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METHODOLOGY OF PROPELLER STRENGTH CALCULATIONS IN SETTLED AND EMERGING WORK CONDITIONS

Key words: propeller strength, numerical calculations, finite element method, added water mass.

Abstract: In the paper, the methodology of propeller strength analyses is presented. Numerical calculations based on the finite element method were used during the analyses. Analyses methodology is based on solid state mechanics with loadings determined by fluid mechanics calculations. Several propulsion working conditions (including steady-state and transient fluid flow) were taken into account. In order to determine the optimal modelling method of the propeller, several different numerical models were compared, including a free model of the whole propeller and single blade with boundary conditions placed in the foot. The propeller optimisation was the main target of the analyses. After numerical calculations, the propeller mass saving (in comparison to classification societies' empirical formulas) was achieved.

Metodologia obliczeń wytrzymałości śruby napędowej w ustalonych i nieustalonych warunkach pracy

Słowa kluczowe: wytrzymałość śruby okrętowej, obliczenia numeryczne, metoda elementów skończonych, dodana masa wody.

Streszczenie: W pracy przedstawiono metodologię analiz wytrzymałościowych okrętowych śrub napędowych. Obliczenia numeryczne zostały oparte o metodę elementów skończonych. Metodologia analiz bazuje na mechanice ciała stałego z obciążeniami wyznaczanymi na podstawie mechaniki płynów. Szereg warunków pracy układu napędowego (włącznie z ustalonymi i nieustalonymi przepływami płynów) zostało wziętych pod uwagę podczas analiz. W celu wyznaczenia optymalnej metodologii obliczeń, zostały porównane różne modele śruby napędowej włącznie ze swobodnym modelem całej śruby napędowej oraz z pojedynczym skrzydłem śruby z warunkami brzegowymi umieszczonymi na jej stopie. Głównym celem analiz była optymalizacja masy śruby napędowej. W wyniku obliczeń numerycznych osiągnięto znaczną oszczędność masy śruby napędowej w porównaniu do śruby zaprojektowanej zgodnie ze wzorami empirycznymi towarzystw klasyfikacyjnych.

Introduction

Strength of the propellers with the skew-back greater than 25° has to be numerically analysed according to marine classification societies [1, 2]. The finite element method (FEM) is advised for those kinds of calculations [3]. Classical and typical propellers (skew-back < 25°) may be designed on the basis of empirical equations given by the societies. The minimal thickness of the propeller blade is determined by the equations. Each classification society has their own empirical equation. Sometimes, a well-designed propeller for one society has insufficient strength according to the other society. What is more, propellers designed according to the empirical formulas might be not optimal.

An example of typical propeller for bulk cargo ship is analysed in the paper (detailed description of the object is presented in the next chapter). For that propeller, the blade thickness (at the height of 0.25 relative radius) of the analysed propeller is determined on the basis of The American Bureau of Shipping (ABS) and Det Norske Veritas (DNV) rules. The blade thicknesses of the propeller were determined as $t_{0.25\,ABS}$ = 239.9 mm, $t_{0.25\,DNV}$ = 239.9 mm. Estimation based on the independent estimation (comparative measurements) gives values equal to *210 mm*. The strength of a lighter propeller has to be proven.

Shipyard designers used numerical calculations very rarely. FEM complication and availability, and difficulties with determining propeller loadings are the cause. Moreover, analysis methodology is unknown. The author developed a calculation methodology for the strength of the propeller working in nominal and emergency (e.g., during ship's crash stop) conditions. The analyses have been performed on the basis of Patran-Nastran software. Non-linear (geometrical and/or material) analyses can be handled by the software.

The finite element method implementation into strength analyses can lead to lighter propellers, low cost, and more efficiency. In some cases, the presented method is the only method for approving the design in given classification society. All main classification societies allow the usage of the finite element method but implemented in verified software.

1. Model for the methodology analyses

Determination of the optimal methodology has been performed on the basis of a typical propeller designed for a bulk cargo ship. The propulsion system is also typical where the propeller is driven by a relatively short shaft line and slow-speed main engine [4–6]. The FEM model of the power transmission system [7] (the crankshaft, the shaftline and the propeller) is presented in Fig. 1.

Fig. 1. FEM Model of the power transmission system

Nominal power of the 6th cylinder main engine is equal to 13330 kW; 87 rpm is a nominal revolutions. The propeller is made of nickel-aluminium bronze; their parameters are presented in Tab 1.

