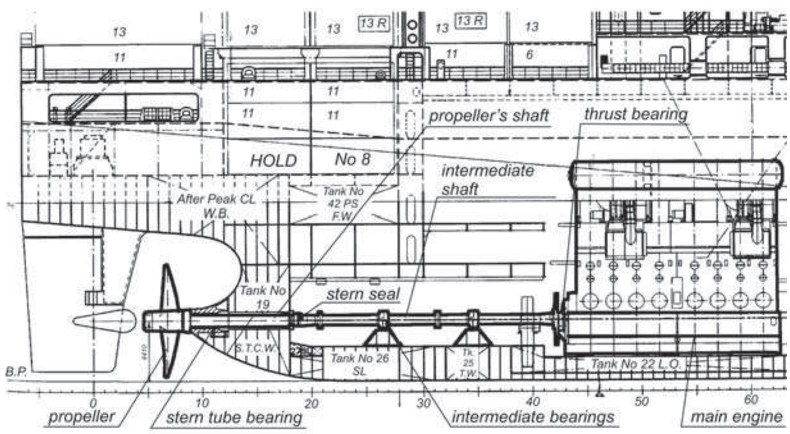

Fig. 1. Typical power transmission system

In the same time, the power from one engine's cylinder has been increased. Therefore engines have less cylinders and the engine room is shortened (cargo space is larger). Described type of propulsion system has a lot of advantages (e.g. efficiency) but is a source of relatively high vibrations level. Vibrations may have dangerous influence on crew comfort and ship's equipments strength and then on the ship safety.

The ability to predict a propulsion system operating parameters by numerical analysis is very important. During the design process all changes are possible and cheap, but when the ship is built they become nearly impossible. One of the most important parameter is torsional vibrations of the power transmission system. Torsional vibrations of the marine power transmission system are usually the most dangerous for the shaft line and the crankshaft [3].

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. Research methods of torsional vibrations have been developed since the 1950s [9, 10]. Despite so intense research, still several elements needs to be investigated; for instance: propeller damping, cylinder damping, moment of inertia of propeller's added water mass, and characteristics of specific shaft line elements like dampers, gears, elastic clutch. All those elements may have nonlinear characteristics. On the other hand, torsional vibrations are one of the main source of coupled longitudinal vibrations and dynamic excitations (on the thrust bearing) of the ship hull and deckhouse.

2. Modeling method

For calculation purposes, the reciprocating and rotating masses of the engine including the crankshaft, the intermediate shaft(s), the propeller shaft and the propeller are, modeled as a system of rotating masses (inertias) interconnected by the torsional spring. An example of model for torsional vibration analysis, of propulsion system with 6^{th} cylinder main engine is shown on Fig. 2. A power transmission system's model with one degree of freedom in each node is used. In general, the multi-node, unbounded vibration form is interesting. There is no problem with any boundary conditions. Therefore, more detailed model of the power transmission system is not required in typical analysis. For instance, detailed FEM model of the crankshaft (see Fig. 3) is used, by the author, only for determining coupling effects between torsional and longitudinal vibrations [7] or some special case of shaft line bending vibrations [8]. The gas pressure and mass forces of the engine act through the connecting rod mechanism on each crank, exciting torsional vibration in the system. Excitations have different frequencies therefore torsional frequencies are complex. The couplings influences on the torsional vibration as opposed to other vibration types are negligible. The torsional vibration is the source of longitudinal vibration excitations but not inversely.

The first question is where the main natural frequency of a system should be situated (obviously: away from the normal operating speed range). 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.

