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# COMPARATIVE ANALYSIS OF THE CALCULATION METHODS OF THE MARINE PROPELLER'S BLADE THICKNESS

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***Abstract***

*Strength of the propellers with the skewback greater than 25º has to be numerically analysed according to marine classification societies. The finite element method (FEM) is advised for that kind of calculations. Classical and typical propellers (skewback < 25º) may be designed on the base 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 formula. Sometimes, well-designed propeller for one society has not enough strength according to the other society. What is more, propellers designed according to the empirical formulas might be not optimal. Comparative analysis of the marine propeller's blade strength has been described in the article. Calculations of the propeller's blade thickness have been done by two international classification societies' empirical formulas (ABS and DNV). The results have been compared with Finite Element Method calculations (NASTRAN program). The methodology of propeller static strength vibration analyses is presented. Numerical calculation methodology is based on solid-state mechanics with loadings determined by fluid mechanics calculations. Steady state and transient fluid flow of the propeller's working conditions were taken into account. In order to determine the optimal modelling method of the propeller several different numerical models were compared, including free model of whole propeller and single blade with boundary conditions placed in the foot. The propeller optimization was the main target of the analyses. Propeller blade thickness might be reduced after FEM method analysis - the propeller mass saving can be achieved.*

***Keywords****: marine propeller, strength, vibration, solid structure in fluid, FEM modelling methodology*

## 1. Introduction

The propeller is one of the main propulsion system elements, which decides about maritime reliability. A static and dynamic hydrodynamic pressure field is generated during the propeller running (rotation). A wake field is the main parameter determining these pressures. The wake is the region of disturbed flow (often turbulent) downstream of a solid body (propeller) moving through a fluid, caused by the flow of the fluid around the body (ship hull). Therefore, the wake field is fixed by the ship hull and a propeller mating. Therefore, the propeller loadings have to be individually determined for each propulsion system.

The bending stresses of propeller blades are usually dominant. Therefore, the blade footing (~0.2 of relative propeller radius) thickness has to be determined during strength analysis. The thickness of the blade tip (~0.7 of relative propeller radius) should also be checked. The stress level has to be determined for each of the working conditions. In particular, the propeller strength at a nominal running condition (the ship going ahead at full sea speed order), as well as in the worst unsteady running condition (usually during a crash stop manoeuvre – full ahead to full astern order), have to be check. Fatigue strength should be taken into account only for a nominal running condition. It is taken into consideration by the proper permissible stresses assumption ($10^8$ load cycles should be assumed). Most of the propeller strength analysis methods are limited to quasi-static analysis. However, the dynamic behaviour of the propeller blade should also be checked in the author's opinion.

The marine classification societies require detailed strength calculations of ship propellers in the case where the skewback angle is greater than 25º [11, 12]. In the case of a smaller skewback angle, the societies recommend empirical formulas. Such a determination method for propeller geometrical parameters contains some drawbacks: sometimes a given propeller is approved by one classification society, but for another one its blade thickness is too small. Moreover, the empirical formulas do not always give an optimum blade thickness. Sometimes, the designers try to avoid a bigger skewback angle because of expected, relatively bigger problems with a propeller's approval.

In today's shipbuilding practice, propeller strength analysis, using FEM software is performed rarely [7]. In the work in question, Nastran software was used which makes it possible to take into account both material and geometrical non-linearity. Implementation of FEM – based software for propeller strength analysis, leads to the lowering their weight and production costs by enabling the selection of an optimum blade thickness. In some cases, from the side of classification societies, it is the only way of obtaining approval of an optimum propeller design.

## 2. Preliminary calculations with the methods of classification societies

The analyses have been performed for typical propeller [5, 6, 10]. A large, typical propeller applied on a tanker of 90000 dwt, was selected for example analysis. The propeller is directly driven by a Sulzer 6 RTA-76 type engine of power: 13,330 kW and nominal speed: 87 rpm. The propellers main particulars are as follows:

– diameter: 7.80 m,
– number of blades: 5,
– pitch ratio: 0.691,
– blade area ratio: 0.600,
– mass: 30,300 kg,
– material: Ni-Al bronze,
– tensile strength: 640 N/mm$^2$,
– yield point: 250 N/mm$^2$,
– permissible stress for nominal work [10]: 59 N/mm$^2$,
– permissible stress for emergency work [10]: 168 N/mm$^2$.

The classification societies empirical formulas usually determine the propeller blade footing thickness (relative radius = 0.25) and sometimes the propeller blade tip thickness (relative radius = 0.7). These formulas have quite a complicated form – there are several constants, dependent on the propeller geometry, propulsion power and revolution, ship speed and thrust, material property etc. Each classification society has its own formulas; therefore, the final results of the blade thickness might be different. The propeller is analysed according to the empirical rules given by two classification societies: the Det Norske Veritas (Norway) and the American Bureau of Shipping.

