LECH MURAWSKI, D.Sc.,M.E.
Ship Design and Research Centre, Gdańsk

A FEM – based procedure for strength calculation of ship propellers and analysis of its results

SUMMARY

This paper concerns a new strength calculation method of ship propellers. Within its scope a propeller modelling technique with the use of a CAD pre-processor (PATRAN), calculation procedure based on Finite Element Method software (NASTRAN), as well as a way of performing numerical calculations, was developed. Results of example propeller calculations for a tanker are also attached.

INTRODUCTION

The ship classification institutions require detail strength calculations of ship propellers in the case if the maximum skew-back angle of their blade tip is greater than 25°.For such calculation analysis it is recommended to apply an approved, FEM-based software [1].In the case of smaller skew-back angles empirical formulas for determination of propeller blade thickness are also permitted. Such determination method of propeller parameters contains many drawbacks : it sometimes happens that a given propeller is approved by one classification society, but for another one its blade thickness is too small; moreover, the empirical formulas not always give an optimum blade thickness.

In the today shipbuilding practice the propeller strength calculations by using FEM software are performed very rarely [5]. In the work in question NASTRAN software was used which makes it possible to take into account both material and geometrical non-linearity [3]. Implementation of FEM – based software to propeller strength calculations leads to lowering their weight and production cost by enabling to select an optimum blade thickness. In some cases it is the only way of obtaining approval of an optimum propeller design from the side of a classification society.

CALCULATION MODEL

A large, typical propeller applied on a large tanker, was selected for an example analysis. The main particulars of the ship propulsion system and the propeller itself are as follows :

Main engine

Type	6 RTA 76
Power	13 330 kW
Speed	87 rpm

Propeller

Diameter	7.80 m
Number of blades	5
Pitch ratio	0.691
Blade area ratio	0.600
Material	Ni-Al bronze
Mass	30 300 kg
Inertia moment in air	341 000 kgm^2
Density of material	$7.6 \cdot 10^3$ kg/m^3
Tensile strength	640 N/mm^2
Yield point	250 N/mm^2
Permissible stress, nominal work [1]	59 N/mm^2
Permissible stress, emergency work [1]	168 N/mm^2

The gained experience indicates that relatively large difficulties appear in analyzing the ship propellers [4]. The main reason is a very complicated screw form, mainly large curvature of its surface. Many problems are created specially by the boss-blade interpenetration region . Highly deformed and degenerated, solid finite elements introduce the computational difficulties.

Within this work many attempts were made to appropriately model the screw, and on this basis the optimum modelling method was finally selected.

Preparation of the model and numerical calculations were carried out by means of a Indygo Silicon Graphic work station. Geometrical model of the propeller for structural calculations as well as their results were elaborated by using Patran software. FE structural model of the propeller is presented in Fig.1. It was formed of 8-node, 3-D finite elements and had 86 320 dof. For the calculations at least 58.5 MB of operating memory and 1822.3 MB of free space on hard disc was required.

