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STRESSED STATE OF INPUT SHAFT FROM THE REDUCER ABOUT ROPE ELECTRIC HOIST

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Разработана модель входного вала планетного редуктора с использованием метода конечных элементов в среде COSMOSWorks — специализированное приложение SolidWorks. Приведены результаты статического анализа и определены напряжения редукторного вала. Определена зона напряженного поля, показывающая евентуальное возникновение трещин в вале.

Розроблено модель вхідного валу планетного редуктора з використанням методу кінцевих елементів в середовищі COSMOSWorks — спеціалізований додаток SolidWorks. Приведено результати статичного аналізу і визначено напруги редукторного валу. Визначено зону напруженого поля, що показує евентуальне виникнення тріщин у валу.

Introduction

In the process of operation many machine elements experience the action of alternating stresses in the time. If these stresses exceed a definite limit, then in the material irreversible alterations begin to run as depending on the accumulated number of cycles the actions of alternating stresses lead to cracking [1, 2]. Furthermore, the known methods for calculating the operational strength of machine elements allow to account for only the force loading with a constant character without accounting-for its probabilistic variation. The stressed-and-srained state of shaft and of assembly is determined by physical-mechanical characteristics of their material, by the values of external forces and moments, by the contact conditions of separate elements, by the distribution character of loading, etc. All these indices have with a probabilistic character and therefore, the stresses provoked by them will also possess a probabilistic character.

Objective of Present Paper

In connection with the statement mentioned above, it is imposed the necessity for determining the fracture probability of shaft in the process of assembly operation to be specified as a criterion for assessment is the stress in the critical section of shaft. Because of that, the objective of present paper is to determine the stresses in input shaft from the reducer about rope electric hoist.

Model of Input Shaft from the Reducer about Rope Electric Hoist

The design of real input shaft from the reducer about rope elec-

Fig. 1. Design of real input shaft from the reducer about rope electric hoist

tric hoist is shown in Fig. 1. The paper deals with the reducer shaft tested by means of laboratory stand according to the joint project between Central Mechanical Engineering Institute-Sofia and BRV-TESMA-Gabrovo as the loading of tested shaft is given in fig. 2 [3].

In the present paper for the objective of study, in the medium of SolidWorks CAD system, a computer model of the considered shaft design has been made. A numerical parametric model of this shaft has been developed by means of the finite element method, which allows the simulation of process and the transmission of torque. The model is made in the medium of COSMOSWorks — specialized application towards SolidWorks [4], destined for solving tasks from mechanics of solid deformable body according to the finite element method as the solved task is from linear statics.

On the units of external cylindrical surfaces of splines, which are in contact with the left radial-axial bearing support (immovable), the following kinematical constraints are imposed: radial and axial zero displacements. On the surface of splines, which are in contact with the splines of central gear, zero angular displacements are imposed. On the surface of contact with the right radial bearing support, zero radial displacement is imposed. On the external cylindrical surfaces of splines towards the clutch, zero radial displacement is imposed as this corresponds with radial bearing support. On the surfaces of shaft

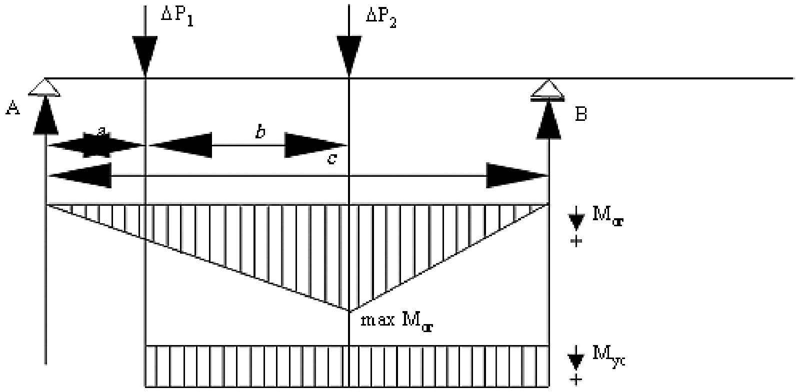
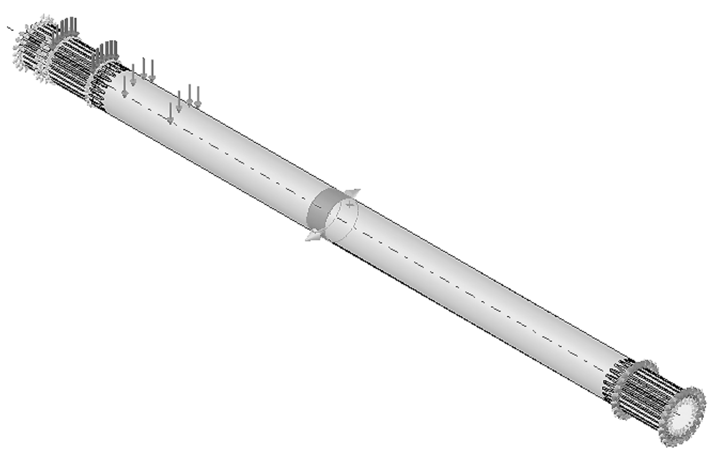