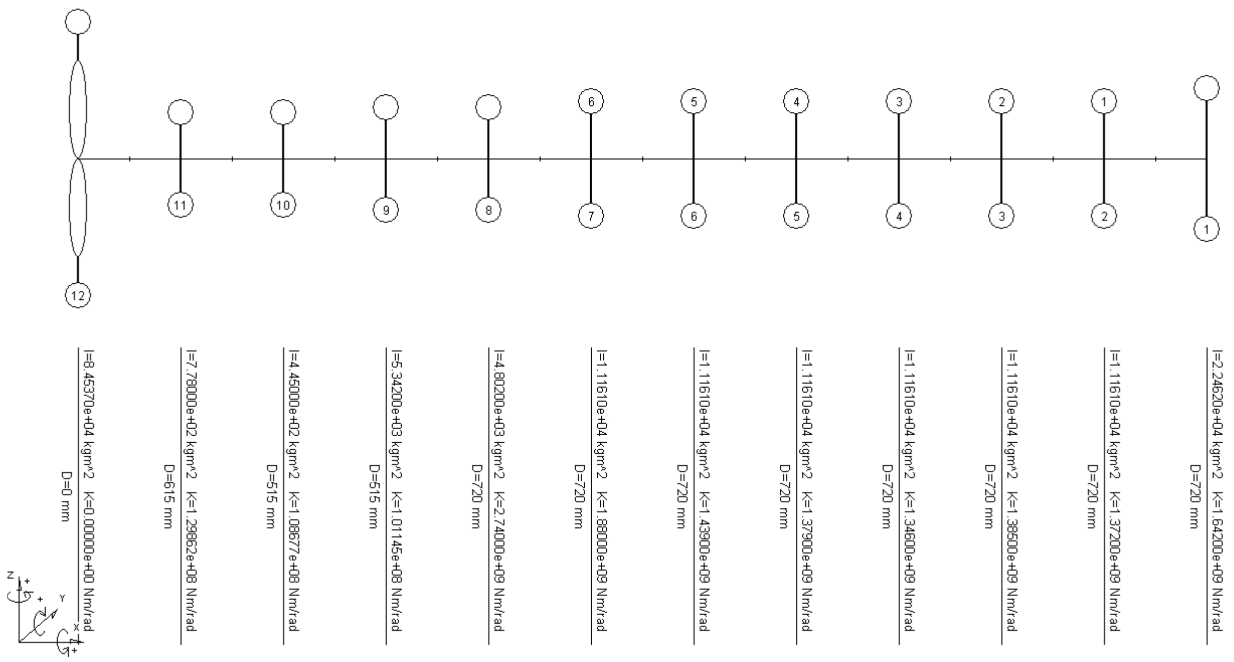

Fig. 2. Model of the power transmission system for the torsional vibration calculation

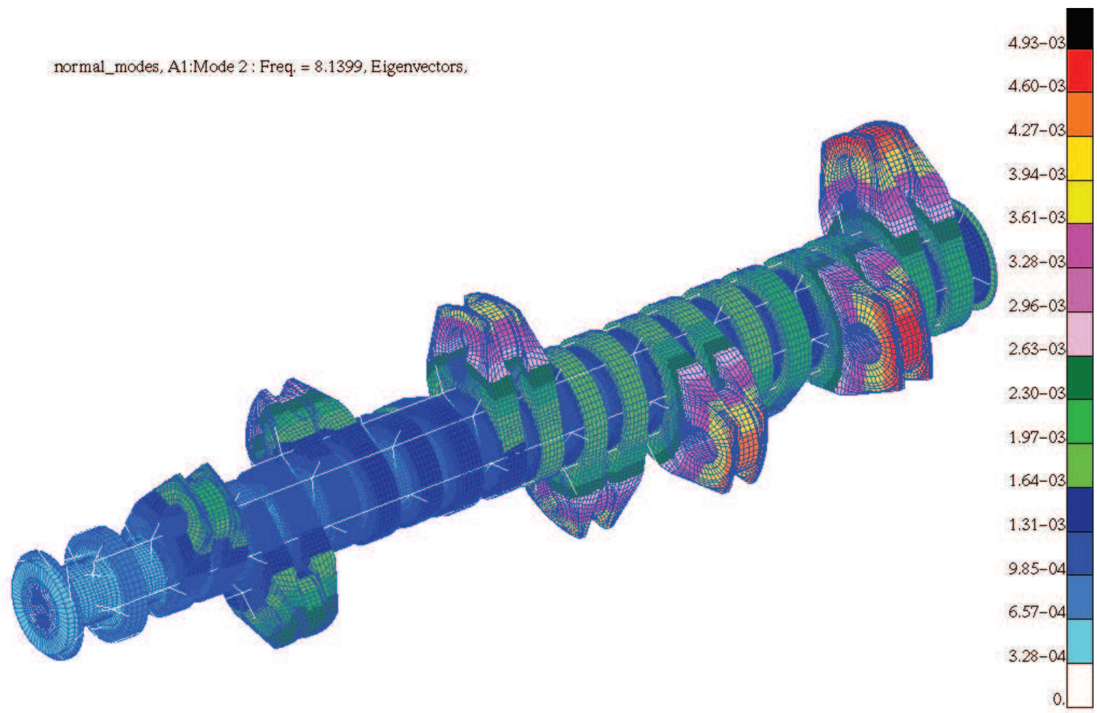

Fig. 3. Natural torsional vibrations of crankshaft of 10 K98 MC main engine

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. Overcritical layout is normally applied for engines with more than four cylinders.

