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A METHOD TO ASSESS TRANSVERSE VIBRATION ENERGY OF SHIP PROPELLER SHAFT FOR DIAGNOSTIC PURPOSES

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ABSTRACT

The article discusses a key problem of ship propulsion system vibration diagnostics, which concerns assessing this part of mechanical energy transmitted from the main engine to the ship propeller which is dissipated due to propeller shaft vibration. A simplified calculation model is proposed which allows the total energy of the generated torsional vibration to be assessed from the shaft deflection amplitude measured at the mind-span point between the supports. To verify the developed model, pilot tests were performed on the laboratory rotational mechanical system test rig. In those tests, cyclic bending moment was applied to a unified (cylindrical) material sample, which modelled, at an appropriate scale, structural and functional properties of a real propeller shaft.

Keywords: rotational mechanical system, propeller shaft transverse vibration energy, calculation model

INTRODUCTION

In an ideal propulsion system, the entire energy delivered from the main engine is effectively converted to basic rotational motion of the mechanical system of propeller shafting. Progressive degradation of technical condition of structural elements composing the ship propulsion system leads to worsening of its dynamic characteristics. As a consequence, additional movements are generated, such as longitudinal, transverse and/or torsional vibrations, which accompany the basic rotational motion of the propeller shafting. These additional movements are undesirable from the point of view of the efficiency of energy conversion and transmission in the entire propulsion system. They are the cause of kinetic energy dissipation in the rotational motion and accumulation of internal energy in structural materials. After exceeding critical limits of these energies, a fatigue failure takes place, the course of which is characterised by residual energy processes of vibroacoustic and thermal nature. These processes generate observable diagnostic symptoms of changes of the technical condition of the ship propulsion shafting [Dragantchev, 2000; Szala and Boroński, 2008].

Operational experience concerning various types of ship propulsion systems show that, regardless of the area of use of these systems, primary causes of excessive shafting vibrations are most often related with the below characterised factors [Korczewski, 2017].

TRANSVERSE VIBRATION:

- wear or ageing of structural materials of elements composing the foundation for torque transmission elements. This includes: corrosion of steel washers, and ageing of rubber shock absorbers and washers made of chemically curing plastics, especially in the presence of high temperatures and chemically active products, such as, for instance, lubricating oil or diesel oil;
- wear or ageing of structural materials of flexible couplings and flexible shafting connections with auxiliary installations;
- subsidence of foundation superstructure, as a result of static deformations or impact loads acting on the ship hull (for instance during ship collision, stranding, and hitting a quay or water obstacle);

- permanent deformation of ship hull (due to structural weakening, for instance) inside which the propulsion system was installed;
- static deformation of propulsion shafting, as a consequence of long-term operational downtime;
- fouling, or material loss (loosing), as a consequence of tribological, corrosion or erosion wear of rotating elements, with the resultant nonuniform mass distribution in the rotating system;
- excitations coming from the propulsion engine and the cooperating working machines (for instance, blade system defects, deformations, weakening of connections in turbine engine rotor units and turbo-machines, or mass unbalance in the rotational and back-and-forth motion of engines and piston machines).

These causes usually lead to the following operating inability states of the ship propulsion system:

- propulsion shafting misalignment or bending;
- displacement of centres of gravity of the rotating shafting elements with respect to the rotation axis (statically or dynamically unbalanced centrifugal force).

In both cases, pressure forces increase in bearings, couplings, and gear connections of the propulsion system, all this leading to the development of various forms of tribological wear, and to the increase of assembly clearances. In those cases, the observable diagnostic symptom is the increased amplitude of transverse vibration generated at characteristic structural nodes of the mechanical system, with all above discussed energy and fatigue consequences. When the shafting misalignment or bending is high, axial vibration may also appear.

LONGITUDINAL VIBRATION RESULTING FROM:

- screw propeller damage (Fig.1);
- nonuniform load of propulsion engine cylinders;
- damage of longitudinal vibration damper.