The propellers have a complicated geometry, which can lead to some difficulties during meshing procedure [3]. A geometric model of the propeller is shown in Fig. 2. The connection between propeller blade and hub is especially difficult from a numerical point of view and FEM. The FEM model was built with using 3-D, 8-node solid elements. The total number of the model's degree of freedom is equal to 86300. Generally, one node has three degree of freedom. The nodes of the FEM model of the whole propeller have been fully blocked at the surface lying on the propeller shaft. Separate analyses show [8] that the calculations can be limited to a single blade with special boundary conditions, when the nodes on the blade foot have been fully blocked.

Table 1. Parameters of the analysed propeller

Parameter name	Value
Diameter	7.8 m
The number of blades	5
Propeller pitch ratio	0.691
Expanded area blade ratio	0.600
Mass	30300 kg
The inertia of the propeller in air	341000 kgm^2
The specific mass of the propeller	7.6×10^3 kg/m^3
Tensile strength	640 N/mm^2
Yield strength	250 N/mm^2
Permissible stresses acc. to ABS for nominal conditions	59 N/mm^2
Permissible stresses acc. to ABS for emergency conditions	168 N/mm^2

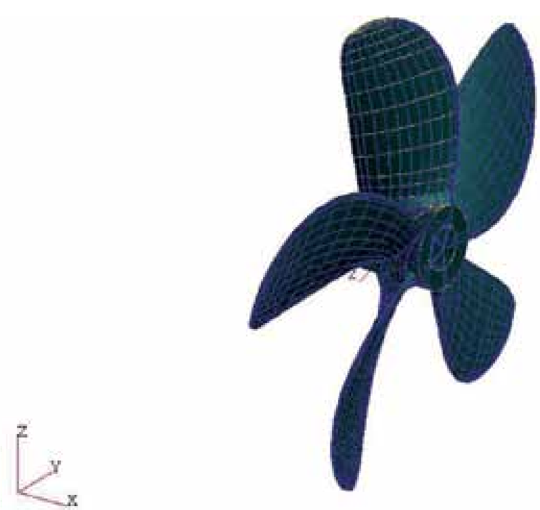

Fig. 2. Geometrical model of the propeller

Loadings determination is one of the main problems during numerical calculations. Generalized hydrodynamic forces and water pressures on the blade were determined by the separate software named UNCA (Unsteady Propeller Cavitation Analysis), authorship by J. Szantyr [9]. According to classification societies' rules [1, 2, 6], permissible stresses have to be checked in all (nominal and emergency) working conditions when the ship is going ahead and astern. If permissible stresses are unsatisfactory during full astern movement, propulsion system's revolutions should be limited to 70% of nominal revolutions. The following five different loading conditions were taken into account during calculations:

1. Steady-state working conditions in full ahead with nominal ME power,

2. Steady-state working conditions in full astern with 70% of nominal revolutions (61 rpm),
3. Steady-state working conditions in full astern with maximal astern ME power (67 rpm),
4. Transient working conditions in full astern with 70% of nominal revolutions (61 rpm), and
5. Transient working conditions in full astern with maximal astern ME power (67 rpm).

First and fifth loading conditions are the most dangerous for the analysed propeller. Generalized hydrodynamic forces for full ahead ship movement in a steady-state condition are shown in Fig. 3. The highest loads are acting when the blade is deviated from the vertical position of 86.4°. The pressure distribution on the pressure and suction side of the blade for that position is shown in Fig. 4.