The main formula, given by the Det Norske Veritas and determining propeller blade thickness, is presented in Eq. (1):

$$t_r = C_1 \cdot \sqrt{\frac{D \cdot (U_2 \cdot K_3 + m_t \cdot K_1)}{Z \cdot c_r \cdot (U_1 - U_2 \cdot S_R)}} \quad [\mathrm{mm}], \tag{1}$$

where:

$t_r$ – blade thickness on the relative radius = r,
$C_1$ – constant value dependant on the analysed blade thickness at the radius in question,
$D$ – diameter of the propeller,
$U_1$ – reversed stress (material constant),
$U_2$ – influence of mean stress on fatigue (material constant),

$K_3$ – mean load dependant on propeller torque and thrust,
$S_R$ – mean load dependant on propeller rake and revolutions,
$m_t K_1$ – cyclic load,
$Z$ – number of blades,
$c_r$ – width of expanded cylindrical section at the radius in question.

The analysed propeller minimum blade thickness at the relative radius = 0.25, determined by Eq. (1), is as follows:

$$t_{0.25\ DNV} = 226.5 \text{ mm}.$$

The main formula, given by the American Bureau of Shipping and determining propeller blade thickness, is presented in Eq. (2):

$$t_{0.25} = S\left[K_1 \cdot \sqrt{\frac{A \cdot H}{C_n \cdot C \cdot R \cdot N}} \pm \left(\frac{C_s}{C_n}\right) \cdot \left(\frac{B \cdot K}{4 \cdot C}\right)\right] \ [\text{mm}], \tag{2}$$

where:
$S$ – constant value dependant on propeller diameter,
$K_1$ – equal to 337 for fixed-pitch propellers,
$A$ – constant value dependant on propeller pitch distribution,
$B$ – constant value dependant on number of blades, expanded blade area, material constant, propeller diameter and revolutions,
$C$ – constant value depended on propeller pitch, width, material constant and B,
$H$ – power at rated speed,
$R$ – propeller revolutions at rated speed,
$N$ – number of blades,
$C_s$ – section area coefficient at the analysed relative radius,
$C_n$ – section modulus coefficient at the analysed relative radius,
$K$ – rake of propeller blade multiplied by propeller radius.

The analysed propeller minimum blade thickness at the relative radius = 0.25, determined by Eq. (2) is as follows:

$$t_{0.25\ ABS} = 239.9 \text{ mm}.$$

The author's assumption is to reduce the blade thickness by around 10%. If the blade thickness equal to 210 mm has satisfactory strength. If yes, in the case of the DNV, the saving can achieve 2400 kg and in the case of the ABS, the saving can achieve 4300 kg of bronze. Apart from lower costs, a propeller with smaller mass has better dynamic characteristics. The assumed propeller's strength has to be checked by finite element method (FEM) calculations.

## 3. Finite element method analyses

The author's experience [4] indicates that relatively large difficulties appear in analysing ship propellers. The main reason is the complicated screw form – mainly the large curvature of its surface. The hub-blade connection region especially creates many problems. Highly deformed and degenerated solid finite elements introduce computational difficulties. Within this work, many attempts were made to model the screw appropriately, and on this basis, an optimum modelling method was finally selected.

The FEM structural calculations of the propeller were performed by using Nastran software. A geometrical model of the propeller, as well as analysis results, was elaborated by using Patran software. The finite element structural model of the propeller is presented in Fig. 1. It was formed of 8-nodes, 3-D finite elements and had 86320 degrees of freedom.

At the beginning, several calculation versions of the fundamental frequencies and modes of the natural vibrations of the propeller were realised in order to determine a degree of detuning of natural vibration characteristics from the excitation frequencies. The aim was to check if there was a hazard of excessive dynamic magnification of the propeller blade deformations (stress) under operational loads – if the propeller natural frequencies are sufficiently detuned from the fundamental excitation frequencies. In the case of at least 20% differences, it would be possible to apply static analysis only. Classification societies' recommendation relating to the propeller FEM model said that only one single blade of the propeller with boundary conditions placed in the blade's foot is sufficient for the analyses. It is another problem worth for analysing. The water added mass was taken into account by increasing the density of the propeller material, in such a way as to obtain a final propeller mass equal to the sum of the propeller mass in the air and the added mass of water [1]. The water added mass was estimated by the R. Dien and H. Schwanecke formula [2]. As a result, the propeller material density was enlarged from $7.6\times10^3$ kg/m$^3$ to $15.36\times10^3$ kg/m$^3$. The water added mass [3, 8] can be modelled by "wetting" finite elements, but this method is usually not well worked out in the commercial FEM software. It might be numerical problems with the analysis or the calculation time which may even increase a number of dozen times. The calculation accuracy might be even worse in comparison to the one using empirical formulas. To sum up, three main cases was analysed: single blade mounted at the foot; full propeller with fixed nodes on the propeller shaft interference surface and full "free" (without any boundary conditions) propeller. The natural vibration frequencies, calculated for different calculation cases, are presented in Tab. 1. The natural vibration modes of the propeller with and without accounting for added water mass were similar. The first three natural vibration modes of the propeller "blocked" on the propeller shaft are presented in Fig. 2.

