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COMPARATIVE STUDY ON THE DISPLACEMENTS, EQUIVALENT ELASTIC STRAIN AND EQUIVALENT STRESS OF THE PROPELLER SHAFT AT DIFFERENT OPERATING MODES

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Abstract: This study presents the sizing calculation and the FEM analysis of the propeller shaft for three different operating modes. The propeller shaft has the role of supporting the engine and transmitting its rotating motion and is the most requested element in the structure of the shaft line. In scenario 1, a "fixed support" type constraint is applied at one end and the maximum torque given by the motor at the other end is applied to lock the propeller. Than to simulate scenario 2, apply the maximum thrust force to the propeller at one end, and the "fixed support" constraint will be inserted at the end of the engine. And the 3 th scenario was the operating of propulsion system in normal mode. Comparing the results from the simulation of the three scenarios, it is observed that the maximum values recorded for displacement, equivalent elastic strain and equivalent stresses were recorded at the engine operation under normal conditions. In conclusion, although the scenarios were a bit exaggerated, the propeller shaft withstood the efforts, falling within the limit of elasticity. This demonstrates that the calculation method of propeller shafts is correct, and safe, as long as their size is not oversized.

Key words: propeller shaft, sizing, elastic strain, stress.

1. INTRODUCTION

The Propeller shaft is the aftermost section of the propulsion shafting in the stern tube in single screw ships and in the struts of multiple screw ships to which the propeller is fitted. Propulsion shafting constitutes a system of revolving rods that transmit power and motion from the main engine to the propeller. The shafting is supported by an appropriate number of bearings. [1]

The propeller shaft system is detailed in Figure 1. The intermediate shafting between the tail shaft and main engine, gearbox or thrust block may be supported in plain, tilting pad or roller bearings. The two former types usually have individual oil sumps, the oil being circulated by a collar and scraper device; roller bearings are grease lubricated. The individual oil sumps usually have cooling water coils or a simple cooling water chamber fitted. Cooling water is provided from a service main connected to the sea-water circulating system. The cooling water passes directly overboard. [1].

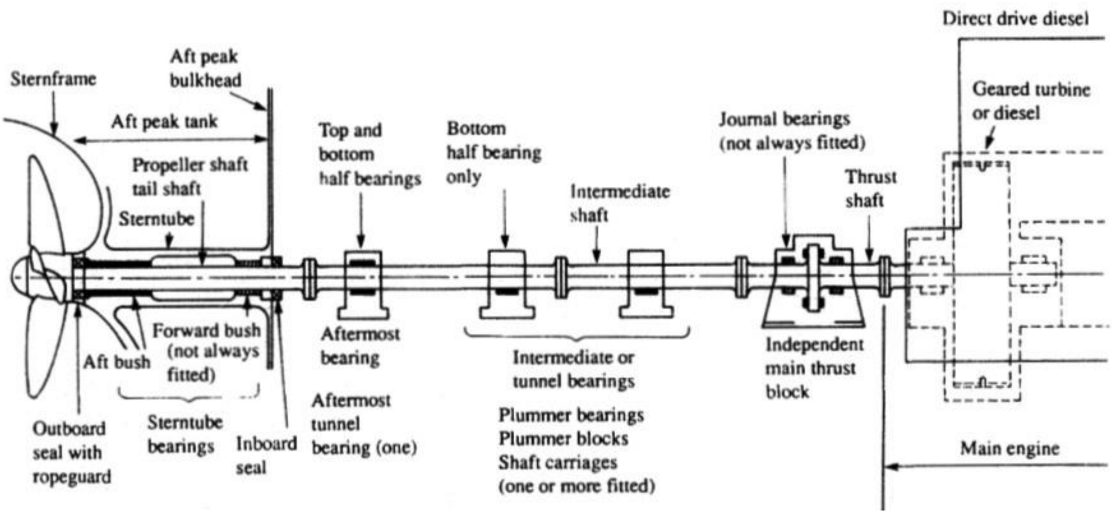

Fig. 1. Propeller shaft system [1]

