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DESIGN AND RESEARCH OF STRUCTURAL ELEMENTS OF THE ELBOW ORTHOSIS PLANETARY GEARBOX

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Abstract. A variety of mechanisms, each designed with distinct specifications, find application in diverse sectors of the machine engineering industry. The utilisation of apparatuses comprising mechanical components in the field of medicine is also evident in the context of patient rehabilitation. These mechanisms are orthoses. The design of such systems is subject to a stringent set of specifications, which include compactness, light weight, and reliability. The purpose of orthoses is to replicate lost functions as closely as possible to a healthy human. It is evident that planetary gears satisfy the criteria for compactness and the requisite technical characteristics. The present paper sets out to model and analyse the stress-strain state of components of a planetary gearbox in the context of an elbow orthosis. The KISSsoft software package was utilised for the design and 3D modelling of components, as well as for determining the stress-strain state. In the course of the design process, the KISSsoft software displays the gear tooth profiles and the line of contact, thus enabling further correction if required. In the modelling of the transmission, the connecting elements were made of 45 steel. The paper provides a detailed analysis of the characteristics of meshing for both cylindrical and bevel gears. The results of the simulation pertaining to the distribution of contact temperature are also presented herein, as are the findings regarding the dependence of reliability and safety factor on service life.

Keywords: Gearbox, stress, deformation, sun, satellite, crown, bevel gear, Kisssoft.

Introduction

In the field of traumatology, the selection of rehabilitation devices is based on well-defined criteria, with key requirements including compact design, ease of use, high reliability, and patient safety. Among such mechanisms, planetary gearboxes stand out due to their small dimensions, stable performance, and the highest efficiency. Mechanical orthoses play a critical role in postoperative rehabilitation and typically consist of an electric motor and a gearbox.

> *Mary Barra (CEO of General Motors):*
> ***"The convergence of electrification, connectivity, and automation is reshaping the future of engineering and manufacturing."***
> *(About the transformation of automotive and engineering industries)*

The design of such orthoses requires precise determination of the drive parameters and motor power to ensure movements that closely mimic those of a healthy elbow, particularly in terms of elbow speed and force.

Problem Statement

Mechanisms often fail during operation due to various factors, with one of the main causes being failure due to stress concentration. It is in these areas that stress and strain reach their maximum values. Planetary gearboxes, frequently used in traumatology, consist of gear mechanisms.

Therefore, during their design, it is essential to carefully consider the gear tooth geometry and potential contact stresses that occur during operation to ensure the reliable and long-lasting performance of the entire mechanism. Since the gears are mounted on shafts, the analysis and modeling of the stress-strain state become crucial steps in the design of mechanical orthoses.

Review of Modern Information Sources on the Subject of the Paper

In recent years, notable advancements have been achieved in understanding the dynamic behavior of rotor systems. A considerable number of studies have focused on modeling both static and dynamic processes using various engineering software tools such as Ansys, KISSsoft, Abaqus, and others. For example, the dynamics of flexible rotors equipped with active magnetic bearings were analyzed in [1] using the finite element method (FEM) within Ansys Workbench. A comprehensive study of a stepped composite rotor shaft, incorporating loads, boundary conditions, vibration characteristics, and a Campbell diagram, is detailed in [2]. The stress-strain state (SSS) of the shaft, evaluated with Ansys tools, is discussed in [3].

In [4], an engine shaft was modeled in CATIA, followed by static, dynamic, and fatigue analyses performed using Ansys. The modal and harmonic analyses of the stress-strain state in planetary gearbox components made of 45 steel and PLA polylactide are presented in [5]. Strength and stiffness assessments of shafts subjected to high torque, relevant for automotive and marine applications, are provided in [6]. The study in [7] focuses on early-stage detection of structural weaknesses by analyzing the static behavior of a pump shaft model under various loading scenarios using FEM.

Shaft optimization and strength assessment employing FEM techniques are reported in [8]. The application of KISSsoft for 3D modeling and stress analysis of planetary gear shafts in an elbow orthosis is described in [9], with a comparative study using von Mises and Tresca criteria. The influence of cracks on the dynamic performance and structural integrity of a rotating stepped shaft is explored in [10] through FEM simulations.

