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IDENTIFICATION OF SHIP PROPELLER TORSIONAL VIBRATIONS

Jan Rosłanowski

Gdynia Maritime University, Faculty of Marine Engineering
Morska Street 81-87, 81-225 Gdynia, Poland
tel.: +48 58 6901323, fax: + 48 58 6901399
e-mail: rosa@am.gdynia.pl

Abstract

The article presents identification methods of ship propeller torsional vibrations in dimensional space and ways for their dumping. Propeller torsional vibrations are caused by static, dynamic and hydrodynamic unbalance of propeller itself and its operation in non-uniform velocity field of ship hull. The propeller working in uneven velocity field of the hull creates cyclic thrust forces, peripheral vibrations of shaft as well as flexural and torsional load of propeller shaft. Vibrations cause additional thrust load on shaft bearings and shaft bending moment. Vibrations of ship hull, particularly its stern section cause variation of pressure field created by propeller. Amplitude of particular parts of the hull comes from the action of cyclic forces created by working ship propeller. They can be very large. Therefore, the ways to reduce vibrations caused by running propeller are very important. Given in the article methods of vibration dumping calculations conduct to determination of vibration dumping coefficients. Calculation results are comparable to measuring results in spite of fact they were determined under assumption of different velocity fields of water flowing through propeller circle. Amplitudes of vibrations depend on number of propeller blades and pulsate with the basic frequency equal to product of the rotational speed and number of blades. Causes of such vibrations and additional load of vessel stern are described in the article. An attention was paid to assure the greatest possible uniformity of velocity field by adequate design of stern shape and accessories. The ways to reduce torsional vibrations of propeller by the appropriate selection of blades number and their accurately machining are also given in the article.

Keywords: *transformation of energy, operation of the engine, dimensional space, dimensional functions*

1. Introduction

There are three causes of vibrations generated by the ship propeller:
1) mechanical and hydrodynamic unbalance of the propeller,
2) heterogeneous velocity field in the propeller circle,
3) pressure pulsation of water stream in the radial direction of the propeller to the shell plating.

Elementary reason of vibrations generated by the propeller is heterogeneous velocity field of water in the propeller circle caused by the presence of the hull, wavy motion of the sea and skew inflow to the propeller. The above factors bring about induction of variables on the blades, and under pressure cause pulsation of the thrust and the moment, which in turn, generate torsional and longitudinal vibrations of the propeller, additionally imposing a load on the hull plating [1-3].

Pulsations of thrust and moment forces in harmonic analysis of the propeller vibrations include component-dominating frequency, so called the blade one, equal to:

$$f = n \cdot z\,, \tag{1}$$

where:
n – propeller revolutions in [1/s],
z – number of blades.

Spectral analysis of propeller vibrations includes also components of lesser values, possessing a frequency, which is a multiple of blade frequency. Amplitudes of variable thrust forces depend on the number of blades. By larger number of blades, amplitudes decrease. For even number of blades, they decrease more slowly than uneven number. Reverse dependence takes place in case of longitudinal vibration of the propeller [4, 6].

2. Calculation methods of torsional vibration of the propeller

It is assumed in calculations, that damping of ship propeller vibration is of viscous character the propeller vibrating in water increases its inertia in the presence of accompanying water. The mass of accompanying water is assumed as an addition of 25% inertia moment of the propeller GD^2. For propellers of trade ships assuming such mass of accompanying water results in positive accordance with reality [4-6].

Rotational moment created on the propeller wing by circumferential forces, can be calculated from the formula:

$$M = C_1 \cdot \omega^q , \tag{2}$$

where:

M – rotational moment in [Nm],

C_1 – coefficient of resistance in [kg·m²/s],

q – power exponent in [-].

The formula (2) is the consequence of the fact that pulsations of rotational moment of the propeller take place at basic wing frequency, and angular velocity of the propeller is a derivative of rotation angle ϕ. Therefore, damping of propeller vibrations can be determined on the basis of the following formula:

$$b_\varphi = \frac{\partial M}{\partial \omega} = C_1 \cdot q \cdot \omega^{q-1} = \frac{M}{2\pi n} \cdot q , \tag{3}$$

where notations as in formula (2).