1.1 Technical characteristics of the reference vessel
The chosen reference vessel is an oil tanker with deadweight of 250950 tdw, intended to transport "black" oil products (crude oil). The construction of the body is of the simple hull type, the cargo area having 2 sealed longitudinal walls and 7 sealed transverse walls (including the bow wall of the pump compartment and the collision wall). The cargo tanks are there located in three rows in the central area comprising a central row of 6 cargo tanks and two side rows (6 pairs) of mixed ballast/cargo tanks, the pump compartment being interspersed between the ER (engine room) and the cargo area. [2]
The main propulsion engine used is a MAN B&W 7S80MC Diesel engine with a maximum continuous power of 79549 kW at 92 r.p.m. (rotations per minute). The engine is 2 strokes (reversible) and has 7 cylinders in line being directly coupled to the engine, normal operation being done with heavy fuel.
The propeller is made of special brass with 4 blades with a diameter of 8200 mm; there is also a spare propeller with the same characteristics.
The heating of the cargo tanks is done with steam supplied by two Aalborg licensed Aquatubular boilers of approx. 2000 tons of steam / hour at 18 bars. For ordinary consumption, the ship is also equipped with a CAUX-ACV-AZL type boiler with a heavy fuel operation with a capacity of 2400 kg steam / hour at 7 bars with automatic operation, as well as an exhaust boiler (with discharge) type CAUX-ACV-GEV which produces 4000 kg steam/hour at 7 kg/cm^2. Boilers can operate both independently and simultaneously [2].

1.2 Propeller shaft material
The minimum dimensions of the shaft diameters, without taking into account the additions for their subsequent turning during the exploitation period, are determined with the formulas given in this paper, in the following paragraph.
In this case it is assumed that the additional stresses produced by the torsional vibrations will not exceed the allowed values. For the calculation of the resistance of the propeller shafts, it is taken into account the fact that they are made of OL 42 [3], a material that has the following properties, according to STAS 500-80: [4]
- tensile strength: 410 … 490 MPa;
- flow limit: 250 MPa;
- elongation at break: 22%;
- permissible tension at twisting: 75 MPa;
- specific gravity: 7850 kg/m^3.

2. THEORETICAL METHODS

In order to realize the three operating scenarios of the propeller shaft, it was necessary to perform the calculation of the shaft line. This calculation provides information about the maximum torque of the engine, the maximum thrust force of the propeller and the dimensions for the propeller shaft.

2.1 Crankshafts line calculation
For the calculation of the intermediate shaft, the additions for the subsequent turning will be neglected during the operation period. It is also considered that the additional stresses generated by the vibrations will not exceed the minimum allowed limits.
The intermediate shaft diameter of must be larger than that determined by the equation (1):

$$d_{in} = \frac{F \cdot K}{3.9} \cdot \left(\frac{P}{n} \cdot \frac{B}{A} \right)^{1/3} \tag{1}$$

where:
P = 79549 [kW] – computing power at the intermediate shaft (rated power at the propulsion engine coupling flange);
n = 1.533 [s^{-1}] – calculation speed of the intermediate shaft;
A = 1 – coefficient taking into account the axial hole in the shaft and which for the intermediate shaft is 1;
F = 100 – factor depending on the type of propulsion system. In this case we are talking about diesel engine propulsion systems;
K = 1 – for shafts with coupling flanges from a shaft coupling or for shafts with coupling flanges mounted without wedge, by pressing [5];
B = 0.918 – coefficient depending on the material from which the shaft is made, OL50. This coefficient was calculated by equation (2):

$$B = \frac{560}{R_m + 160} \tag{2}$$

where:
R_m = 450 [N/mm^2] - breaking strength of the shaft material. [5]
The result is an intermediate shaft diameter of $d_{in} = 963.587 \approx 970 [mm]$

2.2 Propeller shaft calculation
Propeller shaft diameter, d_{pe}, must be larger than the one calculated with the equation (1). The coefficient K is determined for the bow edge between the bow edge of the stern bearing and the bearing shaft, up to the bow face of the propeller hub or, if any, up to the bow surface of the shaft flange on which the propeller is mounted, but in any case not less than 2.5 d_{pe}:
K = 1.22 if the propeller is fixed on the wedge-free shaft, by a method close to the classification registers, or on the one-piece flange made with the shaft;
K = 1.26 if the propeller is mounted with the wedges.