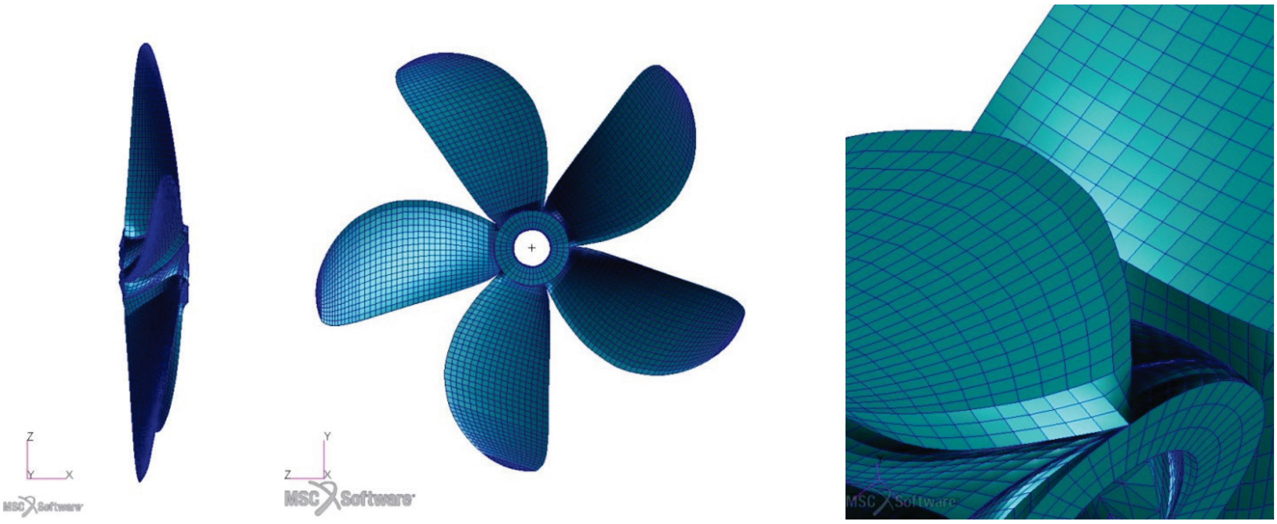


*Fig. 1. The FEM structural model of the propeller*


*Fig. 2. First three natural modes of the propeller*

*Tab. 1. Natural vibration frequencies of the considered propeller*

| Number of vibration mode | Natural frequency [Hz] | | |
|---|---|---|---|
| | Single blade | Propeller with shaft | "Free" propeller |
| 1 | 12.32 | 12.21 | 12.08 |
| 2 | 33.08 | 32.50 | 32.16 |
| 3 | 40.25 | 39.43 | 38.95 |

Assumed boundary conditions have not big influence on calculations results. The differences between natural frequencies determined on the base of different models are on the range of 3% for all analysed normal modes. In shipbuilding practice, the 50 Hz upper limit of the considered vibration frequencies is assumed. Hence, in the case of propeller vibration analysis, limited to that of a single blade only, the introduced error was not greater than 3%. In the case of the analysis of a complete propeller with "blocked" FE nodes on the hub-shaft surface, the greatest error was 1%. Therefore, the application of the calculation model, complying with that case is recommended; however, the single blade structural analysis could be justified if a rush analysis is necessary.

The relative detuning on the main natural frequencies on the immersed propeller are 29% and 41% respectively, as the fundamental frequency on excitations, due to the main engine operation of the tanker in question amounted to 8.7 Hz, and that of the pressure pulsations around the propeller – to 7.25 Hz. The detuning is sufficiently large to calculate the deformations and stresses of the propeller using static analysis only, and neglecting dynamic considerations. Performing such a check is recommended for every propeller that is to be analysed for the first time.