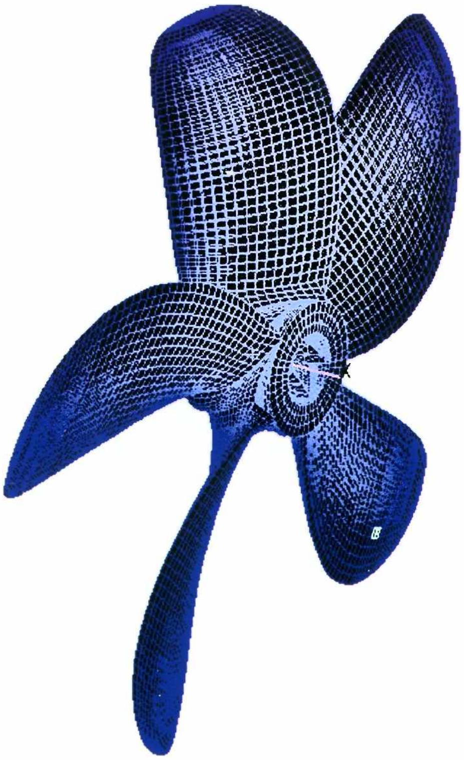

Fig.1. *FE structural model of the propeller*

FEM CALCULATION PROCEDURE

FEM structural calculations of the propeller were performed by using Nastran software [7]. At the beginning several calculation versions of the fundamental frequencies and modes of the natural vibrations of the propeller were realized in order to determine a degree of detuning of natural vibration characteristics of the propeller from the excitation frequencies, i.e. to check if there is any hazard of excessive dynamic magnification of the propeller blade deformations (stresses) under operational loads. For the static analysis the basic linear solver was applied and, to check correctness of its results, another solver was also utilized, which accounted for non-linear effects during the analysis. Also, computational options of large deformations, deformation-following-up loading and material non-linearity were used. The screw blade was loaded by the hydrostatic pressure equivalent to the maximum water pressure which occurs during operation of the ship's propulsion system.

To determine load distribution over the propeller the UNCA 93 software (Unsteady Propeller Cavitation Analysis) was used [6]. The software makes it possible to calculate the generalized hydrodynamic forces as well as the induced pressure distribution over the propeller operating within the non-uniform velocity field of water behind the ship hull. Time-dependent cavitation phenomena and propeller initial geometry deformation are also accounted for.

Two most hazardous cases of hydrostatic pressure loading on the propeller blade were considered. The first of them occurs in steady-state working conditions, i.e. the maximum continuous rating of the propulsion system working ahead at 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.

RESULTS OF DYNAMIC ANALYSIS

At first, natural vibrations of the propeller were calculated to assess if their frequencies were sufficiently detuned from the fundamental excitation frequencies. In the case of at least 20% differencies it would be possible to apply the static analysis only.

The following calculation versions of the propeller natural vibrations were performed :

Case 1d : Natural vibrations of the single blade (fixed at its base) in air.
Case 2d : As in Case 1, but with accounting for added mass of water.
Case 3d : Natural vibrations of the entire screw in air, at fixed FE nodes on the propeller-shaft interference surface.
Case 4d : As in Case 3, but with accounting for added mass of water.
Case 5d : Natural vibrations of the free propeller (without any boundary constraints) whose mass was modelled by means of the standard „lumped" mass matrix.
Case 6d : As in Case 5, but with modelling the propeller's mass by using the „coupled" mass matrix.

The added mass of water was taken into account by increasing the density of the propeller material in such a way as to obtain the final propeller mass equal to the sum of the real propeller mass in air and the added mass of water calculated in accordance with Schwanecke's formula [2].

The formula for the added mass M_H at axial displacements is as follows :

$$M_H = \frac{\rho \cdot D^3 \cdot \pi}{z} \cdot \left(\frac{A_E}{A_0}\right)^2 0.2812$$

where :

ρ	–	sea water density [kg/m³]
D	–	propeller diameter [m]
z	–	number of blades
A_E/A_0	–	propeller area ratio.

In result, the propeller material density was enlarged from $7.6 \cdot 10^3$ kg/m³ to $15.36 \cdot 10^3$ kg/m³. Nastran sofware is not provided with a programmed ability of „wetting" the solid finite elements. Therefore an attempt was made by this author to perform calculations of the propeller with weightless plates of low stiffness, artificially fitted on its surface, and to make them „wet". In this case the time of computation increased about 50 times (!) which exceeded operational capacity of the computers in hands. The natural vibration frequencies calculated for all calculation cases are presented in Tab.1. The natural vibration modes of the propeller with and without accounting for added mass were identical. Three first natural vibration modes of the complete propeller „blocked" on the propeller shaft (Case 3) are presented in Fig.2,3 and 4.

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

Vibration mode number	Frequency [Hz]					
	Case 1	*Case 2*	*Case 3*	*Case 4*	*Case 5*	*Case 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

In Tab.2. the relative accuracy errors of determining the natural vibration frequencies are presented, expressed in percentage of the values calculated for Case 6 (deemed the most correct way of calculation).