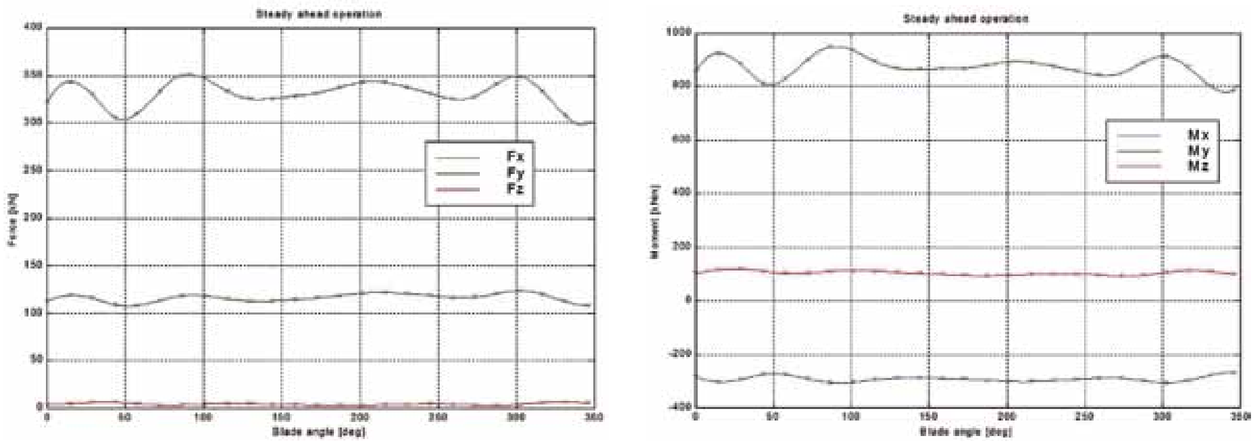

Fig. 3. Generalized hydrodynamic forces in steady-state and full ahead condition

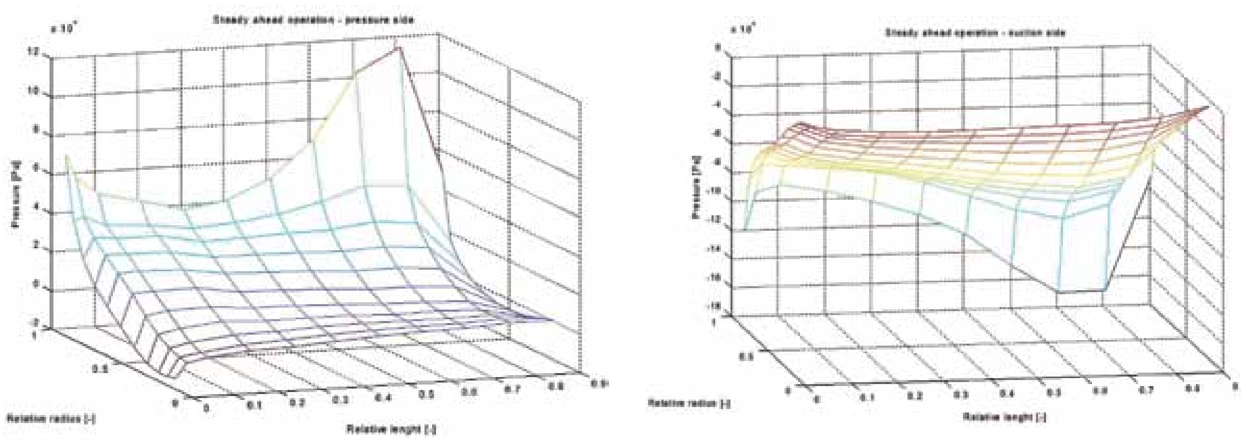

Fig. 4. The pressure distribution on the pressure and suction side in steady-state and full ahead condition when the blade is in 86.4° position

The next important problem during dynamic analyses of wetted marine structures is added water mass [10]. There are several formulas describing propeller inertia of the added water mass value [11]. The best one, in the author's opinion, has been derived on the basis of Parson's theory (the equation No. 1 and Tab. 2).

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

where:

J_H – inertia of entrained water [kgm²],
D – propeller diameter [m],
r – specific mass of sea water (usually 1025 kg/m³),
CJ_i – coefficients given in table 14,
A_e/A_0 – expanded area blade ratio,
P/D – propeller pitch ratio.

Table 2. 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

For the analysed propeller added water mass is taken into account by changing mass density. The original mass density of the bronze (7.6×10^3 kg/m^3) is increased to 15.36×10^3 kg/m^3. Taking account of added water mass is crucial during dynamic analyses like normal modes or transient response vibration analyses.

2. Natural vibrations

In the first step of the analyses, the natural frequencies of the propeller were checked. If the detuning value between main natural frequencies and frequencies of excitation forces is greater than 20%, the propeller can be analysed by a quasi-static solver. The influence of added water mass on the results of dynamic analyses is the next question [12, 13]. Should the propeller be modelled as a single blade (according to classification societies) with boundary conditions or as whole body (see Fig. 2) with blocked nodes on the inner part of the propeller hub? And the last question: What kind of matrix mass should be used? There are two types of matrix mass: simpler "lumped" (with non-zero elements only on matrix diagonal) and more accurate but numerically more complicated "coupled" matrix. The following variants of the calculations were performed:

1) Normal modes of propeller's single blade in the air,
2) Normal modes of propeller's single blade with added water mass,
3) Normal modes of whole propeller with blocked nodes on the inner part of the hub in the air,
4) Normal modes of whole propeller with blocked nodes on the inner part of the hub with added water mass,
5) Normal modes of whole free (without any boundary conditions) propeller in the air with usage of "lumped" mass matrix, and
6) Normal modes of whole free propeller in the air with usage of "coupled" mass matrix.