Analytical and numerical evaluation of the stress-strain state of hydraulic unit shafts, accounting for specific design features, is carried out in [11]. Advanced modeling and optimization of the SSS for high-strength rotor and shaft components used in high-speed machinery are presented in [12, 13]. The static strength, fatigue durability, and bending stiffness of a bevel spur gear shaft are examined in [14]. Lastly, [15] investigates the structural behavior and strength of a steel coil shaft used as a drive shaft in wire rope winding under progressively increasing load conditions.

Objectives and Problems of Research

Dieter Zetsche (former Chairman of Daimler AG):
"In the future, it will not be the big fish eating the small fish, but the fast fish eating the slow one. Agility in engineering is everything."
(About the importance of flexibility and rapid adaptation in mechanical engineering)

To create an efficient and reliable design of a planetary gearbox that meets all technical requirements, it is essential to conduct detailed mechanism design and analyze the stress-strain state of its components. This allows identifying potential critical points in the structure that could lead to malfunction, which, in turn, may pose a risk to patient safety. Given this, numerical modeling of the stress-strain state is an important task during the design phase. The use of FEA significantly reduces the calculation time compared to traditional methods of shaft design and calculation described in the engineering literature, helps prevent errors in calculations, and improves their accuracy. In this work, the software KISSsoft was used to perform the necessary calculations for the planetary gearbox components.

Main Material Presentation

Since the output shaft of the planetary gearbox must be perpendicular to the input shaft, this configuration can be achieved using either a bevel or a worm gear. A worm gear is not considered in this case, as it is generally avoided in applications where high efficiency is required. Therefore, the proposed mechanism incorporates a bevel gear, as illustrated in Fig. 1 [9].

The maximum recommended gear ratio for a single planetary stage is 9. In our case, to meet the technical requirements, a total gear ratio of at least 107 is needed. This implies that the gearbox must consist of at least three planetary stages. From the perspective of manufacturability and compact design, it is reasonable to use identical planetary stages throughout all three stages.

The highest load in the selected configuration is expected to act either on the teeth of the bevel gear or on the meshing teeth of the sun and planet gears in the third planetary stage [5].

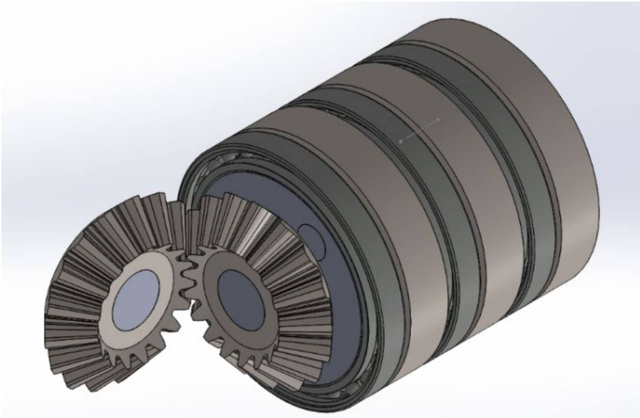


Fig. 1. 3D model of planetary gearbox for an elbow orthosis

These gear pairs are considered sequentially, and their dimensions are selected accordingly. The gear dimensions were determined in KISSsoft using a two-step approach: initially, based on minimal input data (such as gear module ratios, wheel parameters, material properties, and applied loads), preliminary gear sizes were estimated. These were then refined using specified ranges for the number of teeth, module, center distance, etc., depending on the design requirements. A detailed calculation can then be carried out based on the refined data. For the simulation, 45 steel was selected as the material for the gears [9]. According to the technical specifications, the output shaft rotates at a speed of 2.8 s^{-1}, which means the sun gear of the final planetary stage rotates five times faster. The required power to operate the mechanism is 30 W.

The parameters of the planetary gear elements used in the design are given in Table 1 [9]. Mechanical characteristics of the materials from which the structural elements are made are given in Table 2 [9].