Tab.2. *Relative accuracy errors of determining the natural vibration frequencies*

Vibration mode number	Relative accuracy error [%]			
	Case 1	*Case 3*	*Case 5*	*Case 6*
1	1.92	1.05	-0.06	0
2	2.55	0.74	-0.31	0
3	2.73	0.65	-0.57	0

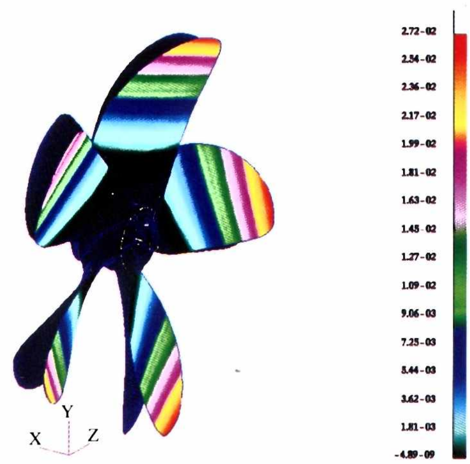

Fig.2. 1st natural vibration mode of the complete propeller (Case 3)

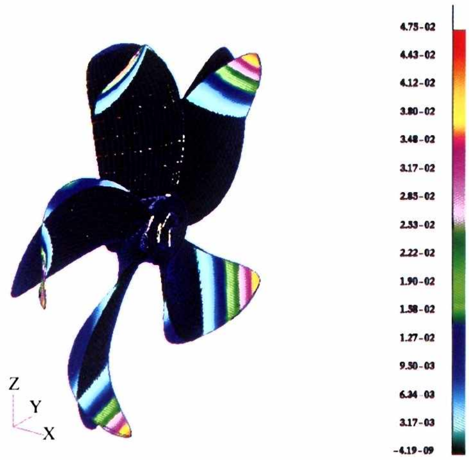

Fig.3. 2nd natural vibration mode of the complete propeller (Case 3)

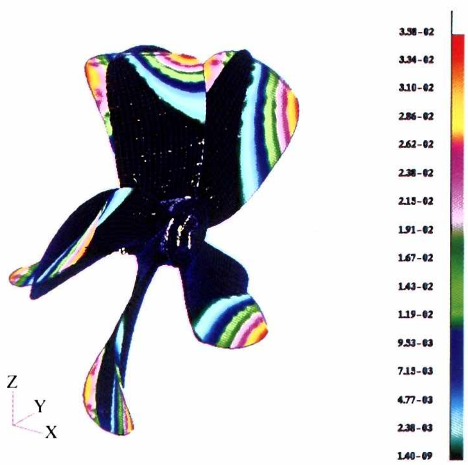

Fig.4. 3rd natural vibration mode of the complete propeller (Case 3)

RESULTS OF STATIC ANALYSIS

The following cases of static calculations of the propeller in question were performed :

Case 1s. Linear structural analysis of a single propeller blade in nominal working conditions (The similar model as in the Case 1d).
Case 2s. 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 3s. Linear structural analysis of the complete propeller in nominal working conditions (The similar model as in the Case 3d).
Case 4s. Linear structural analysis of the complete propeller in nominal working conditions with taking into account pressure loads from propeller boss-shaft interference.
Case 5s. Linear structural analysis of the complete propeller in emergency working conditions (The similar model as in the Case 3d).

The maximum values of the blade tip deformation and reduced (Huber – von Mises) stress at the blade base for all calculation cases are presented in Tab. 3.

In Fig.5 distributions are illustrated of the calculated deformations and stresses of a complete propeller in nominal working conditions (Case 3s), and in Fig.6 – the principal stress distribution within the propeller boss, calculated with accounting for pressure loads from propeller boss-shaft interference (Case 4s). Results of calculation of the propeller working in emergency conditions (Case 5s) are shown in Fig.7.