An example of first four normal modes of propeller's blade is presented in Fig. 5. Frequencies of natural vibrations for all calculation variants are presented in the Tab. 3.

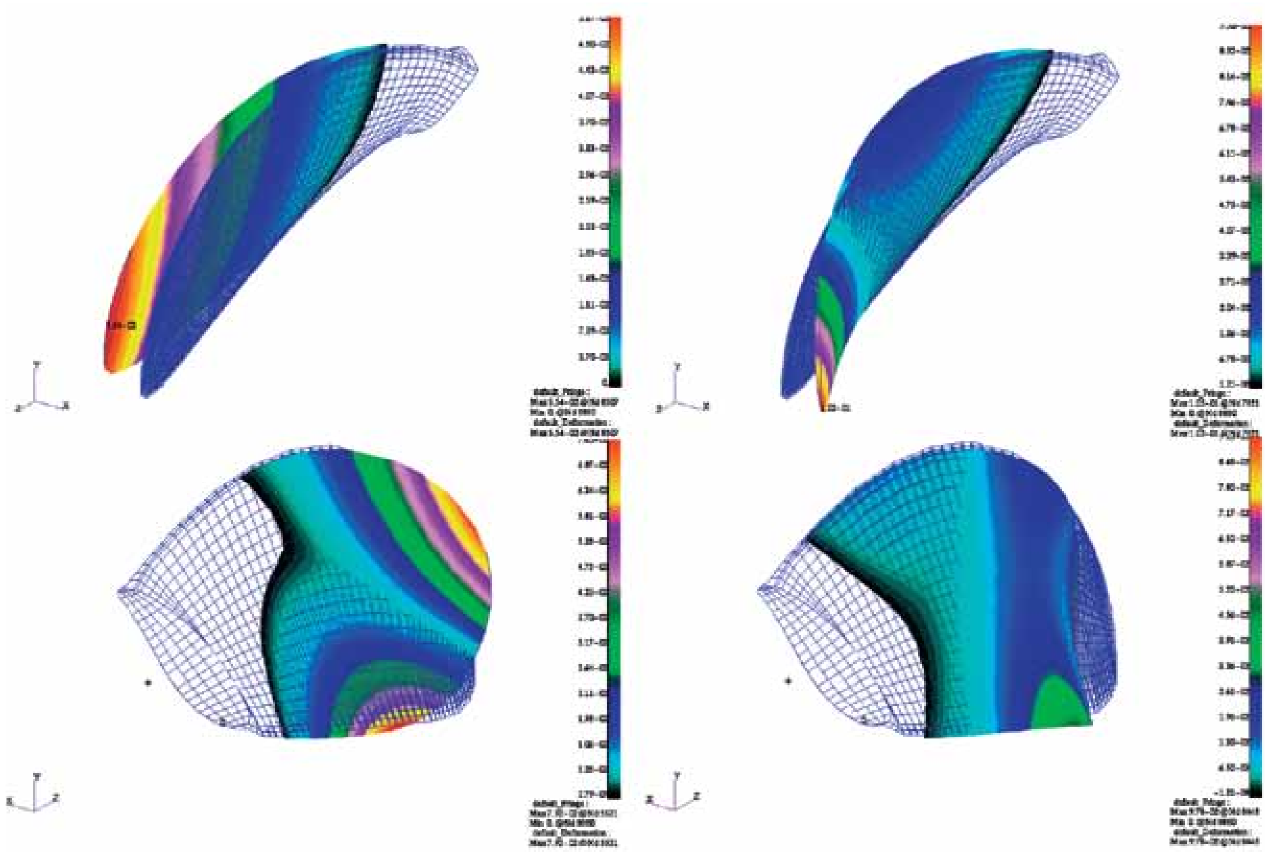

Fig. 5. First four normal modes of propeller's blade

Table 3. Natural frequencies of the propeller

Number of normal mode	Frequency [Hz]					
	Variant 1	*Variant 2*	*Variant 3*	*Variant 4*	*Variant 5*	*Variant 6*
1	17.51	12.32	17.36	12.21	17.17	17.18
2	47.03	33.08	46.20	32.50	45.72	45.86
3	57.22	40.25	56.06	39.43	55.38	55.70

The main frequency of the excitation forces derived from the main engine is equal to 8.7 Hz (87 rpm and 6 cylinders). The first excitation frequency derived from the propeller is equal to 7.25 Hz. The offset between natural and forced frequencies is greater than 40%. Therefore, the propeller can be calculated with the usage of the static analysis method – the assumption of quasi-static working conditions checks out.