Propeller shaft diameter, d_{pe}, is [6]:

$$d_{pe} = \frac{F \cdot K}{3.9} \cdot \left(\frac{P}{n} \cdot \frac{B}{A} \right)^{1/3} = 1125 \cong 1130 [mm]$$

The thickness t of the bronze protective bushing of the shaft shall not be less than that determined by the equation:

$$t = 0.03 \cdot d_{pe} + 7.5 = 41.4 \approx 45 [mm]$$

2.3 Joining intermediate shafts

The diameter of the bolts of the connecting shafts of the intermediate shaft, thrust shafts and propeller shafts must be larger than that determined by equation (3), [6]:

$$d_d = 0.65 \cdot \left[\frac{d_{in}^3 \cdot (R_{ma} + 160)}{i \cdot D \cdot R_{mb}} \right]^{1/2} \quad (3)$$

where:

i = 20 – number of connecting bolts;

D = 350 [mm] – the diameter of the circle of the connecting bolts centers;

R_{ma} = 450 [N/mm^2] – breaking strength of the shaft material;

R_{mb} = 450 [N/mm^2] – breaking strength of the bolt material [6].

Following the calculations, the diameters for the coupling bolts for the intermediate and thrust shafts, respectively the propeller shaft, result:

- for intermediate / thrust shaft: d_d = 164.145 [mm];
- for propeller shaft: d_d = 191.221 [mm].

The calculations were performed separately for the intermediate shafts and the propeller shaft, resulting in the following values:

- for intermediate / thrust shaft: 5.417 < l < 13.788 [m], the value of 7 m was adopted;
- for propeller shaft: 5.847 < l < 14.882 [m], the value of 9 m was adopted.

3. NUMERICAL ANALYSIS

The constructive complications and the complex demands to which the internal combustion piston engines are subjected make the application of the classic resistance calculation schemes, corresponding to the simple elements known from the resistance of materials (bars, plates, etc.), elements with which some components of these motors can be assimilated with a greater or lesser degree of approximation.

Currently, the finite element method (FEM) is the most widely used numerical method for calculating complex structures represented by internal combustion engine parts, strongly stressed statically, dynamically, thermally, at stability, in linear elastic regime or in various nonlinear conditions.

In order to define the basic concepts of the model of a structure represented by some component of an internal combustion engine that is calculated by FEM, a simple and intuitive possibility is to look at this model as a generalization of the model for calculating bar structures in displacement method.

For to achieve the proposed topic, the Ansys program was used, namely Ansys Static, the part that deals with the static analysis study.

3.1 Model building

The shaft line is required by two large sources of excitation: the main engine by the engine torque and the propeller by the thrust force.

The formula for calculating the maximum engine torque given by the engine is equation (4):

$$M_t = \frac{1000}{2 \cdot \pi} \cdot \frac{P}{n} = \frac{1000 \cdot 60}{2 \cdot \pi} \cdot \frac{79549}{92} = 8316255 [Nm] \quad (4)$$

The maximum thrust force of the propeller has the maximum value corresponding to the minimum speed of 3611.69 kN and the value 3339.58 kN for the normal operating regime at the maximum speed of 24 Nd.

The FEM analysis of the propeller shaft for different operating modes will be presented below.

For the simulation, we proposed three scenarios, as follows:

Scenario 1: The engine runs at full power and the propeller locks instantly;

Scenario 2: The ship moves with the engine stopped, if it is towed or in the event of a major main engine failure;

Scenario 3: The engine operates at maximum power and with a maximum thrust force. This is the normal operating mode of the propulsion system when the ship is moving at maximum speed of 24 Nd.

The propeller shaft has the role of supporting the engine and transmitting its rotating motion and is the most requested element in the structure of the shaft line. And because of this reason it will be analysed.

The constructive dimensions of the propeller shaft obtained from the previous calculations are: diameter 1130 mm and length 8000 mm.

To create the model in Ansys Geometry Design, will be used three successive cylinders with the next dimensions, diameter x length: 1280x100; 1130x8000 and 1280x100 using the "primitives" function.