Table 1

Planetary gear parameters

Parameter	Sun	Planet gear	Ring gear
Dividing diameter	9.6	13.6	38.4
Diameter of tooth tips	10.840	14.957	37.985
Initial diameter	10.097	14.303/13.381	37.871
Center distance	12.20	12.20	26
Module	0.4	0.4	0.4
Number of teeth	24	34	96
Width of the crown	8	8	8
Engaging angle	26.686	-	17.253
Tooth height	0.799	0.799	0.9
Tooth thickness	2.155	2.261	1.227
Average specific sliding	0.419	0.512	0.316
Face overlap	1.121	1.12	1.583
Safety factor of the tooth pedicle	5.333	2.548	3.415
Safety factor of the lateral surface of the tooth	1.541	1.609	2.231
Efficiency factor	0.978		

Table 2

Material properties

Parameter	Units of measure	Steel 45
Density	kg/m^3	7850
Thermal expansion coefficient	C^{-1}	$1{,}2 \cdot 10^{-5}$
Young modulus	Pa	$2 \cdot 10^{11}$
Poisson's ratio		0,3
Shear modulus	Pa	$7{,}69 \cdot 10^{10}$

According to the characteristics given in Table 1, the KISSsoft program was used to model the planetary transmission. The modeling results are shown in Fig. 2. The presented 3D model of the transmission consists of the sun, three satellites, and the crown.

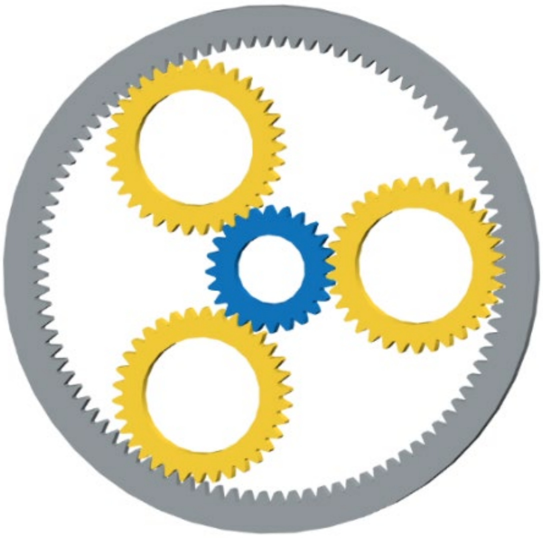
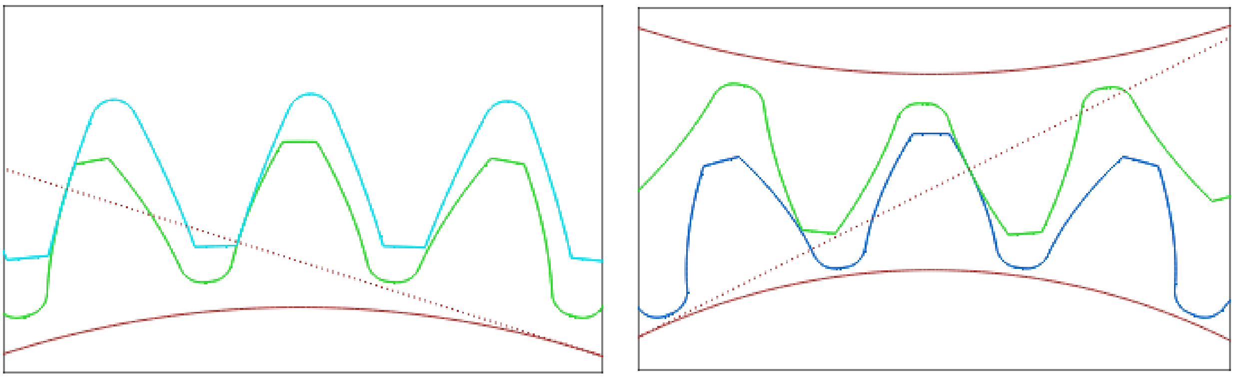


Fig. 2. General view of planetary transmission

The engagement of transmission elements is an important step in the design process. Engagement of the satellite-crown and sun-satellite planetary transmission proposed in this paper for the elbow orthosis gearbox is shown in Fig. 3.