The influence of added water mass on the results of dynamic analyses of the propeller is very big. The difference between the natural frequency of the propeller in air and in water is greater than 40% (variant 1–2 and 3 – 4). But, the natural forms (the shape of natural modes – eigenvectors) are nearly the same for both models.

Assumed boundary conditions do have a big influence on calculation results (Variants 1–3–5). The differences between natural frequencies determined on the basis of different models are in the range of 3% for all analysed normal modes. Classification societies' recommendation relating to the propeller model for that type of analyses is good – one single blade of the propeller with boundary conditions placed in the blade's foot is sufficient for the calculations. Moreover, the type of mass matrix does not have a big influence on calculation results (Variants 5–6). A simpler mass matrix (faster calculations) named „lumped" can be used during the presented analyses.

3. Static strength of the propeller

According to analyses presented in the previous chapters, only one blade of the propeller was analysed. The quasi-static loads (pressures on the suction and pressure side of the blade) were assumed on the basis of separate calculations (see Figs. 3 and 4). The analyses was performed for the two most dangerous working variants of the propeller: steady-state working conditions in full ahead with nominal power of main engine and transient working conditions in full astern with maximal astern power of main engine. Von-Misses stresses distributions for both analysed variants are presented in Figs. 6–9.

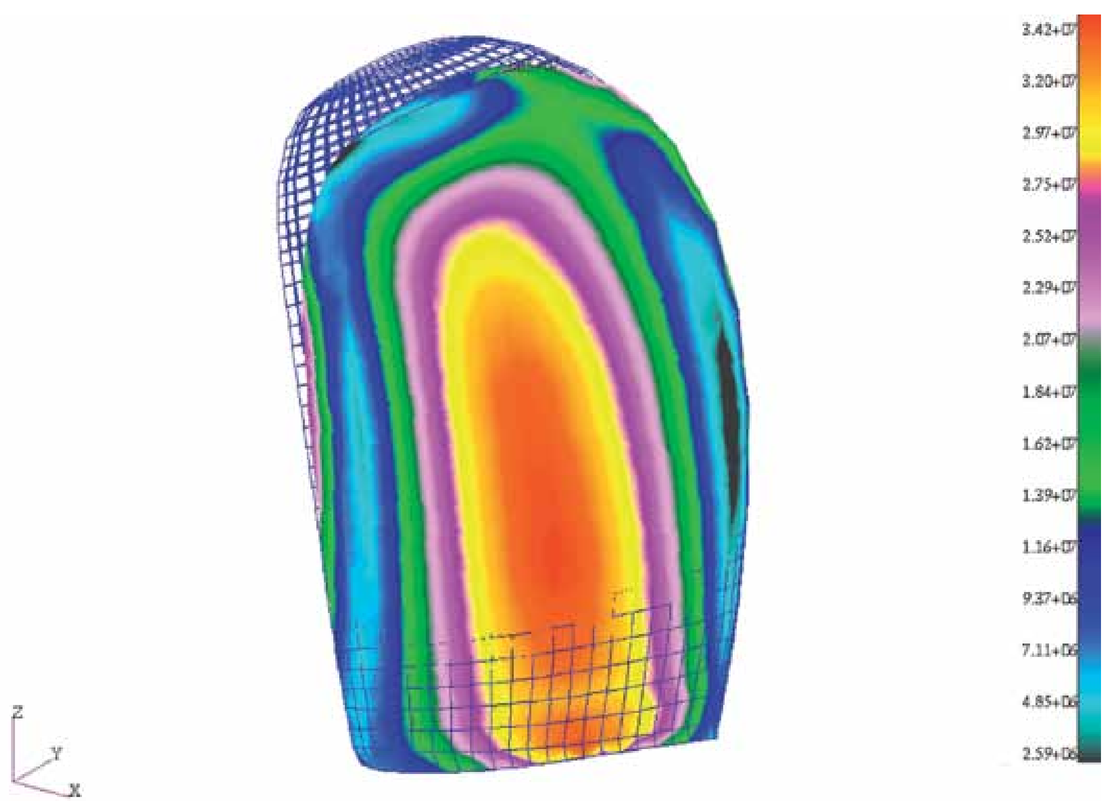

Fig. 6. Von-Misses stresses distribution for steady-state working conditions in full ahead – propeller's suction side