Research carried out by Frahm pointed out that power exponent of angular velocity $\boldsymbol{q}$ in propeller rotational moment formula is included in the following range [3, 4, 6]. Therefore, Frahm proposed 3.8 value [6]. In calculation, methods of propeller vibrations damping proposed by Archer and Lerbs, they use formula (3) and hydrodynamical characteristic of the propeller, assuming constant speed of the ship:

$$b = \frac{\Delta M}{\Delta \omega} = \frac{M}{2\pi n} \cdot (2 - E_M) , \tag{4}$$

where:

E_M – coefficient of propeller load by rotational moment, $E_M = \frac{\mathrm{d}k_M}{\mathrm{d}J} \cdot \frac{J}{k_M}$,

k_M – coefficient of rotational moment, $k_M = \frac{M}{\rho \cdot n^2 \cdot D^5}$,

J – coefficient of propeller advance, $J = \frac{v_p}{n \cdot D}$,

D – diameter of propulsion propeller in [m],

n – rotational propeller velocity in [1/s],

ρ – thickness of sea water in [kg/m³].

Coefficient of propeller load moment E_M determines the sensivity with which it reacts to a change of quantity of flowing water. It possesses negative values, which means that an increase of propeller rotations causes additional positive change of advance coefficient, and an increase of speed has a negative influence. The above calculations of propeller vibrations damping assume a steady inflow of water in a stationary velocity field. It means that calculations by these methods are true only for vibrations of very small frequency. Calculations of hydrodynamic forces of vibrating propeller, which take advantage of theory of flat non-stationary flow through its circle, are true only for infinitely long blades. It results in the fact that received values of forces are overestimated. Absolute length of blades causes decrease of quantities of hydrodynamic forces caused by flowing around the ends of the blades. In case when the frequency of water rotated by the propeller is a multiple of blade frequency, then the quantity of hydrodynamic forces defines damping and additional mass of accompanying. Additional mass of water rotating with the propeller does not depend on vibration frequency.

Tab. 1. Setting up of formulas given by Schwanecke and Schuster that allows calculating torsional vibrations of the propeller [1]

Quantities characterizing vibrations	Formulas proposed by Schuster	Formulas introduced by Schwanecke
Mass of accompanying water m_{hyd}	$\frac{\rho \pi D^5}{128}\cdot\left[\frac{1-\frac{(H/D)^2}{4}}{1-r_0^*}\cdot\left(\frac{A}{A_0}\right)^2\cdot\frac{(H/D)^2}{z}\right]$	$\frac{\rho D^5}{\pi z}\cdot\left(\frac{H}{D}\right)^2\cdot\left(\frac{A}{A_0}\right)^2\cdot 0.0703$
Damping b_φ	$\frac{\rho D^5 \omega}{128}\cdot\left[\left(1-\frac{(H/D)^2}{4}\right)\cdot\left(\frac{A}{A_0}\right)\cdot\left(\frac{H}{D}\right)^2\right]$	$\frac{\rho D^5 \omega}{\pi}\cdot\left(\frac{H}{D}\right)^2\cdot\left(\frac{A}{A_0}\right)\cdot 0.0231$

In table one there are formulas proposed by Schwanecke and Schuster [6] allowing in an approximated way to calculate damping of torsional vibrations generated by the propeller. These formulas consist of dimensional and nondimensional parts. Dimensional part includes diameter of the propeller and thickness of the sea water. Nondimensional part consists of surface coefficient and the pitch of the propeller. Such calculations can be compared with the calculations received on the basis of hydrodynamic characteristic of the propeller. Such formulas are introduced by Frahm and are of the following form:

$$m_{hyd}=\frac{M}{\omega^2}\cdot C_{m_\varphi}=\frac{M}{\omega^2}\cdot\left[\frac{4\pi^3}{128\cdot k_M}\left(\frac{1-\frac{(H/D)^2}{4}}{1-r_0^*}\cdot\left(\frac{A}{A_0}\right)^2\cdot\frac{(H/D)^2}{z}\right)\right],$$
$$b_\varphi=\frac{M}{\omega}\cdot C_{b_\varphi}=\frac{M}{\omega}\left[\frac{4\pi^2}{128\cdot k_M}\cdot\left(1-\frac{(H/D)^2}{4}\right)\cdot\left(\frac{A}{A_0}\right)\cdot\left(\frac{H}{D}\right)^2\right], \quad (5)$$