This model is presented in Figure 2.

The following figure shows the discretized structure of the propeller shaft. This was done in the program using the "generated mesh" and "insert refinement" (Figure

3).

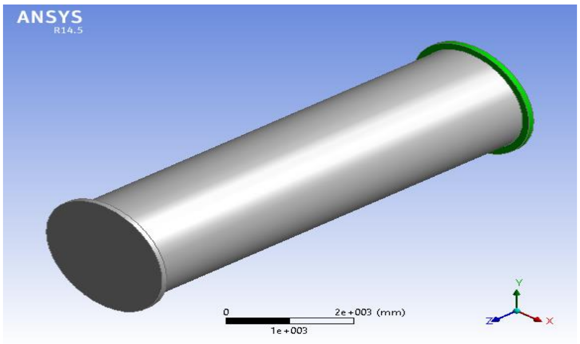

Fig. 2. The 3D design of propeller shaft

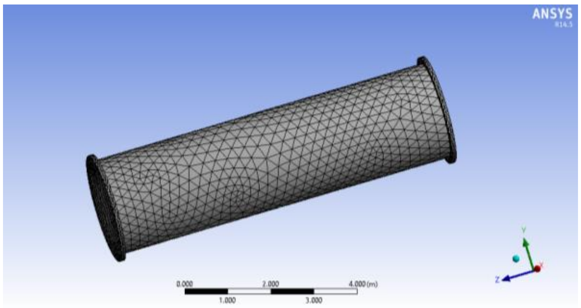

Fig. 3. The meshed design of propeller shaft

4. RESULTS AND DISCUSSION

After creating the propeller shaft 3D geometry and discretizing it were performing in the Ansys program the application of the constraints characteristic to the scenarios proposed.
This section presents the simulations of the three scenarios and then compares the values obtained for equivalent elastic strain and equivalent stress.

4.1 Scenario 1

In the first phase, the propeller is considered to be locked by applying a “fixed support” constraint at one end and the maximum torque given by the engine at the other end (Figure 4). Thus is obtained the diagram of total deformation, of equivalent elastic strain and of equivalent stresses for this case of load (Figure 5).

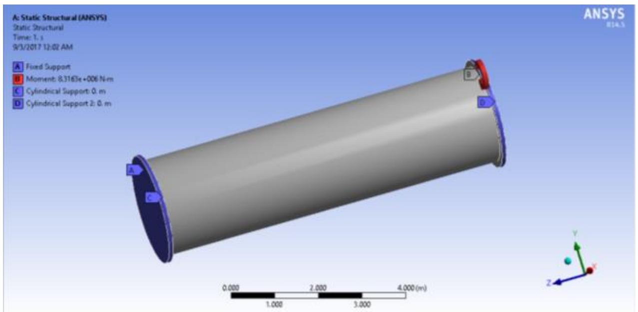

Fig. 4. Propeller shaft loading – scenario 1

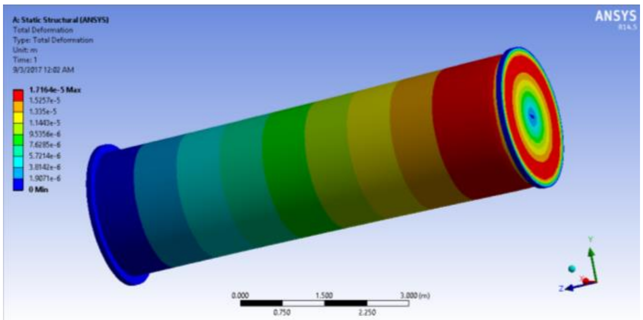

Fig. 5. Total deformation diagram – scenario 1

It can be seen that the maximum deformation of 0.01716 mm occurs near the moment and the maximum efforts appear when joining the flange with the propeller shaft body (Figure 5). The maximum equivalent elastic strain was $8.6068 \cdot 10^{-5} m/m$ and the minimum value was $5.2501 \cdot 10^{-11} m/m$ (Figure 6). For equivalent stress the maximum value was $1.7214 \cdot 10^{7}\,Pa$ and the minimum was $10.5 Pa$ (Figure 7).