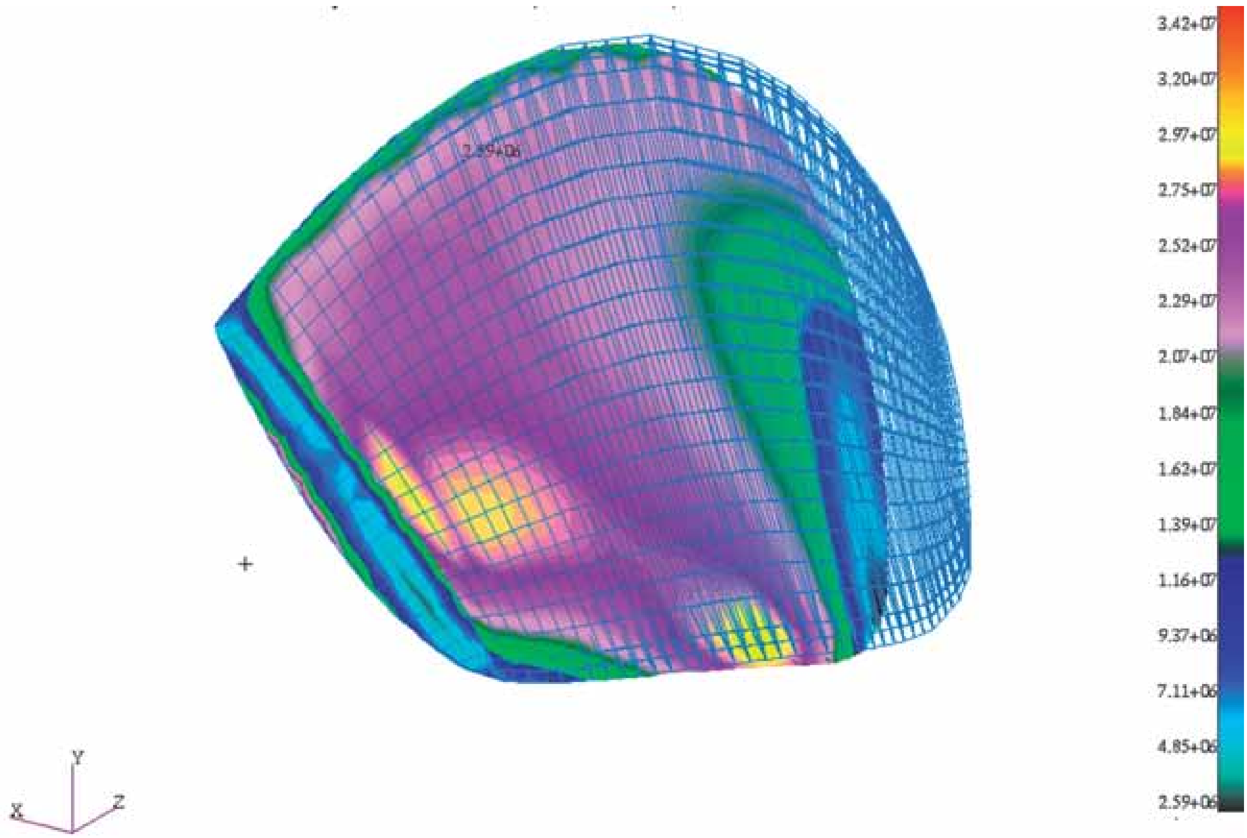

Fig. 7. Von-Misses stresses distribution for steady-state working conditions in full ahead – propeller's pressure side