a *b*

Fig. 3. Planetary gearbox: a) satellite-crown engagement, б) sun-satellite engagement

Stresses on the lateral surface of the tooth due to contact are calculated Eq. (1) according to ISO 6336:

$$\sigma_H = Z_B \sigma_{H0} \sqrt{K_A K_V K_{H\beta} K_{H\alpha}},\ \sigma_{H0} = Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t(u+1)}{d_1 bu}} \tag{1}$$

Where F_t - circular force; u - transmission ratio of the gearbox; b - width of the tooth ring; Z_H - coefficient of tooth shape; Z_E - coefficient, depends on the wheel material; Z_ε - coefficient of the total length of contact lines; Z_β - coefficient, depends on the angle of the teeth; Z_B - coefficient, depends on the design of the gear or wheel; K_A - load unevenness coefficient; K_V - dynamic load coefficient; $K_{H\beta}$ - coefficient of uneven load distribution over the tooth surface; $K_{H\alpha}$ - coefficient, depends on the gearing geometry.

Bending stress on the tooth pedicle Eq. (2):

$$\sigma_F = \sigma_{F0} K_A K_V K_{F\beta} K_{F\alpha},\ \sigma_{F0} = Y_{Fa} Y_{Sa} Y_\varepsilon Y_\beta \frac{F_t}{bm_n} \tag{2}$$

where m_n - engagement module; Y_{Fa} - coefficient, depends on the shape of the tooth; Y_{Sa} - coefficient, depends on the stress correction; Y_ε - coefficient, depends on the overlap coefficient; Y_β - coefficient, depends on the angle of the teeth; $K_{F\beta}$ - coefficient of uneven load distribution over the tooth surface; $K_{F\alpha}$ - coefficient, which depends on the gearing geometry.

A flat plate triangular finite element with 5 degrees of freedom was used to calculate the stress-strain state (SSS). Figs. 4 and 5 show the results of modeling the stress and deformation distributions, and Table 3 shows the largest stresses and strains, respectively.