For static, strength analysis, a basic linear solver was applied and, to check the correctness of its results, another solver was also utilised, which accounted for non-linear effects during the analysis. The computational options of large deformations, deformation following up loading and material non-linearity were used. The screw blade was loaded by a hydrostatic pressure equivalent to the maximum water pressure, which occurs during the operation of the ship propulsion system.

To determine load the distribution over the propeller UNCA software (Unsteady Propeller Cavitation Analysis) was used [9]. This software makes it possible to calculate the generalised hydrodynamic forces as well as the induced pressure distribution over the propeller, operating within a non-uniform velocity field of water behind the ship hull. Time-dependant cavitation phenomena and propeller initial geometry deformation are also accounted for.

According to the classification society's rules, there is no strict necessity to fulfil a propeller strength in conditions when a ship is running astern with full main engine power. If the permissible stresses of the propeller are exceeded, then the astern engine speed has to be limited to 70% on the nominal revolutions. Generally, the propeller's five loading cases should be taken into account:

– full ahead ship speed (with maximum continuous propulsion power) in constant operating conditions,
– full astern ship speed (with maximum continuous propulsion power) in constant operating conditions,
– full astern ship speed in unsteady operating conditions (during the crash stop manoeuvre – fast full ahead to full astern thrust changing),
– astern ship running with 70% of maximum propulsion power, in constant operating conditions,
– astern ship running with 70% of maximum propulsion power, in unsteady operating conditions.

The two most hazardous cases of hydrodynamic pressure loading on the propeller blades were considered. The first one occurs in steady-state working conditions, i.e. the maximum continuous rating of the propulsion system, working ahead at the service speed of the ship. The other case was the maximum (critical) load over the propeller, which occurs during its operation astern at the maximum continuous astern rating of the propulsion system (at 70 rpm) and at null-speed of the ship (unsteady operating condition). A maximal loading in the nominal, steady-state working conditions occurs when the propeller blade is rotated at 86.4°. Loading on both sides of the propeller (pressure side and suction side) has been taken into account.

Several variants of the propeller static analysis were performed in order to find out the best (easy and accurate) analysis method as well as to determine the propeller strength. The following cases of static calculations of the propeller in question were performed:

– *case 1:* linear structural analysis of a single propeller blade in nominal working conditions,
– *case 2:* non-linear structural analysis of a single propeller blade in nominal working conditions (large displacements, deformation-following-up loads and non-linear properties of material),
– *case 3:* linear structural analysis of the complete propeller in nominal working conditions,
– *case 4:* linear structural analysis of the complete propeller in nominal working conditions, taking into account pressure loads from propeller boss-shaft interference,
– *case 5:* linear structural analysis of the complete propeller in emergency working conditions.

The maximum values of blade tip deformation and reduced (the Huber – Von Misses) stress at the blade base for all calculation cases, are presented in Tab. 2. The calculated deformations and stresses of a complete propeller in nominal working conditions (case 3), are presented in Fig. 3. The results of the calculation for the propeller working in emergency conditions (case 5), are shown in Fig. 4.

*Tab. 2. Results of the static structural analysis of the propeller*

| | case 1 | case 2 | case 3 | case 4 | case 5 |
|---|---|---|---|---|---|
| Max. deformation [mm] | 27.7 | 26.7 | 28.7 | 29.7 | 53.0 |
| Max. reduced stress [MPa] | 83.1 | 78.6 | 83.2 | 590 | 126 |

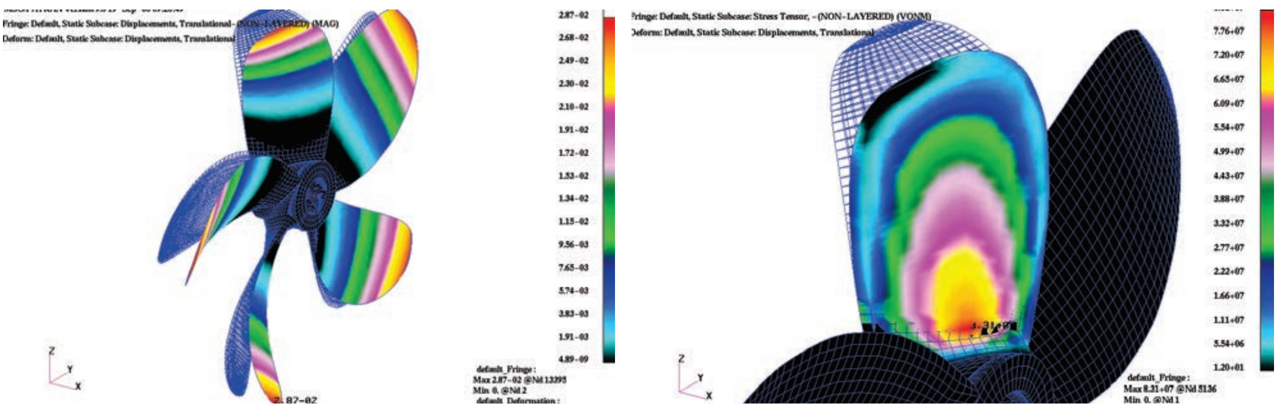

*Fig. 3. The propellers deformations and Von-Misses stresses in nominal working conditions*