where:

r_0^* – nondimensional main radius of the wing, $r_0^*=2R_N/D$,
R_N – radius of the propeller circle in [m],
H/D – coefficient of the propeller pitch in [–],
A/A_0 – developing coefficient of blade surface in [–],
D – diameter of the propeller in [m],
z – number of propeller blades [–],
ω – angular velocity of the propeller in [rad/s],
M – torque produced by circumferential forces of the propeller blades in [Nm],
m_{hyd} – mass of water accompanying in torsional vibrations in [kg],
b_φ – damping of torsional vibrations in [Nms].

Damping of vibrations by the propeller, determined on the basis of hydrodynamic characteristic is proportional to rotating velocity, and is proportional to advance coefficient. Circumferential forces on propeller blades cause the moment being approximately a square function of rotating speed. It enables to determine mass of water rotating with the propeller in given work conditions through dividing the moment by ω^2. An increase of rotating speed of the propeller causes proportional increase of vibration damping. In Fig. 1, one can see the coefficient of torsional vibration damping in the function of propeller advance coefficient. One can see the diagram with the values of calculated from the formulas given in Tab. 1 and proposed by Archer.

Coefficient values of torsional vibration damping, received from different formulas are in accordance with one another with small exceptions.

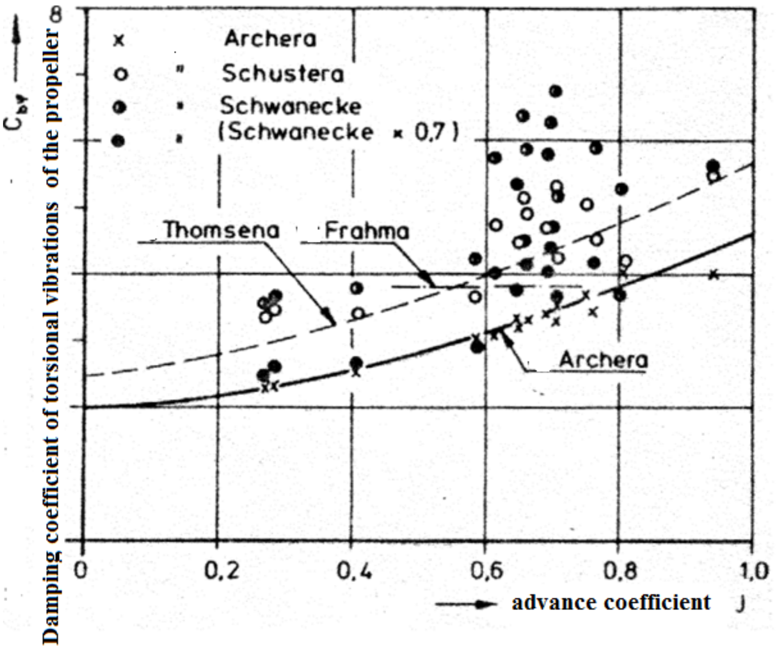

Fig. 1. Coefficient of torsional vibrations damping of the propeller and function of advance coefficient with plotted values received from different formulas [1]

The values calculated from the formula by Schwanecke are higher than those of the formula by Schuster are. In order to receive consistent values, one should multiply then by 0.7. On the basis of research, Thomson claimed the necessity to increase by 25% of the values of damping coefficients received from Archers formula. The formula proposed by Thomson subjects the size of correction to a number of blades and coefficient of propeller surface in the following way:

$$\chi_{Th} = \frac{1}{1.05 + \frac{9.6}{z^2} \cdot \left(\frac{A}{A_0} \right)^2}, \tag{6}$$

where:

z – number of blades,

A/A_0 – coefficient of propeller surface.