Specialised software, for the marine power transmission system torsional vibration's calculations, has been made by the author. The algorithm is based on the Finite Element Method, written in Builder Borland C++. In each node only one, torsional degree of freedom is active. All characteristic matrixes (masses, dampings and stiffnesses) are related to a rotational degree of freedom. There are no geometrical stiffness matrix and gyroscopic effects. Some untypical algorithms are applied in the software; for instance elements, with nonlinear characteristics, depended on shafts rotational speed, like elastic coupler's stiff and damping characteristics, propeller and cylinder damping. Therefore the calculations have been performed as an iterative process.

There are several formulas describing propeller inertia of added water mass value [4]. The best one, in author opinion, has been derived on the basis of Parson's theory (the equation no. 1 and Tab. 1).

$$J_H = D^5\rho\left[CJ_1 + CJ_2\frac{A_e}{A_0} + CJ_3\frac{P}{D} + CJ_4\left(\frac{A_e}{A_0}\right)^2 + CJ_5\left(\frac{P}{D}\right)^2 + CJ_6\frac{A_e}{A_0}\frac{P}{D}\right], \quad (1)$$

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 14,
A_e/A_0 – expanded area blade ratio,
P/D – propeller pitch ratio.

Tab. 1. Coefficients for propeller inertia of entrained water

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 the most difficult data to determine, when a reliable damping value is lacking. What is more, there are several different methods of characterising damping phenomena. A conversion method of different damping definitions is shown in Tab. 2. 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. Some manufacturers (mostly engine factories) give us damping factors corresponding to their products.

Tab. 2. Conversion table of different damping values

	c	**ε**	**ψ**	**Q**
c	1	$\frac{\varepsilon \cdot k}{\omega}$	$\frac{\psi \cdot k}{2 \cdot \pi \cdot \omega}$	$\frac{k}{\omega} \cdot \sqrt{\frac{1}{Q^2 - 1}}$
ε	$\frac{c \cdot \omega}{k}$	1	$\frac{\psi}{2 \cdot \pi}$	$\sqrt{\frac{1}{Q^2 - 1}}$
ψ	$\frac{2 \cdot \pi \cdot \omega \cdot c}{k}$	$2 \cdot \pi \cdot \varepsilon$	1	$\frac{2 \cdot \pi}{\sqrt{Q^2 - 1}}$
Q	$\frac{\sqrt{k^2 + c^2 \cdot \omega^2}}{c \cdot \omega}$	$\frac{\sqrt{1 + \varepsilon^2}}{\varepsilon}$	$\frac{\sqrt{4 \cdot \pi^2 + \psi^2}}{\psi}$	1

where:

c – linear viscous damping [Nms/rad],
ε – undimensioned damping factor,
ψ – ratio of damping energy,
Q – vibration magnifier,
ω – phase velocity of vibration [rad/s],
k – stiffness [Nm/rad].

Beside the engine cylinder's damping (received from the producers), usually only the propeller's dampings are significant. During torsional vibration analysis other dampings are negligible, except elastic couplings and torsional dampers, if applied. Some authors (the Archer theory) make the propeller's damping dependant on torque and revolutions. But more popular formulas, and better in author opinion, make the propeller's damping dependant on its geometry. An advanced formula has been worked out by H. Dien and H. Schwanecke [4] (equation no. 2).

$$c_p = \frac{\rho \cdot \omega}{\pi} \cdot D^5 \cdot \left(\frac{P}{D}\right)^2 \cdot \frac{A_e}{A_0} \cdot 0.0231 \ \ [Nms/rad] \tag{2}$$

where:

D – propeller diameter [m],
ρ – specific mass of sea water (usually 1025 kg/m3),
ω – angular frequency [rad/s],
A_e/A_0 – expanded area blade ratio,
P/D – propeller pitch ratio.