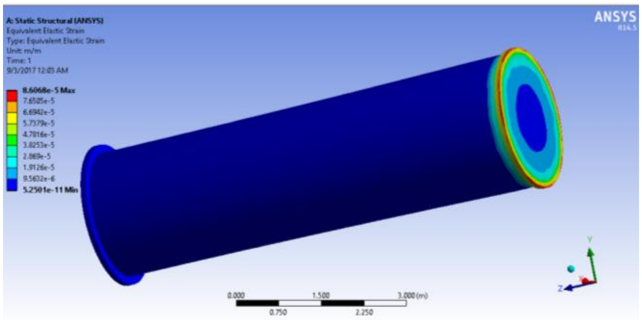

Fig. 6. Equivalent elastic strain-scenario 1

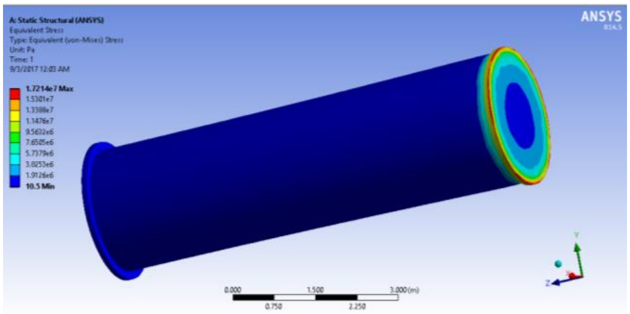

Fig. 7. Equivalent stress – scenario 1

4.2 Scenario 2

To simulate scenario 2, apply the maximum thrust force to the propeller at one end, and the "fixed support" constraint will be inserted at the end of the engine (Figure 8).

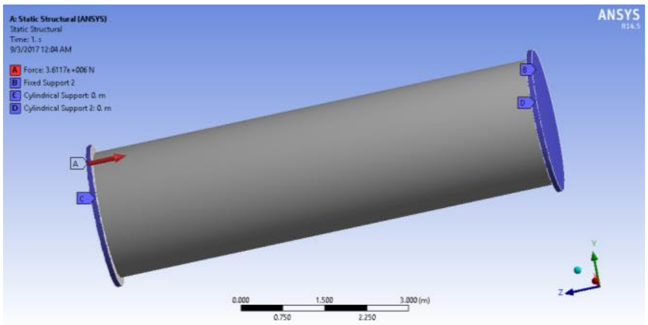

Fig. 8. Propeller shaft loading – scenario 2

In this case, the maximum deformation was recorded at the end where the excitation was applied, and the maximum stress and elastic strain appear at the junction of the flange with the shaft body (Figure 9). For this scenario, the maximum equivalent elastic strain was $5.7214 \cdot 10^{-5} m/m$ and the minimum value was $1.1329 \cdot 10^{-10} m/m$ (Figure 10). For equivalent stress the maximum value was $1.1443 \cdot 10^{7}\, Pa$ and the minimum was $22.657 Pa$ (Figure 11).