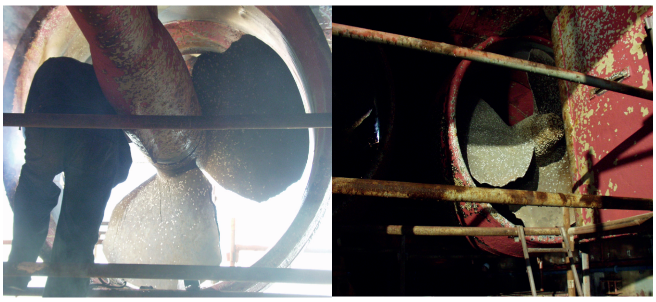


Fig. 1. Damaged blades of ship screw propellers

In those cases, the main sources of excitations are: periodically changing screw propeller thrust, transverse vibration of the hull (vertical hull movements) which provokes longitudinal vibration in thrust bearing, and periodically changing gas and mass forces in the propulsion engine which, acting on the crankpins, deform cranks and make pins move along the shaft axis.

TORSIONAL VIBRATION CAUSED BY:

- nonuniform load of propulsion engine cylinders;
- damage of torsional vibration damper.

As a consequence of excessive amplitudes of transverse, torsional and/or longitudinal vibrations, all elements involved in torque transmission from the engine to the propeller undergo accelerated fatigue wear [Korczewski, 2012 and 2017].

The above considerations show that the key metrological issue in diagnosing the rotational mechanical ship propulsion system is the ability to assess the energy of the vibrating propeller shaft from the measurement of shaft vibration parameters at selected (available) structural nodes. Excessive transverse vibration of the ship propulsion shafting is the most frequent cause of its fatigue damages, at the same time being the basic source of mechanical energy loss in the propulsion system. In this context, the problem of assessing the amount of energy dissipated by the shaft undergoing transverse vibration takes on particular relevance as a basic diagnostic symptom characterising the above discussed operating inability states of ship propulsion shafting.

CALCULATION MODEL OF SHAFT UNDERGOING TRANSVERSE VIBRATION

Even the most precisely balanced propeller shaft deflects statically under its own weight, which results in the displacement of its centre of mass with respect to the axis of rotation [Adams, 2001; Bently and Hatch 2002]. When the ship propulsion system is in operation, the generated centrifugal force is the cause of dynamic shaft deflection, which sums up with its static deflection.

The pulse which excites torsional vibration of the shaft comes from the unbalanced centrifugal force F_r applied to the shaft's centre of mass **ś**.*m*. and rotating with the shaft at angular speed $\boldsymbol{\omega}$ – Fig. 2. The action of the centrifugal force and the shaft weight force, F_G, on the propeller shaft is compensated by the restoring force F_{spr}, defined by the shaft stiffness k, and the damping force $F_{tł}$ in shaft material, bearing supports, and (possibly) oil vibration dampers (this last damping force component is characterised by a generalised damping coefficient b):

$$F_r + F_G = F_{spr} + F_{t^3} \quad (1)$$

$$m \cdot y \cdot \varpi^2 + m \cdot g = k \cdot y + b \cdot \dot{y} \quad (2)$$

The unbalanced centrifugal force is the cause of considerable dynamic loads of bearings in the analysed mechanical system. To reduce destructive effects of transverse shaft vibration, oil damping is introduced in flexible bearing supports. This solution was also applied in the physical model of rotational propulsion system [Korczewski, 2017] in which the stiffnesses of supports are not the same in all directions in the plane perpendicular to the geometrical axis of shaft rotation. Hence, an assumption was made that the trajectories of the centre of mass of the deflected shaft and its ends (with supports) have elliptical shapes, with major axes perpendicular to the axis of rotation in vertical direction – Fig.2.