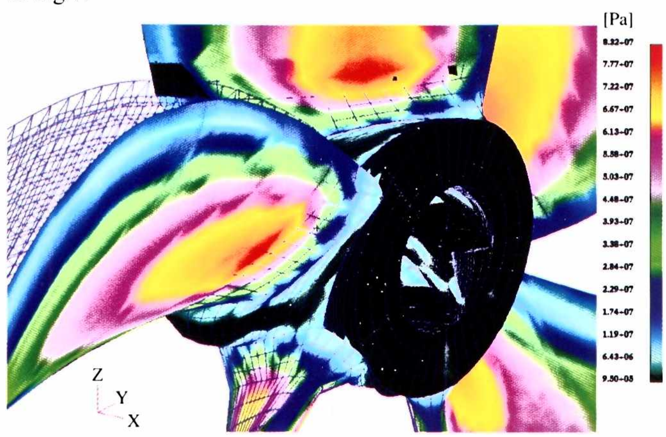

Fig.5. The reduced (Huber–von Mises) stresses of the propeller under nominal loading

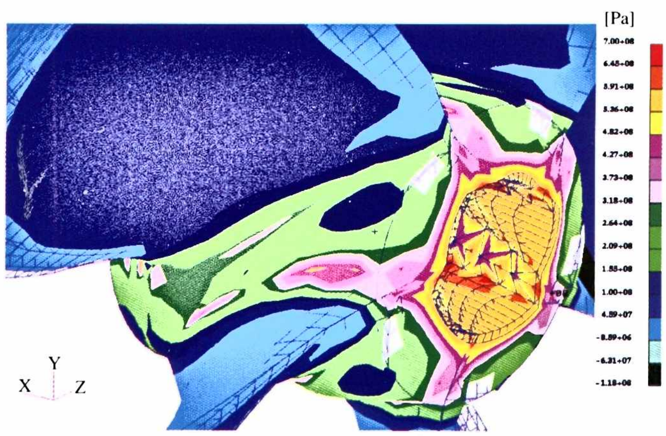

Fig.6. The principal stresses in the propeller boss under nominal loading with accounted for boss- shaft interference

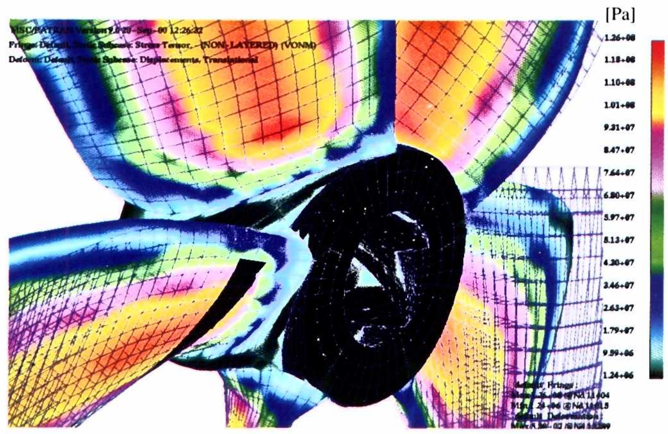

Fig.7. The reduced (Huber–von Mises) stresses of the propeller under emergency loading

Tab.3. Results of the static structural analysis of the propeller

	Case 1s	Case 2s	Case 3s	Case 4s	Case 5s
Max. deformation [mm]	27.7	26.7	28.7	29.7	53.0
Max. reduced stress [MPa]	83.1	78.6	83.2	~645	126

CONCLUSIONS AND RECOMMENDATIONS

- The relative detuning of the the main natural frequencies of the immersed propeller were 29% and 41%, respectively, as the fundamental frequency of 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 by using the static analysis only, and neglecting dynamic considerations. Performing such check is recommended for every propeller to be analyzed first time.
- In shipbuilding practice the 50 Hz upper limit of the considered vibration frequencies is assumed. Hence in the case of the propeller vibration analysis limited to that of single blade only, the introduced error was not greater than 3% (see Tab.2). In the case of the analysis of a complete propeller of the „blocked " FE nodes on the boss-shaft surface, with using the simplified mass matrix (Case 3d) the greatest error was 1%. Therefore application of the calculation model complying with the Case 3d is recommended; however the single-blade structural analysis (Case 2d) could be justified if a rush analysis is necessary.
- In the case of emergency operation of the propeller the reduced stresses were by over 50% greater than those calculated for 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 the emergency conditions and that in the nominal conditions with 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 of the propeller strength.