Fig. 2. Loading of reducer shaft tested on laboratory stand
(Central Mechanical Engineering Institute-Sofia, BRV-TESMA-Gabrovo)

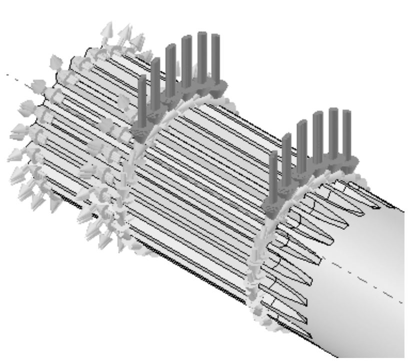
splines, which are in contact with the splines of clutch, the torque M_{yc} = 85 Nm is set. The external cylindrical surfaces of splines (upper ones — according to the scheme), which are in contact with the central gear, are six in number and are loaded by the force P_1 = = 1800 N that is parallel to the Y-axis. The surface of eventual contact of the shaft with the planet carrier is loaded by the force P_2 = = 4200 N that is also parallel to the Y-axis (fig. 3), at which an unfavourable scheme of loading is considered. The finite element mesh of reducer shaft is shown in fig. 4 as control over the size of finite elements has been established and the mesh is concetrated with the purpose of achieving the solution adequacy and accuracy [5].

Results from Carried-out Static Analysis about Reducer Shaft

The developed finite-element model of reducer shaft affords an



a



b

Fig. 3. Model of the reducer shaft: a — loading and constraints of shaft; b — spline part of shaft that is in contact with the central gear

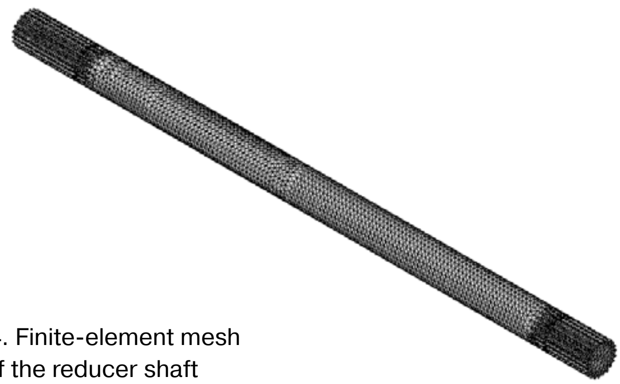


Fig. 4. Finite-element mesh of the reducer shaft

apportunity to determine the following stresses: the equivalent stresses (fig. 5) as well the tangential stresses τ_{yc} (fig. 6), the bending stresses σ_{or} (fig. 7), the shearing stresses τ_{xz} and τ_{xy} (fig. 8), respectively [6]. It can be seen by the graphics about tangential stresses in fig. 6, b that the biggest value is τ_{yc} = 29,7 MPa, which is after the circlip slot. It can be seen by the graphics about bending stresses in fig. 7, b that the biggest value in a longitudinal section is σ_{or} = 26,4 MPa, which is in the circlip slot. These two values of τ_{yc} and σ_{or} correspond to the same ones, obtained at the testing of shaft on the laboratory stand according to the joint project between Central Mechanical Engineering Institute-Sofia and BRV-TESMA-Gabrovo as well as they are proof about the model adequacy and accuracy. It can be seen by the diagrams about shearing stresses τ_{xz} and τ_{xy} in fig. 8 and by the diagram about equiv-