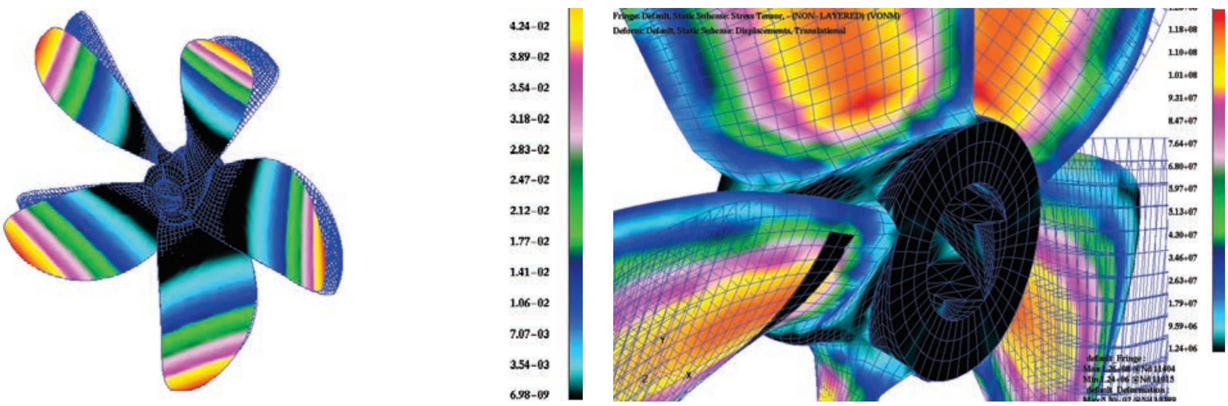

*Fig. 4. The propellers deformations and Von-Misses stresses in emergency conditions*

In the case of simplified, linear structural analysis, the relative estimation error (case 1 to case 2) of the propeller blade deformation amounted to 3.7%, and that of the reduced stresses, to 5.7%. This means that when using linear analysis, both the deformation and stress values are over--estimated. As the cost of carrying out non-linear analysis is many times greater than that of a linear one, it is recommended to use this only in cases when the calculated stresses are close to those permissible.

From the analysis of the calculation results of cases 3 and 4, it can be concluded that the deformation and stresses of the propeller blade and those of the propeller boss, are independent of each other. Hence, both elements can and should be (to obtain clear-cut results) analysed separately. In addition, the strength of the propeller boss should be estimated by means of a more exact method – as boss-shaft interference is a typical contact problem.

In the case of emergency operation of the propeller, the reduced stresses were over 50% greater as those calculated for the nominal working conditions of the ship propulsion system in question. For Ni-Al bronze, the material applied for the considered propeller, the ratio of the permissible stress in emergency conditions, and that in nominal conditions accounting for fatigue strength, is equal to 2.85; for other materials it is not smaller than 2.5. Hence, it can be concluded that the nominal working conditions are decisive for propeller strength.

## 4. Conclusions

The offset between first frequencies of the excitation forces and main natural frequencies is sufficient for treating the model as a static. The propeller can be calculated with usage of static analysis method - assumption of quasi-static working conditions is check out. Influence of added water mass on the results of dynamic analyses of the propeller is very big. However, the natural modes are nearly the same for all models. Assumed boundary conditions have not big influence on calculations results. 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 propeller strength estimation. As the cost of carrying out non--linear analysis is many times greater than that of a linear one, it is recommended to use this only in cases when the calculated stresses are close to those permissible.

Reduced stresses level in the transient working conditions is much higher in comparison to stresses level in the steady-state working conditions but also permissible stresses are different. The loads (pressure distribution on the blade) determination during transient working condition is difficult and burdened with relative big error. 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 the both working conditions are coming from fatigue analyses. For structure like propeller, at least 100 million cycles should be taken into account during nominal working condition. Therefore, usually propellers strength is determined by the nominal working condition.

Numerical analyses based on finite element method of the propeller can be very useful – 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. What is more, the dynamic characteristics of the propeller can be checked during analyses based on FEM.

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