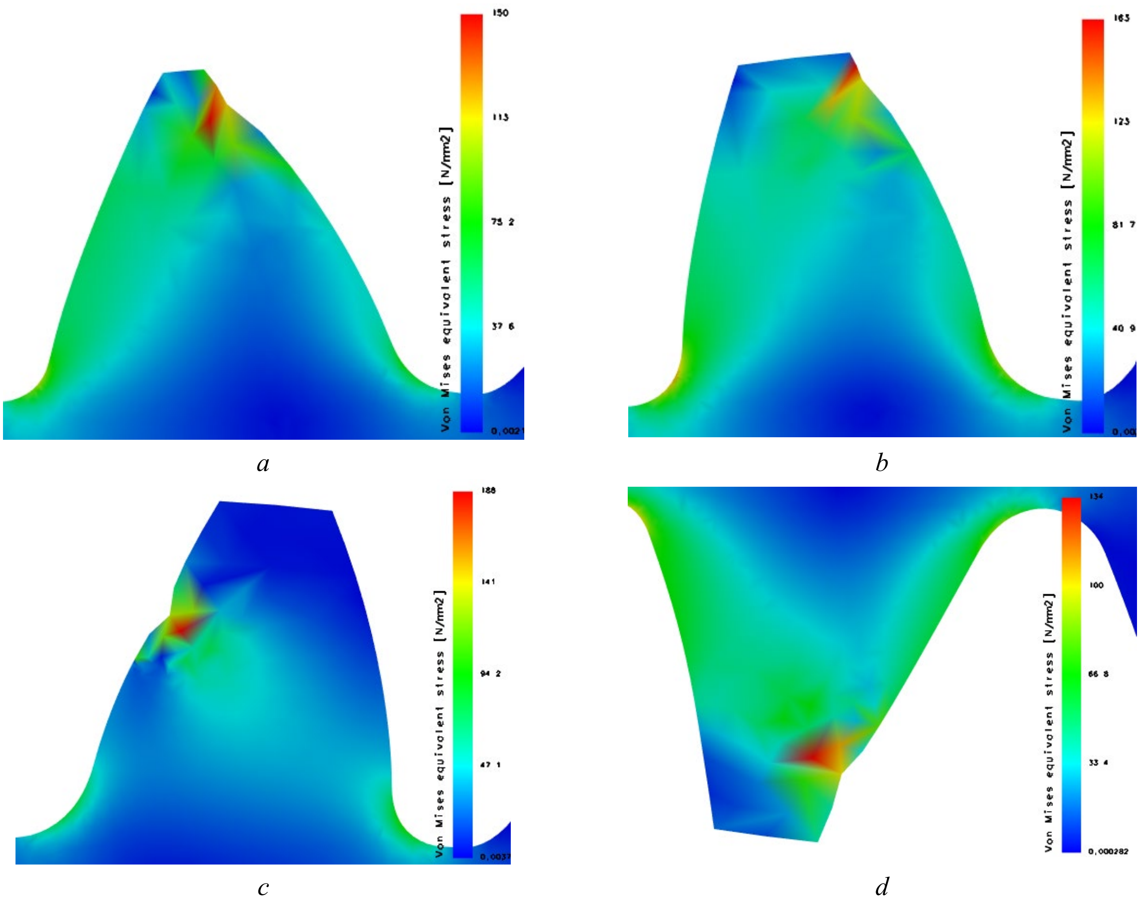


Fig. 4. Stresses in the tooth root according to the FEM: a) Sun, b) Satellite in contact with the sun, c) Satellite in contact with the crown, d) Crown