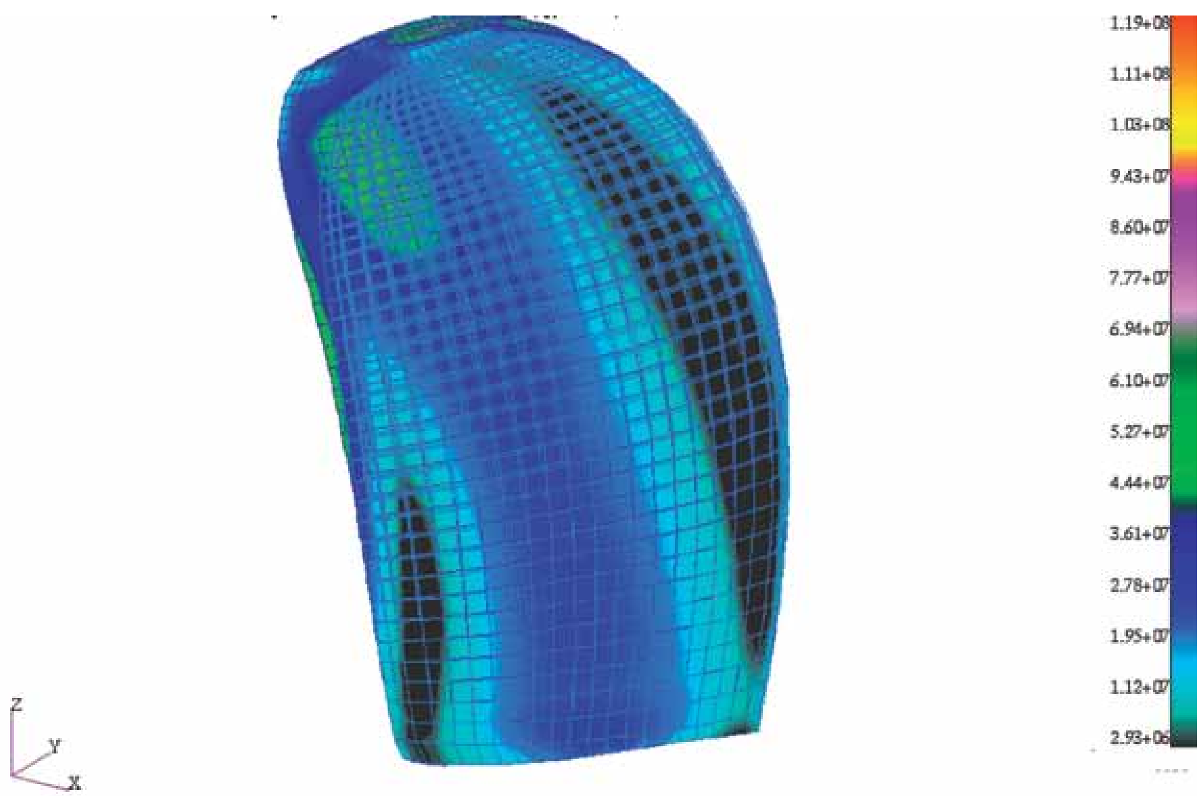

Fig. 8. Von-Misses stresses distribution for transient working conditions in full astern – propeller's suction side