3. Example of torsional vibration calculations

Analysis of the power transmission system torsional vibrations was performed by the author's specialised FEM software. Propulsion system of the tanker ship (245 m length, 104 000 DWT) was analysed. The propulsion system is based on a slow-speed, two-stroke, six-cylinder main engine: MAN B&W 6 S60 MC-C type. The engine main parameters are as follows: power – 13600 kW and nominal speed – 105 rpm. The propulsion system was equipped in five-blade propeller: diameter 7.2 m and mass in air – 28400 kg. Propulsion system has been designed as overcritical - main torsional resonance (43.8 rpm) is placed 58% below nominal main engine speed. The FEM model of the power transmission system is presented in fig. 2. The first five modes and frequencies of the natural vibrations were determined. One and two-node mode is presented on Fig. 4. In that case, only the first of the natural vibration mode (with one node in the intermediate shaft region) is significant. In the intermediate shaft region minimal torsional amplitudes and maximal torsional stresses are expected. Therefore, torsional vibrations verifying should be performed by strain/stress measurements (by strain gauges or optical FBG sensors). Torsional amplitudes measurement (laser method) is not good method in that case. Maximal torsional amplitudes are expected in propeller and crankshaft free-end.

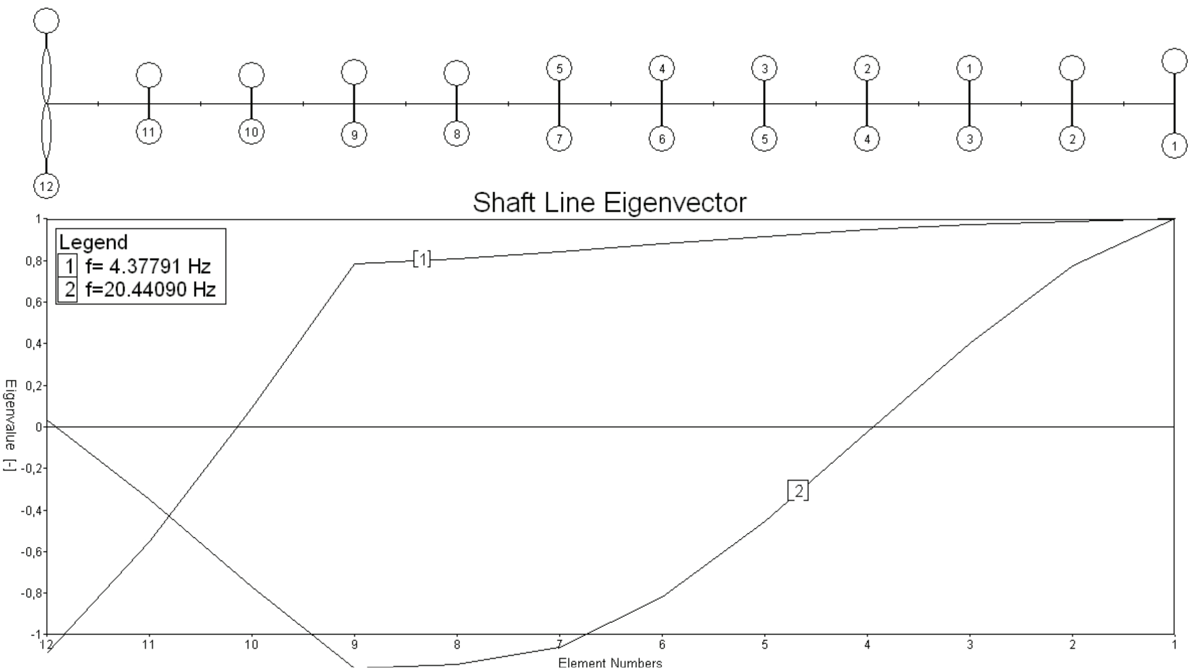

Fig. 4. Power transmission system's natural torsional vibrations

Torsional vibration amplitudes (comes from forced vibration analysis) of a propeller are presented on Fig. 5. The amplitudes are equal to 2.5° in torsional resonance region (43.8 rpm). The revolutions range between 33 and 55 rpm should be forbidden (barred speed range) for normal continuous work. Torsional vibration amplitudes in nominal main engine speed are very low; about 0.03°. In case of undercritical propulsion system amplitudes equal to 0.5° will be expected in the nominal revolutions. Therefore, overcritical power transmission system is better for analysed ship. For this engine type (two-stroke, six-cylinder) the most dangerous is the 6th harmonic component. But in some revolutions over components might be dominant. For instance 3rd stress harmonic component is the highest in the 87.5 rpm. Torsional stresses for all significant harmonic components are presented on Fig. 6. The diagram is prepared for intermediate shaft because the highest stress level. Similar figure for stresses expected in crankshaft is presented on Fig. 7.