In fig. 1 the values of damping coefficient calculated by Thomsen, have been pointed out [7]. One should pay attention to a high accordance in values despite the fact that coefficient calculations were carried out, assuming that, there were different velocity fields of water flowing through the propeller circle. Coefficient of torsional vibrations damping, calculated with the help of the above formulas, have been verified with the measurements results of propeller vibrations directly cooperating with the 8-cylinder propulsion engine. The above engine cooperated with the propeller of 3 or 5 blades, in order to avoid its overloading. Measurements results enabled them to determine mean value of torsional propeller vibrations coefficient of 3 or 5 blades equal to $C_{b\varphi} = 4.5$ [1].

In Fig. 2, we can see the coefficient of torsional vibrations damping $C_{m\varphi}$, caused by the mass of water rotating with the propeller in the function of advance coefficient. In the above figure, one can see the values of vibrations damping calculations caused by the mass of rotating water, calculated according to Schuster and Schwanecke formulas and their correction. Damping through the mass of water accompanying the propeller, depends in a curvilinear way on the advance coefficient, as well as, the damping of the propeller as well (see Fig. 1).

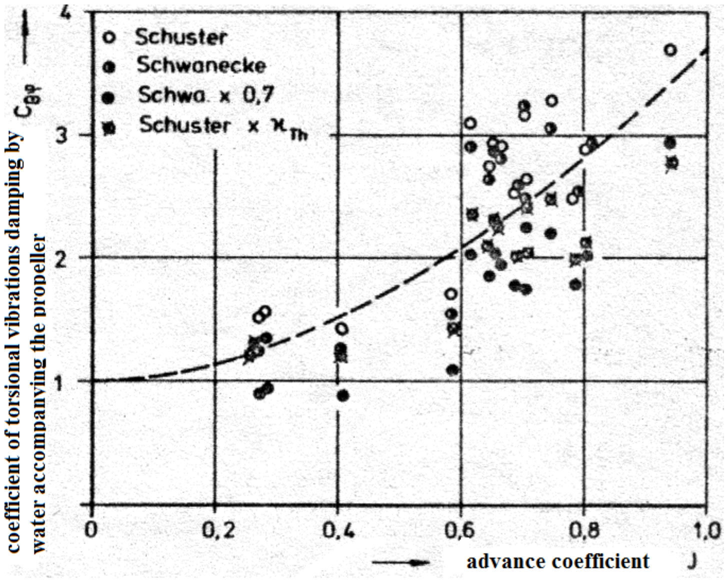

Fig. 2. Coefficient of torsional vibrations damping by the mass of water accompanying the propeller in the function of advance coefficient with calculations according to Schuster and Schwanecke [1]

Pulsations of hydrodynamic force caused by the propeller are unmistakably determined by:
– conditions of the water flow through the propeller circle,
– geometric position of the propeller in a stream of flowing water,
– deflection of the centre of gravity of propeller blades from the plane perpendicular to the axis.

Distribution of water speed flowing through the propeller circle depends on currents of limiting layer and dead water. Because of this, a certain dependence between the shape of stern frame and component water speed comes into being. Components of velocity changes of low frequency are the reason of pushing water up to the surface. This phenomenon disappears, however by increased number of components. Vibrations caused by the velocity components of flowing water through the propeller circle may possess great energy [1-3, 5, 7].

In Fig. 3, we can see exemplary changes of the moment in time, caused by torsional vibrations of the 5-blade propeller. Fig. 3 presents dimensionless quantity of moments amplitude and its formula $\bar{M}_{x_{mz}} / M = 0.02$ with mean value $v/n = 0.1$ and responding angle of phase displacement equal to $\varphi_{M_{x5}} = +2.5$ [rad] which vas read of from Fig. 4 and 5.

Changes of the propeller moment in the arrangement shown in Fig. 3a are expressed by the following formula:

$$M_{x_{mz}}(t) = \bar{M}_{x_{mz}} \cdot \cos\left(m\, z\, \omega\, t - \varphi_{M_{x_{mz}}}\right), \tag{7}$$

where:
$\bar{M}_{x_{mz}}$ – amplitudes mean value of the rotating moment in [Nm],
z – number of blades,
m – number defining the form (order) of propeller vibrations, m = 1, 2, 3,
t – time of propeller rotation in [s],
$\varphi_{M_{x_{mz}}}$ – phase displacement of moments pulsation in [rad].