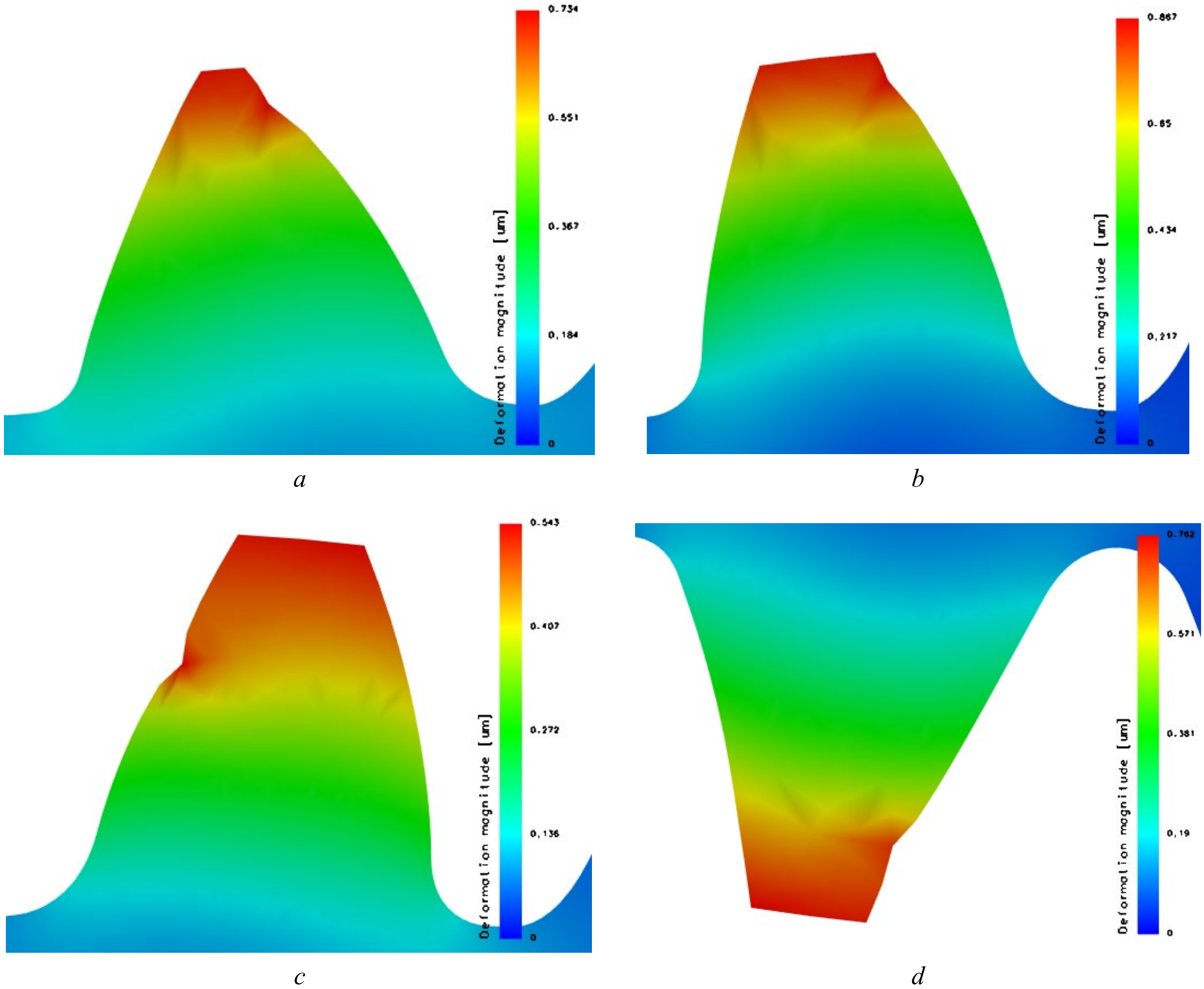


Fig. 5. Deformations according to the FEM:
a) Sun, b) Satellite in contact with the sun,
c) Satellite in contact with the crown, d) Crown

The maximum values obtained as a result of numerical modeling are shown in Table 3.

Table 3

Stresses and deformations

Body	Stresses (MPa)	Deformations (nm)
Sun	156	0.734
Satellite in contact with the sun	163	0.867
Satellite in contact with the crown	183	0.543
Crown	134	0.763

The sizing of a bevel gear can begin with a refined selection based on a known top diameter of less than 35 mm to ensure a compact gearbox. The standard helix angle is 35° and the tooth width is greater than 8 mm to ensure strength. Fig. 6 shows the possible options for meshing characteristics in the KISSsoft program. The optimal characteristics are selected based on strength.

Nr.	a_v [mm]	b_1 [mm]	b_2 [mm]	m_{mn} [°]	α_n [°]	β_{m2} [°]	z_1	z_2	z_1+z_2	x_{hmn1}	x_{hmn2}	x_{smn1}
0	**0.0000**	**9.0000**	**9.0000**	**0.9091**	**20.0000**	**0.0000**	**37**	**37**	**74**	**0.0000**	**0.0000**	**0.000**
1	0.0000	8.0000	8.0000	1.1730	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
2	0.0000	8.0000	8.0000	1.1735	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
3	0.0000	8.0000	8.0000	1.1739	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
4	0.0000	8.0000	8.0000	1.1744	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
5	0.0000	8.0000	8.0000	1.1749	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
6	0.0000	8.0000	8.0000	1.1754	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
7	0.0000	8.0000	8.0000	1.1759	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
8	0.0000	8.0000	8.0000	1.1764	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
9	0.0000	8.0000	8.0000	1.1768	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
10	0.0000	8.0000	8.0000	1.1773	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
11	0.0000	8.0000	8.0000	1.1778	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
12	0.0000	8.0000	8.0000	1.1783	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
13	0.0000	8.0000	8.0000	1.1788	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
14	0.0000	8.0000	8.0000	1.1792	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
15	0.0000	8.0000	8.0000	1.1797	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000
16	0.0000	8.0000	8.0000	1.1802	20.0000	35.0000	17	17	34	0.0000	0.0000	0.000