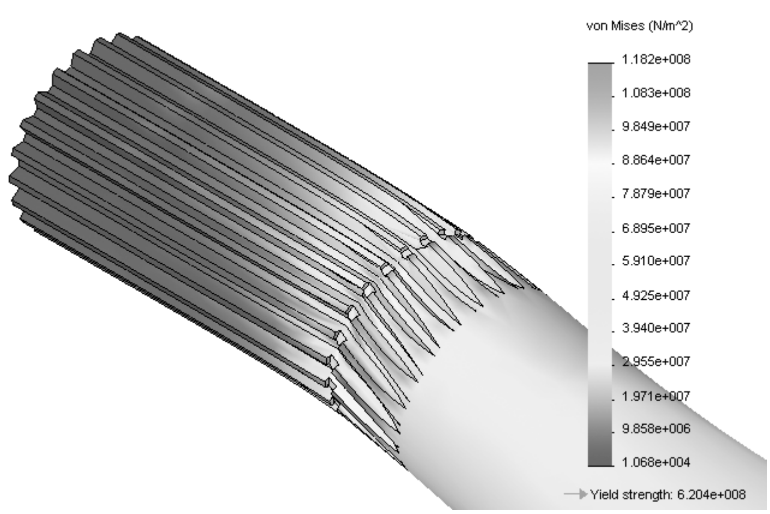
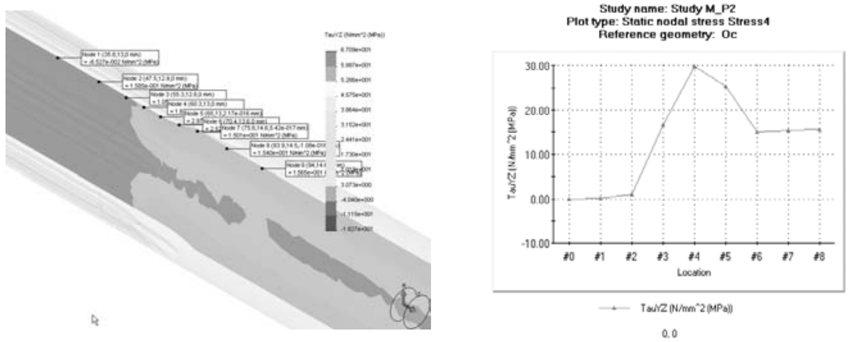


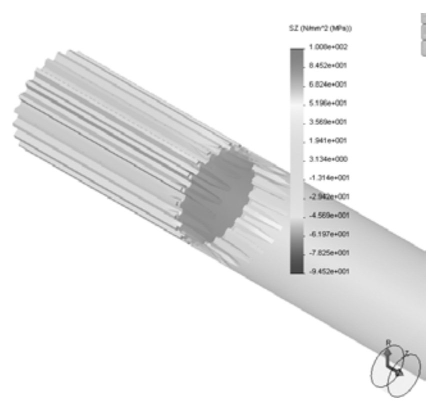
Fig. 5. Equivalent stresses in the reducer shaft



a b

Fig. 6. Tangential stresses τ_{yc} in the reducer shaft: a — tangential stresses in a longitudinal section; b — graphics about τ_{yc}

Fig. 7, a. Bending stresses σ_{or} in the reducer shaft: a — in the slot of circlip



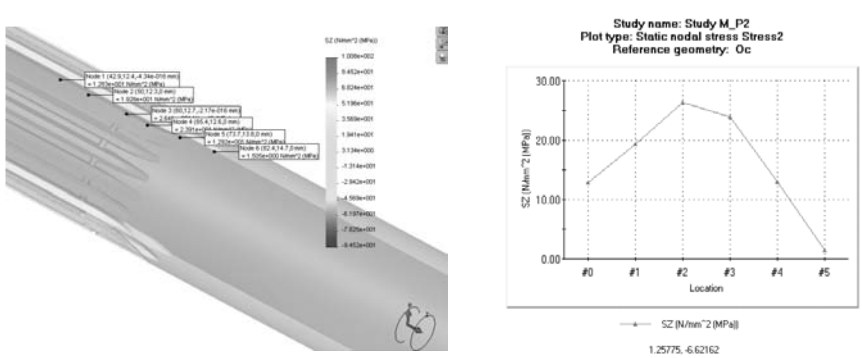


Fig. 7, b. Bending stresses σ_{or} in the reducer shaft: b — in a longitudinal section and graphics about σ_{or}