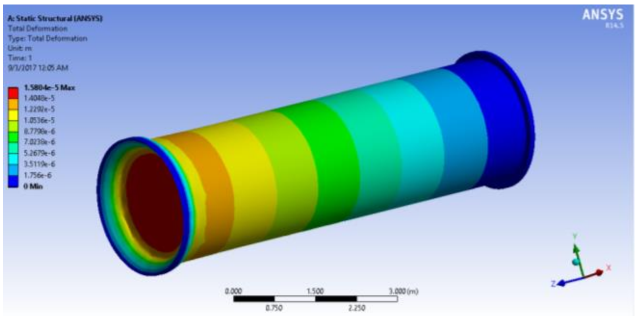

Fig. 9. Total deformation diagram – scenario 2

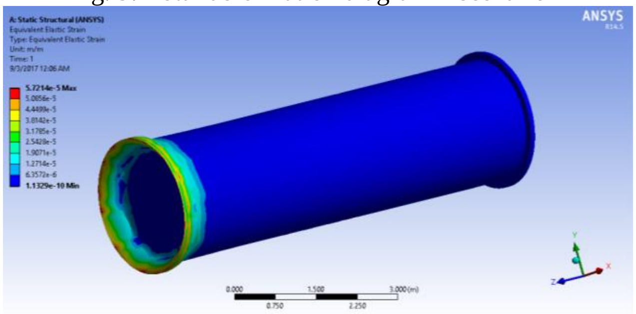

Fig. 10. Equivalent elastic strain–scenario 2

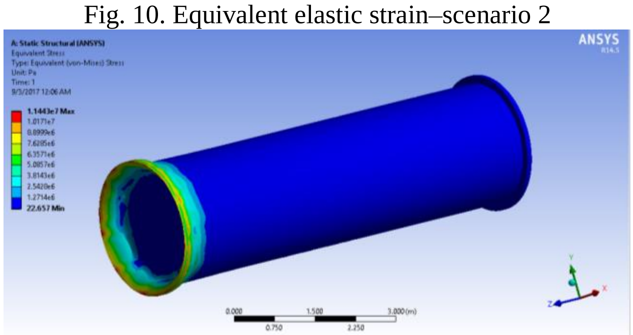

Fig. 11. Equivalent stress. – scenario 2

4.3 Scenario 3

The last case of loading the propeller shaft is the normal operating mode, in which it is loaded with the engine torque on one side and the propeller thrust on the other side (Figure 12).

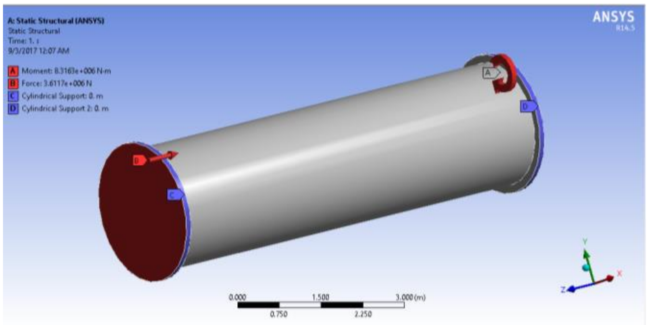

Fig. 12. Propeller shaft loading – scenario 3

When joining the propeller shafts, a "cylindrical support" type constrain will be introduced to simulate their support (Figure 13). The maximum equivalent elastic strain was $8.9802 \cdot 10^{-5} m/m$ and the minimum value was $6.0527 \cdot 10^{-7} m/m$ (Figure 14). For equivalent stress the maximum value was $1.796 \cdot 10^{7}\, Pa$ and the minimum was $1.2105 \cdot 10^{5}\, Pa$ (Figure 15).