  Determination of the pressure distribution on the propeller during its emergency operation is difficult due to non-stationary character of the involved phenomena. Therefore application of a simplified method to estimate such distribution seems to be justified.
- In the case of the simplified, linear structural analysis the relative estimation error (Case 1s to Case 2s) of the propeller blade deformations amounted to 3.74%, and that of the reduced stresses to 5.73% (see Tab.3). It means that when using the linear analysis, both the deformation and stress values are over-estimated. As the cost of carrying out the non-linear analysis is many times greater than that of linear one it is recommended to use it only in the case when the calulated stresses are close to those permissible.
- From the analysis of the calculation results of the Case 3s and 4s it can be concluded that the deformations and stresses of the propeller blade and those of the propeller boss are independent of each other. Hence both the elements can and should (to obtain clear-cut results) be analyzed separately. And, strength of the propeller boss should be estimated by means of a more exact method – as the boss-shaft interference is a typical contact problem.

Appraised by Tadeusz Koronowicz, Prof.,D.Sc.,N.A.

BIBLIOGRAPHY

1. ABS Rules of Ships. 1998
2. Dien R., Schwanecke H.: *Die propellerbedingte Wechselwirkung zwischen Schiff und Maschine-Teil 2*. Motortechnishe, Vol.34, No 11. 1973
3. MSC/Nastran : *Dynamic Analysis*, NAS 102; *Static Analysis*, NAS 101; *Nonlinear Analysis* NAS 103.
4. Murawski L.: *Strength analysis of the ship propeller in steady state and emergency working conditions*. III Conference of MSC Software Users. Gdańsk 1998
5. Reed F. E. , Bassett N.: *Further studies of propeller-exited vibrations on Great Lakes cargo ship*. SNAME Transactions, Vol. 97. 1989
6. Szantyr J.: *The lifting surface program for hydrodynamic analysis of marine propellers*. Ship Design&Research Centre report no RH-93/R-064. Gdańsk 1993
7. Weaver W., Johnston P.: *Finite Elements for Structural Analysis*. Prentice-Hall. New Jersey 1984

Conference

A MARITIME SYMPOSIUM

On 28 March the headquarters of the Central Maritime Museum in Gdańsk hosted the interesting symposium on :

„Polish school ships of the years 1920÷2000"

In 1918, just after regaining independence by Poland, efforts were undertaken towards arranging and carrying out education of own maritime personnel for merchant and naval fleets, then under organization. The symposium was just devoted to Polish school ships supporting the maritime education.

Prof. Aleksander Walczak of Maritime University of Szczecin presented the main paper on :

„Role of the school ships in education of the officers for merchant and fishing fleets"

The presentation was completed by representatives of the Polish army who discussed education and training of crews onboard inland navigation ships in the years 1919÷1939 and coast guard ships in the years 1951÷1965.

Two papers were devoted to presentation of the Polish school ships being in service during the last 80 years, such as : LWÓW, DAR POMORZA, DAR MŁODZIEŻY, ISKRA, ISKRA II, as well as the training-cargo ship ANTONI GARNUSZEWSKI and the EDWARD DĄBROWSKI on which the Maritime College operated in the years 1973÷1985.

Other contributors discussed experience gained from exploitation of school sail ships as well as compared performance features of the DAR POMORZA (now ship - museum) and DAR MŁODZIEŻY and their influence on navigation safety of the ships.

Finally, Mr Zygmunt Choreń, M.Sc., presented the latest trends in building the school sail ships and pleasure – training sail ships.

It is worth mentioning, by the way, that today two school sail ships : DAR MŁODZIEŻY of Gdynia Maritime Academy and ISKRA II of Polish Navy still serve the maritime education purposes, and two new, research-training motor ships: NAWIGATOR XXI of Maritime University of Szczecin and HORYZONT II of Gdynia Maritime Academy are also in service.