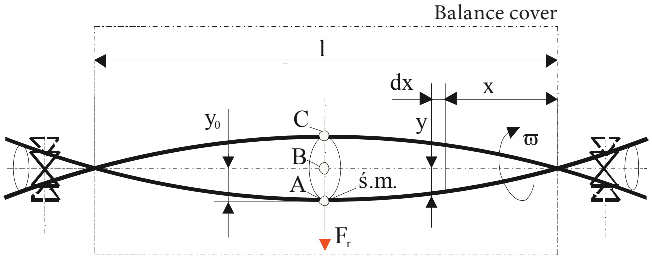


Fig. 2. Scheme of propeller shaft deflection at dynamic load with damping in bearing supports

At the first stage of the analysis, dynamic processes in flexible bearing supports are omitted. To obtain the energy balance for the shaft undergoing transverse vibration, a balance cover was defined which reduces the length of the tested shaft to the distance between two node points. Then, a formula can be worked out for total energy of the shaft mass during one transverse motion cycle, which is the sum of kinetic and potential energy – Fig.3:

$$E_c = E_k + E_p \tag{3}$$

After expanding:

$$E_k = \frac{1}{2} \cdot m \cdot v_y^2 \;\; i \;\; E_p = \frac{1}{2} \cdot k \cdot y^2 \tag{4}$$

hence

$$E_c = \frac{1}{2} \cdot m \cdot v_y^2 + \frac{1}{2} \cdot k \cdot y^2 \tag{5}$$

where:
m – shaft mass,
y – shaft deflection
v_y – velocity in transverse motion,
k – shaft stiffness.

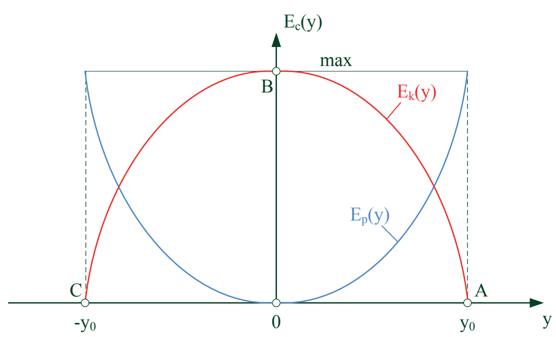


Fig.3. Changes of potential and kinetic energy in transverse shaft motion as function of shaft deflection

The kinetic energy reaches the maximum for the equilibrium position at point *B*, as the transverse motion velocity at this point is the highest (and the acceleration equals zero). On the other hand, the maximal values of potential energy occur at points *A* and *C*, at which the transverse motion velocity decreases to zero and the acceleration takes the maximum value at the maximal shaft deflection.

Under the static mass load the shaft deflects by the amplitude y_{st}, according to the weight and restoring force balance condition – Fig. 4:

$$F_G = F_{spr} \tag{6}$$

$$m \cdot g = k \cdot y_{st} \tag{7}$$

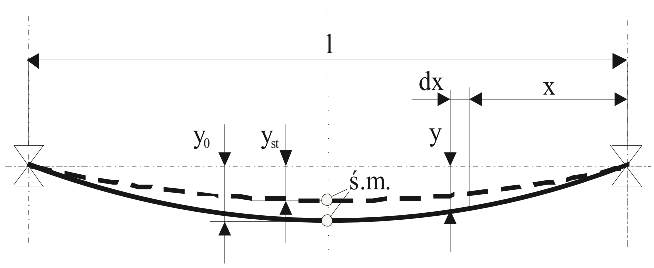


Fig. 4. Scheme of shaft deflections at static load (broken line) and dynamic load (continuous line); damping in bearing supports neglected

In the steady state the shaft stiffness for the concentrated mass positioned at the mid-span point between the supports is given by the relation:

$$k = \frac{m \cdot g}{y_{st}} \tag{8}$$

where: y_{st} – static deflection amplitude (under static mass load of self-aligning bearings).