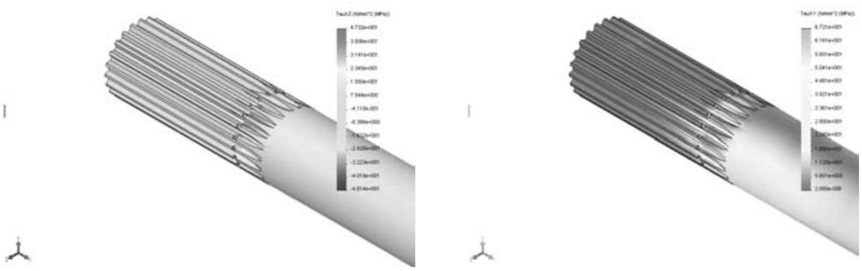


Fig. 8. Shearing stresses τ_{xz} and τ_{xy} in the reducer shaft

Fig. 9. External type of destroyed reducer shaft

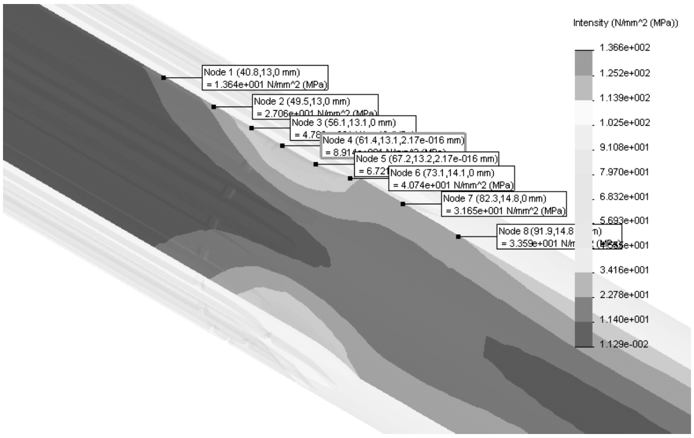


Fig. 10, a. Stress intensity: a — in a longitudinal section

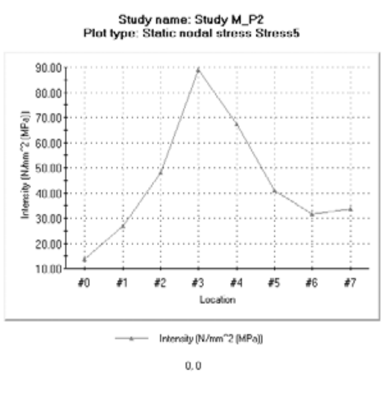


Fig. 10, b. Stress intensity:
b — graphics about intensity
in the reducer shaft

alent stresses in fig. 5 that the maximum values are immediately after the circlip slot as well as they are considerably smaller than the admissible stress [τ] = 620 MPa of shaft material.

When the stresses exceed a definite limit, then in the material irreversible alterations begin to run as depending on the accumulated number of cycles the actions of alternating stresses lead to cracking. The cracks are gradually come off and lead to the sudden fracture of detail. The external type of reducer shaft, destroyed at testing on the laboratory stand according to the joint project beween Central Mechanical Engineering Institute-Sofia and BRV-TESMA-Gabrovo is shown in fig. 9. It can be seen by the figure, that the reducer shaft is destroyed in the spline part near to the central gear.

COSMOSWorks provides a possibility for determining the diagram and the graphics about stress intensity (fig. 10). It can be seen by the graphics in fig. 10, b that the biggest value is 89,1 MPa and it is immediately after the circlip slot. The intensity is specificity of stressed field and it shows eventual initiating of the crack. It yields the zone, which is connected with the critical opening of crack and this leads to the fracture of shaft [7].

Conclusion

The diagrams about equivalent stresses have been presented as well about tangential, bending and shearing stresses upon carried-out static analysis for the tested reducer shaft. These diagrams should be used for analyzing the propagation of fatigue crack in the shaft and for determining the probability of fracture. The zone, whose sizes are connected with the critical opening of crack, is of special interest and it can be determined by means of the diagram about stress intensity. The results from carried-out static analysis for the reducer shaft are a base of implementing the fatigue analysis of shaft in the medium of COSMOSWorks — specialized application towards SolidWorks.

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