It has been assumed that circumferential forces are generated on the radius line of the propeller blade. This line is a geometric place of midpoints of propeller blade surface *YZ*, has is the plane going through chord centres of blade profile at the length of 0.7 propeller radius from the midpoint of rotation. At such assumption, circumferential force on the propeller blade generates the highest positive moment in a position where its generating line and the perpendicular create the 28.7° angle, measured in the direction of the propeller rotations (corresponding with 2.5/5 rad).

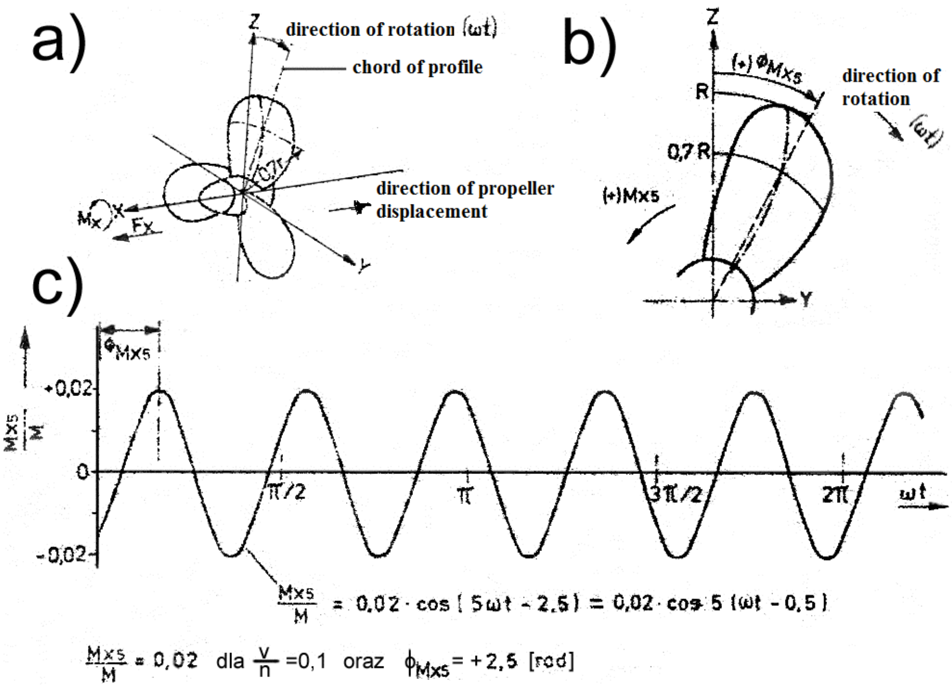

Fig. 3. Course of the moment caused by torsional vibrations of 5 blade propeller; a) arrangement of coordinates describing motion of the propeller; b) plane of torsional vibrations; c) course of rotation moment vibrations

Calculations of torsional vibrations of the propeller require the knowledge of condition concerning inflowing stream of water on the propeller. In order to achieve the above knowledge, it is necessary to know the results of hydrodynamic research concerning the model of designed ship.

To describe vibrations of different propulsion propeller it is comfortable to use nondimensional quantities. Such quantities can be obtained after eliminating geometric parameters of the propeller from the moment formula (7). In this way, determined nondimensional quantities present the moment of the propeller for mean values of geometric parameters.

Research theoretical works exposed in [1, 5-7] showed that amplitudes of the moment are proportional to the product of rotational and progressive velocity of the propeller $n \cdot v$. On the other hand, the mean moment is proportional to the square of rotational speed of the propeller, which means that dimensionless quantities depend on the size of relation v/n and the number of blades. Phase shifts between pulsations of circumferential forces generated on the propeller blades and causing pulsations of the moment depend on their position on the hub.

Figure 3a presents the propeller blade mounted in the hub vertically upwards, which produces an axial force of thrust. Phase displacement between amplitudes of moment and thrust is equal to π and depends on the choice of positive coordinate X.

Figures 4 and 5 present dual mean values, the highest and mean values of the three highest dimensionless amplitudes of vibrating moment for the propeller of blade numbers 3, 4, 5, 6 and 7 working in different conditions. Amplitudes doubling of the moment in Fig. 4 allow analysing vibrations of frequency higher than the basic ones. Asymmetric mean moment is possible to be introduced only by means of cumulative quantities of vibrations, for example, vertex – vertex. Similarly, introduction of phase displacement angles of the moment, in this case, is impossible.