When the system begins to rotate with angular speed **ω**, the centrifugal force F_r applied to the centre of mass of the shaft positioned on the principal axis of inertia deflects the shaft by the amplitude y_0 measured at the mid-span point between the supports. For an arbitrary point along the shaft length, an approximate relation can be derived between the average velocity v_y, time τ, and the distance *y* covered by the material point in the transverse motion:

$$v_y = \frac{y}{\tau} \tag{9}$$

where:

$$\tau = \frac{1}{f} \tag{10}$$

f – frequency of shaft deflections (transverse vibration).

Hence, the average values of material point velocity and acceleration in transverse motion can be determined from the following formulas:

$$v_y = y \cdot f \tag{11}$$

$$a_y = \frac{v_y}{\tau} = v_y \cdot f = y \cdot f^2 \tag{12}$$

Based on this assumption, the shaft stiffness can be calculated from:

$$k = \frac{m \cdot a_y}{y} = m \cdot f^2 \tag{13}$$

Assuming that the shaft deflection line is a sine curve:

$$y = y_0 \cdot \sin \pi \cdot \frac{x}{l} \tag{14}$$

and y_0 is the maximal deflection, and adopting the following boundary conditions with respect to node points:

for x=0 and x=l, deflection y=0,

the elementary kinetic energy of the shaft segment in transverse motion can be determined as:

$$dE_k = \frac{1}{2} \cdot A \cdot \rho \cdot y^2 \cdot f^2 dx \tag{15}$$

where:
A – cross-section area of the analysed shaft segment,
ρ – density of structural material of the shaft.

Taking into account the adopted boundary conditions and assuming continuous shaft mass distribution, the total kinetic energy of the shaft undergoing transverse vibration is given by the formula:

$$E_k = \frac{1}{2} \cdot A \cdot \rho \cdot f^2 \int_{x=0}^{x=l} y^2 dx \tag{16}$$

or, taking into account the assumption (14):

$$E_k = \frac{1}{2} \cdot A \cdot \rho \cdot f^2 \int_{x=0}^{x=l} y_0^2 \cdot \sin^2\left(\pi \cdot \frac{x}{l}\right) dx \tag{17}$$

Assuming that the maximal shaft deflection is constant, Equation (17) can be written in the following form:

$$E_k = \frac{1}{2} \cdot A \cdot \rho \cdot f^2 \cdot y_0^2 \int_{x=0}^{x=l} \cdot \sin^2\left(\pi \cdot \frac{x}{l}\right) dx \tag{18}$$

Since the solution of the definite integral in Equation (18) is the expression, hence:

$$E_k = \frac{1}{2} \cdot A \cdot \rho \cdot f^2 \cdot y_0^2 \cdot \frac{1}{2} \cdot l \tag{19}$$

and the final formula for kinetic energy of transverse shaft vibration is:

$$E_k = \frac{1}{4} \cdot m_p \cdot f^2 \cdot y_0^2 \tag{20}$$

where: – total mass of the analysed shaft segment.

The formula for potential energy of shaft mass in transverse motion can be derived in the same way. Taking into account the relation (13), this formula takes the following form:

$$E_p = \frac{1}{4} \cdot m_p \cdot f^2 \cdot y_0^2 \tag{21}$$

As a result, the total energy of the shaft mass undergoing transverse motion is given by the formula[1]:

$$E_c = \frac{1}{2} \cdot m_p \cdot f^2 \cdot y_0^2 \tag{22}$$

or

$$E_c = \frac{1}{2} \cdot m_p \cdot v_{y_0}^2 \tag{23}$$

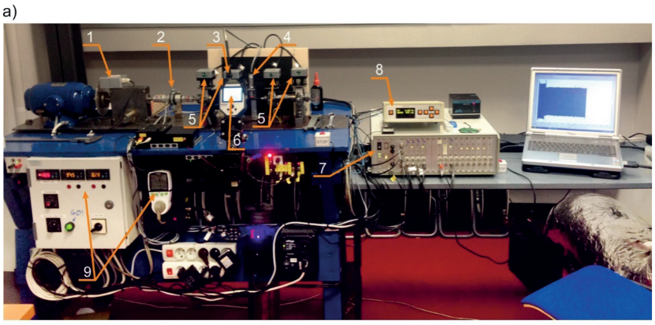
where:– amplitude of the shaft's centre of mass velocity in the transverse motion between two extreme deflections from the steady position.