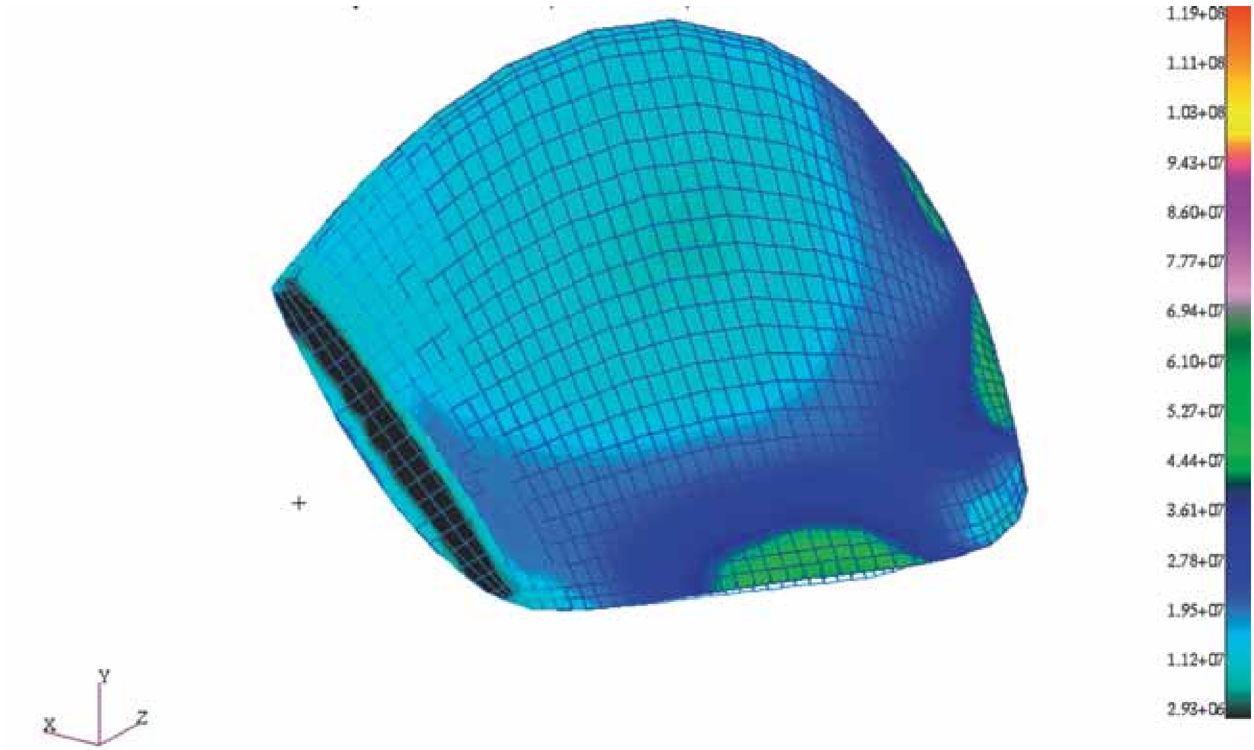

Fig. 9. Von-Misses stresses distribution for transient working conditions in full astern – propeller's pressure side

The most important is the steady-state working conditions of the propeller when the ship is running with full ahead command and nominal, maximal power of main engine. Maximal blade's deformation is equal to 18.2 mm. Maximal stress level is located at the height of 0.25 relative radius of the propeller and is equal to 36.5 MPa. Another region with a higher stress level is located at the height of 0.7 relative radius of the propeller and is equal to 33.4 MPa. It is compatible with classification societies' recommendations. The empirical formulas determine the blades thickness at the height of 0.20–0.25 relative radius for classical propellers and at the height of 0.70–0.75 relative radius for propellers with big skew-back.

Maximal blade's deformation and Von-Misses stress level for the propeller working in transient conditions, which is when the ship is running with full astern with maximal astern command and maximal power of main engine, is equal to 21.7 mm and 127.0 MPa. Another region with a raised stress level is placed at the height of 0.9 relative radius of the propeller and is equal to 52.6 MPa.

In both working conditions, the propeller strength is sufficient (The admissible stress level for steady-state working condition is equal to 59 MPa, and for transient working conditions, it is equal to 168 MPa). The design of the propeller is good even though the empirical formulas of some of classification societies said that the blade thickness should be a little bit greater. The blade thickness of the analysed propeller is equal to 210 mm (at the height of 0.25 relative radius); however, according to ABS empirical formula, it should be equal to 240 mm and 227 mm according to DNV. Numerical analyses based on the finite element method of the propeller can be very useful, because the optimisation might be efficient. After numerical calculations, the propeller mass saving is acceptable. 2400 kg bronze saving can be achieved in comparison to DNV empirical formula, and even 4300 kg saving according to ABS formulas.

Conclusions – main points of the methodology

The offset between the first frequencies of the excitation forces and the main natural frequencies is sufficient for treating the model as static. The propeller can be calculated using the static analysis method, because the assumption of quasi-static working conditions checks out. The influence of added water mass on the results of dynamic analyses of the propeller is very big, but the natural modes are nearly the same for all models. Assumed boundary conditions do not have a big influence on calculation results. Classification societies' recommendation relating to the propeller model for that type of analyses is good, i.e. one single blade of the propeller with boundary conditions placed in the blade's foot is sufficient for the calculations. The type of mass matrix does not have a big influence on calculation results. Simpler mass matrix named "lumped" can be used during this kind of analyses.

Reduced stress levels in the transient working conditions are much higher in comparison to stress levels in the steady-state working conditions, but permissible stresses are also different. The load (pressure distribution on the blade) determinations during transient working conditions are difficult and burdened with relatively large errors. Therefore, for preliminary calculations (optimisation of the propeller design), only steady-state working conditions of the propeller when the ship is running with full ahead command and nominal, maximal power of main engine may be used. The differences between permissible stress levels in both working conditions are coming from fatigue analyses. For a structure like a propeller, at least 100 million cycles should be taken into account during nominal working conditions. For nickel-aluminium bronze (the most popular material for the propellers), the quotient between static and fatigue permissible stress level is equal to 2.85. For other materials, that quotient is never less than 2.5. Therefore, propellers strength is usually determined by the nominal working conditions.

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