Calculated torsional vibration stresses are located below permissible stress levels in whole range of main engine speed except barred speed range. Therefore the following limitation of the propulsion system is necessary: “a main engine speed range: 33-55 rpm has to be prohibited for continuous running in all conditions of operation; it is permitted to pass this range as quickly as possible”.

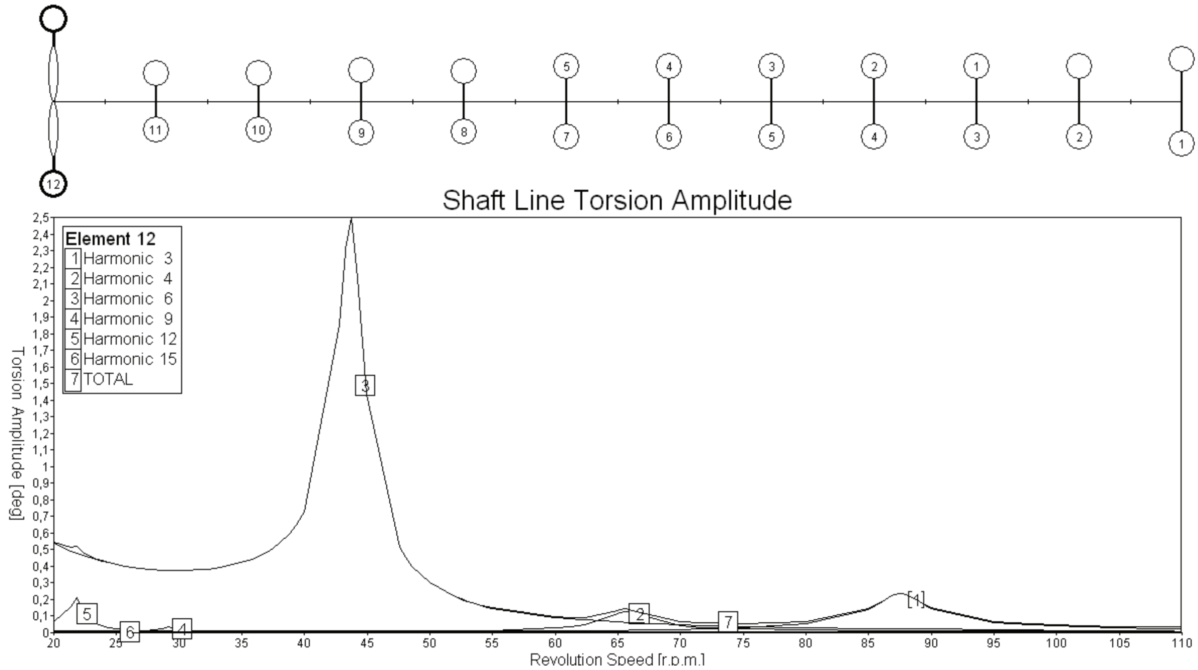

Fig. 5. Torsional vibrations amplitudes of a propeller

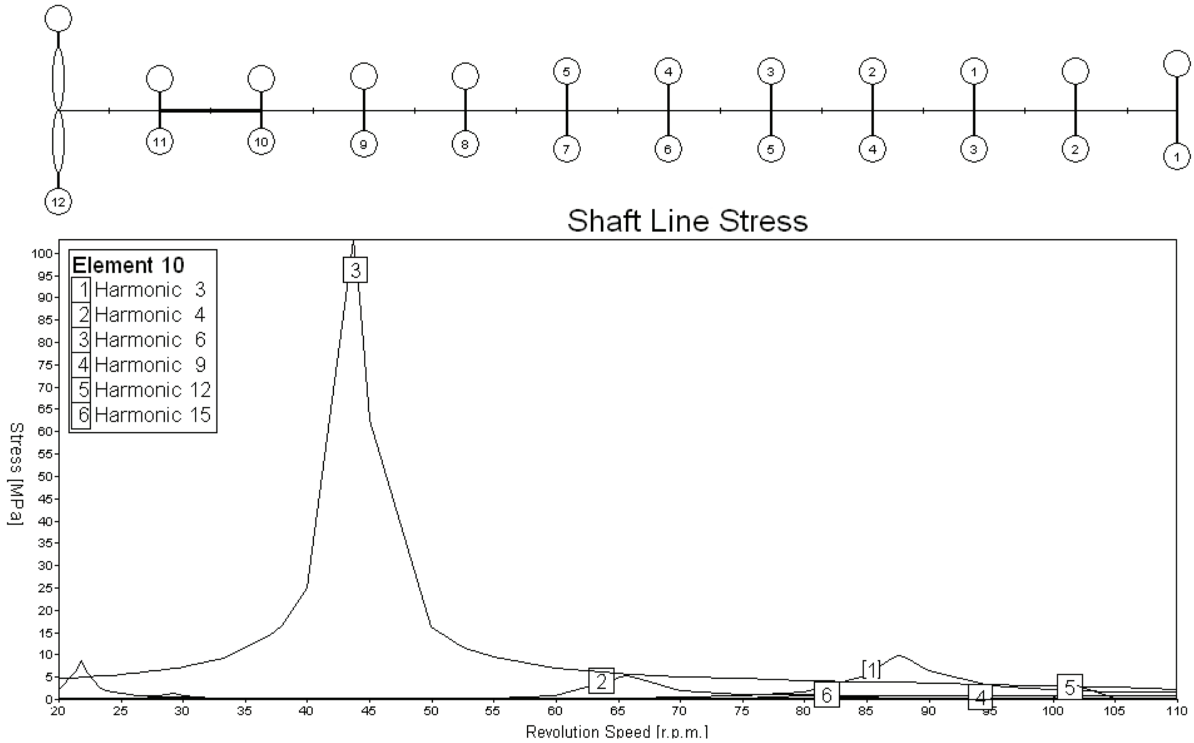

Fig. 6. Torsional vibrations stresses of an intermediate shaft

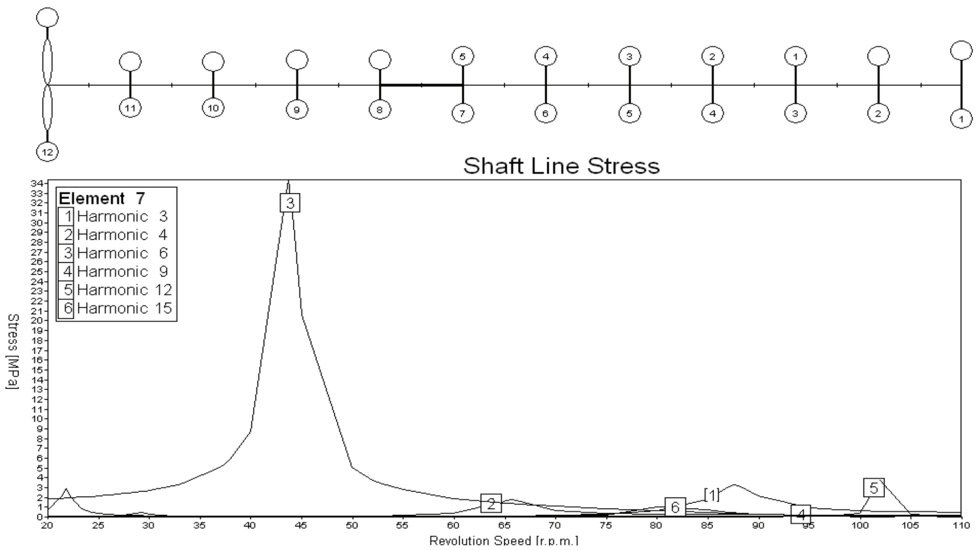

Fig. 7. Torsional vibrations stresses of an crankshaft

Similar calculations have to be performed for an engine with one cylinder misfiring [2, 5, 6]. Calculations were made for two conditions of main engine operation: cylinder no. 1 and no. 6 were not firing. After this analysis the next preliminary limitation is necessary: "a main engine speed reduction to 85 rpm is needed in the misfiring condition". Confirmation of these limits has to be done by measurements during a ship sea trial.

The calculations presented above have been performed to verify the author's algorithm. The results have been compared with the analogical calculations made by independent numerical programs [1] in an independent design office. The results conformity is very good – the differences were less than 1%.

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