1 Assuming that the motion is carried out without any energy loss for overcoming friction resistance.

RESULTS OF MEASUREMENTS AND THEIR ANALYSIS

The adequacy of the developed calculation model of energy dissipation of the propulsion shaft undergoing transverse vibration was experimentally verified on the laboratory rotational mechanical system test rig – Fig. 5 [Korczewski, 2017]. The performed tests consisted in measuring deflections of a unified (cylindrical) material sample loaded with bending-torsional moment. The tested sample modelled, at an appropriate scale, structural and functional properties of a real propeller shaft of a sea-going ship. This propeller shaft composes a dynamic system in which three vibration forms, i.e. transverse, longitudinal, and torsional vibration occur simultaneously. The object of energy analysis was only the transverse vibration of the system. Selected representative measurement results are shown in Fig. 6 in the form of time-histories of cyclic deflection changes of the centre of mass of the tested material sample rotating with angular speed of 1800 rpm and loaded with mass of 30 kg. When analysing the nature of the recorded time-histories, we can conclude that they are not pure harmonic, but are a combination of periodical signals of different amplitudes and frequencies. Therefore, the transverse vibration energy of the material sample was determined according to the relation (23) and using the square of the RMS value of the transverse vibration velocity amplitude, as this parameter takes into account both the history of the recorded signal, and its amplitude (the amplitude value better characterises the intensity of the generated vibration).

a)



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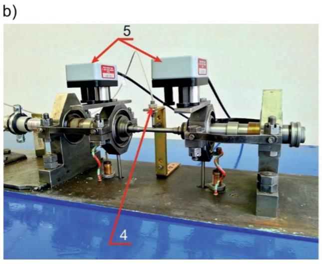

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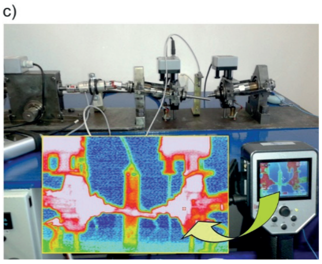


Fig. 5. General view of laboratory rotational propulsion system test rig: a) rig with measuring instrumentation – recorders: thermovision (6), acoustic emission (7), torque and rotational speed (8), electric current parameters (9); converters: rotational speed (1), torque (2), vibration acceleration (3), sample eccentricity (deflection) (4), acoustic emission (5); b) method of fastening of acoustic emission converters (5) and sample eccentricity (deflection) converter (4); c) observation of radiative emission of the sample

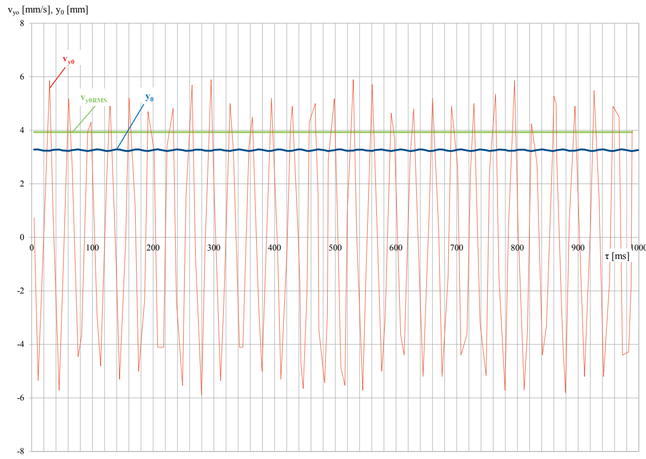
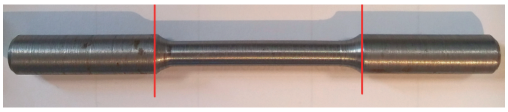