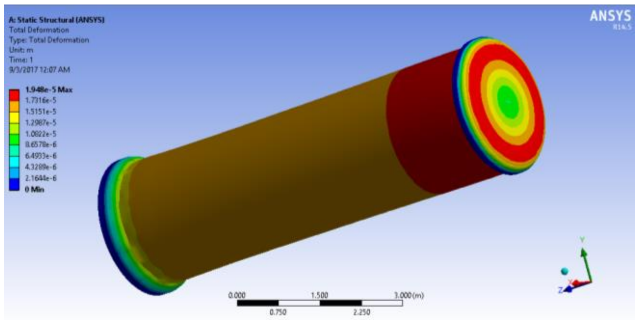

Fig. 13. Total deformation diagram – scenario 3

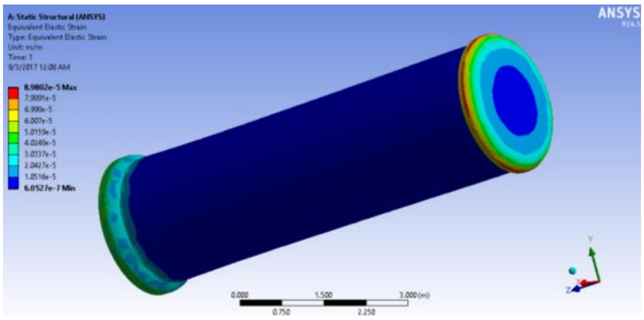

Fig. 14. Equivalent elastic strain–scenario 3

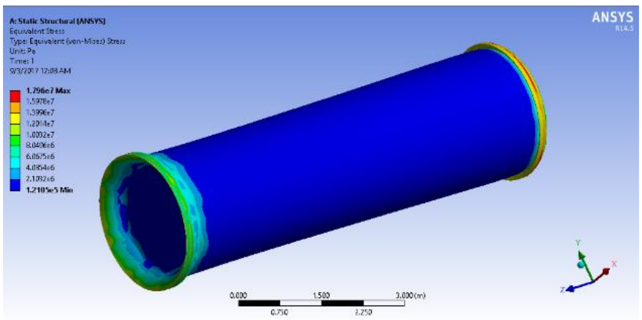

Fig. 15. Equivalent stress. – scenario 3

5. CONCLUSIONS

Crankshaft is the most important moving part of the engine, taking over the gas force from cylinder by means of pistons and connecting rods, transforming together with the connecting rod the rectilinear motion of the piston into rotational motion, transmitting torque to the propeller or electric generator with which the engine is coupled.

Propeller shaft is the aftermost section of the propulsion shafting in the stern tube and constitutes a system of revolving rods that transmit power and motion from the main engine to the propeller.

The aim of the paper was the behavioural analysis of the propeller shaft at different operating modes of the naval engine. This study requires knowledge of the construction dimensions of the propeller shaft, but also the torque and thrust. All this information was obtained by a numerical calculation of the sizing of the tree line.

The behavioural analysis of the propeller shaft was processed in the Ansys program, and the results were presented below each characteristic diagram. From here the following can be concluded:

- the maximum deformation is 0.01948 mm (scenario 3);
- the maximum value of equivalent elastic strain in $8.9802 \cdot 10^{-5}$ m/m (scenario 3);
- the maximum value of equivalent stress is $1.796 \cdot 10^{7}$ Pa.

Comparing the results from the simulation of the three scenarios, it is observed that the maximum values recorded for displacement, equivalent elastic strain and equivalent stresses were recorded at the engine operation under normal conditions (scenario 3).

In conclusion, although the scenarios were a bit exaggerated, the propeller shaft withstood the efforts, falling within the limit of elasticity. This demonstrates that the calculation method of propeller shafts is correct, and safe, as long as their size is not oversized.

6. ACKNOWLEDGMENTS

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7. REFERENCES

1. Propeller shaft bearings - various type, Copyright @ 2010-2016, 9/1-3/2017, 12/3-5/2021, Available from: http://generalcargoship.com/propeller-shaft-bearings.html, Accessed on: 17/02/2022.

2. MarineTraffic, 12/3-5/2021, Available from: https://www.marinetraffic.com/en/ais/details/ships/shipid:429064, Accessed on: 27/03/2022.

3. Germansicher Lloyd, G.L. I –Part 1, Section 4, B. Materials, 1, Approved materials, pp. 4-1, Available from: https://rules.dnv.com/docs/pdf/gl/maritimerules2016July/gl_i-1-2_e.pdf, 9/1-3/2017, Accessed on: 17/02/2022.

4. STAS 500/2-80, Structural steels for general purpose. Grades. Cod CUPS 228000, Available from: https://qdoc.tips/queue/stas-500-2-80-oteluri-marci-pdf-free.html,9/1-3/2017,12/3-5/2021, Accessed on: 15/12/2021.

5. Germansicher Lloyd, G.L.I., – Part 1, Section 4, Main Shafting, C. Shafting Dimensioning, 2, Minimum diameter, Available from: https://rules.dnv.com/docs/pdf/gl/maritimerules2016July/gl_i-1-2_e.pdf p 4-4, p 4-5, 9/1-3/2017,12/3-5/2021, Accessed on: 17/11/2021.

6. Germansicher Lloyd, G.L.I., – Part 1, Section 4, Main Shafting, 4. Coupling connections, p 4-6, p 4-7, Available from: https://rules.dnv.com/docs/pdf/gl/maritimerules2016July/gl_i-1-2_e.pdf, 9/1-3/2017,12/3-5/2021, Accessed on: 20/12/2022.

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