In addition, in figures 4 and 5 they have shown limiting values which cause, acceptable by shipyards, vibrations of the ships stern frame caused by the working propeller [6]. So as not to cross limiting values, one should choose, either optimum number of propeller blades or the suitable shape of the hulls stern frame. In fig. 6, one can notice phase displacement of vibrations of the propeller rotating moment with different number of propeller blades.

The ranges, within which one can find the values constituting 50% of all calculated phase angles, have also been marked.

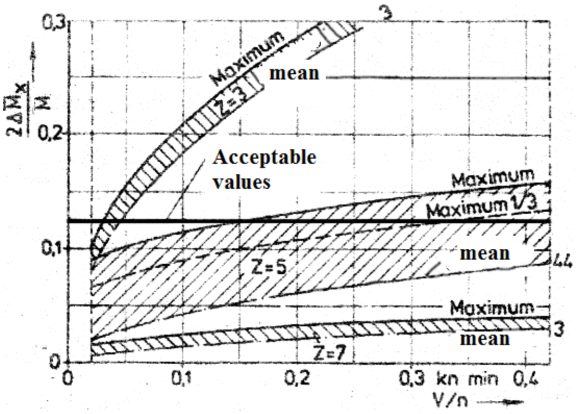

Fig. 4. Double mean values, maximum and mean values of the three highest dimensionless amplitudes of vibrating moment for the propeller of blade numbers 3, 5 and 7 in different work conditions [1]

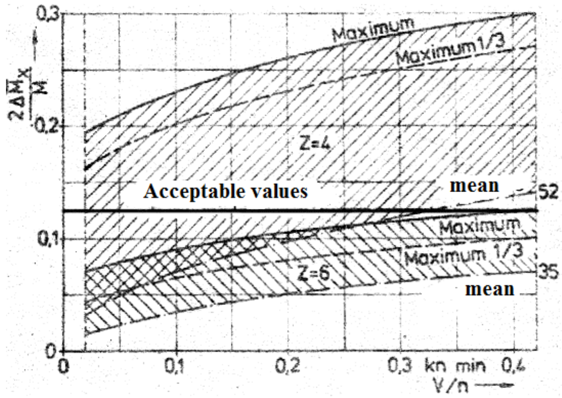

Fig. 5. Double means values, the highest and mean values of the three highest dimensionless amplitudes of vibrating moment for the propeller of blade numbers 4 and 6 in different work conditions [1]

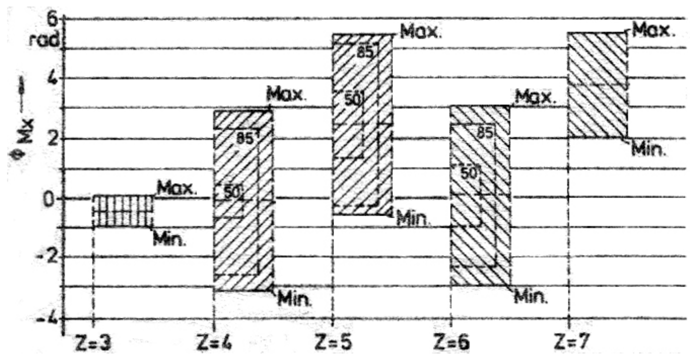

Fig. 6. The phase displacement of vibrations of the propeller rotating moment with different number of propeller blades 3, 4, 5, 6 and 7 [1]

3. Conclusions

Analysis of propeller torsional vibrations, by means of formulas proposed by different authors, allows formulating, below mentioned, observations:

1. Coefficients values of torsional vibration damping by the propeller calculated, assuming, stationary and nonstationary velocity field of water inflow are compatible.
2. Friction of inflowing water on the propeller blades has influence on damping of torsional vibrations.
3. Calculations of torsional vibrations damping by the propeller, can be carried out by means of formulas given in Tab. 1.

4. Coefficient value correction of torsional vibration damping by the propeller, should be, best of all, carried out according to Thomsen formula (6).
5. Corrected values of coefficient damping of torsional vibrations can be compared with diagrams presented in Fig. 1 and 2.

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