Fig. 6. Time-histories of the centre of mass deflection (y_0), transverse vibration velocity (v_{y0}), and the RMS value of transverse vibration velocity amplitude (v_{y0RMS}) of the tested material sample

The mass m_p of the selected sample segment was 0,028 kg (the mass of the entire sample was 0,108 kg) – Fig. 7. After placing numerical values of physical quantities to Equation (23), the total energy of the selected sample segment undergoing transverse motion was obtained. Its value for one cycle of periodic motion was equal to $0{,}22{\cdot}10^{-6}$ J. Thirty cycles of this motion are executed during one second, therefore the total amount of energy generated by transverse vibration in one second was $6{,}6\cdot10^{-6}$ J.

a)



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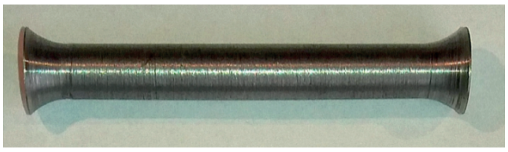

Fig.7. Tested material sample: a) general view, b) selected (tested) sample segment

Taking into account the value of the kinetic energy of the selected sample segment undergoing rotational motion, which in that time (for one second) was approximately equal to 33 J (Fig. 8), the amount of useful energy dissipated in the transverse vibration can be considered negligible. On the other hand, it is of some importance for long-term operation of a real mechanical system, where the masses of propulsion shafting equal several tonnes, as it amounts to several kilojoules per one hour of system operation.

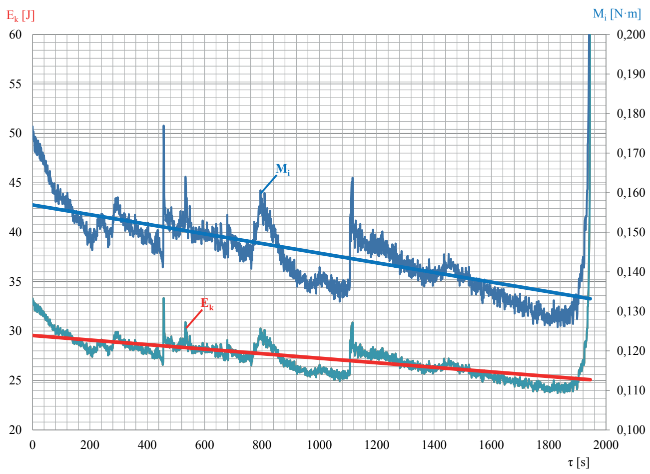

Fig.8. Time-histories of transmitted torque (Mi) and kinetic energy (Ek) in rotational motion of the selected material sample segment, with corresponding trend lines

FINAL REMARKS AND CONCLUSIONS

The performed model tests and experimental research have shown that the centre of mass deflection measurement of a selected propeller shaft segment in the rotational mechanical system can be used for assessing the energy of the generated transverse vibration. The amount of shaft energy relating to its transverse vibration is the measure of kinetic energy dissipation in the rotational motion of the system, and can be used in ship propulsion system diagnostics to identify propulsion shafting misalignment or bending, without the need to shut the ship propulsion system down.

Moreover, estimating the amount of energy generated by the propeller shaft undergoing transverse vibration provides opportunities for evaluating the operation of the ship propulsion system during energy transmission and conversion to work and heat forms. The final result of this activity will be the function describing the fatigue life of ship shafting loaded with bending moment, which can be used in operating diagnostics of ship propulsion systems.

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