Fig. 6. Selection of bevel gear characteristics in KISSsoft

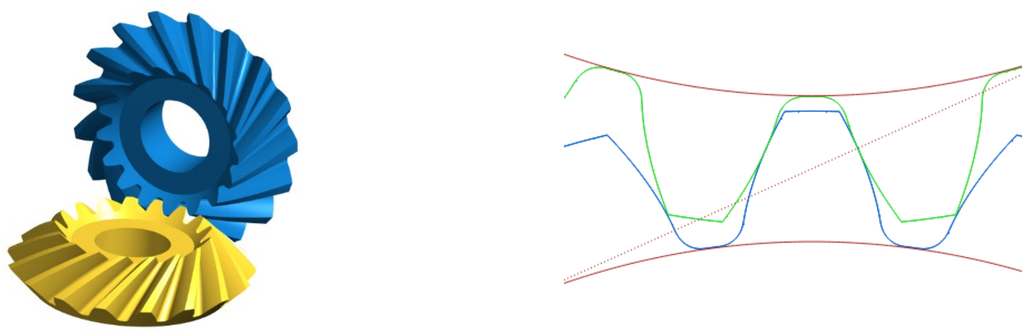
The selected transmission parameters are shown in Table 4.

Table 4

Bevel gear parameters

Parameter	Value
Number of teeth	17
Width of the tooth ring	8mm
Module	1.2mm
Engaging angle	20
Engaging spiral angle	35
Dividing diameter	30.561mm
Diameter of tooth tips	32.643mm
Dividing cone angle	45
Face/normal overlap	1.231/1.734
Safety factor of the tooth pedicle	6.125
Safety factor of the lateral surface of the tooth	1.942
Efficiency factor	0.98

Since the safety factor is over 1.5, further design is possible. The internal diameter of the wheels was preliminarily selected to be 10 mm. Fig. 7 shows the meshing and the 3D model of the bevel gear wheels in the KISSsoft software package.



a *b*

Fig. 7. 3D model of bevel gear a) and bevel wheel meshing b)

The following results were obtained as a result of designing a bevel gear transmission for an elbow orthosis in the KISSsoft software package:

1. Contact temperature. This is the local temperature on the tooth at the moment of contact. Based on the contact temperature and its location on the lateral surface of the tooth profile, the necessary corrections (e.g., tooth profile modification) can be made to reduce the temperature.

Tooth surface destruction is possible as a result of seizure, which occurs when the temperature in the contact zone rises significantly due to high heat generation from tooth friction and poor cooling. Therefore, it is necessary to maintain the temperature of the tooth surface within the normal range. As can be seen from the graph (Fig. 8), the highest temperature occurs at the edge of the tooth and is 98°C.

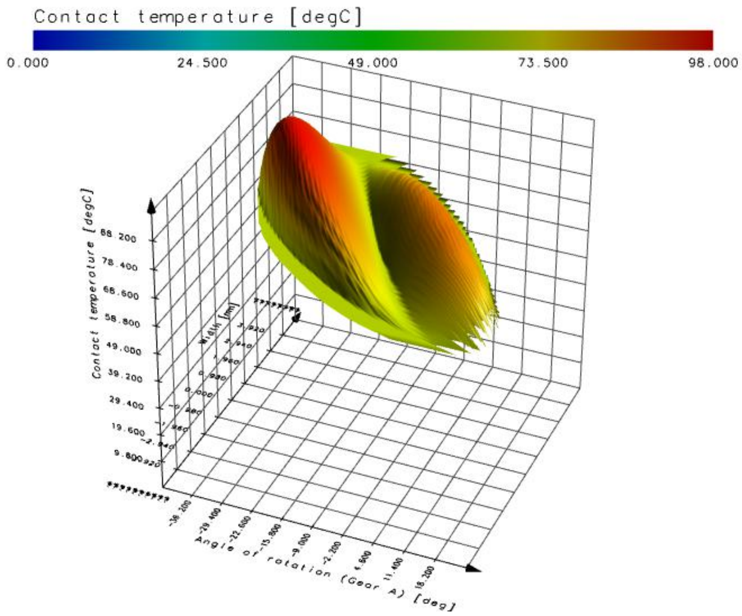


Fig. 8. Graphical distribution of the contact temperature

2. Weller's curve. A curve of dependence of the number of load cycles before failure on the maximum stresses occurring in a structural element. The ordinates of the points of this curve represent the amplitudes of stresses, and the abscissas represent the number of cycles they can withstand corresponding to these amplitudes (Fig. 9).

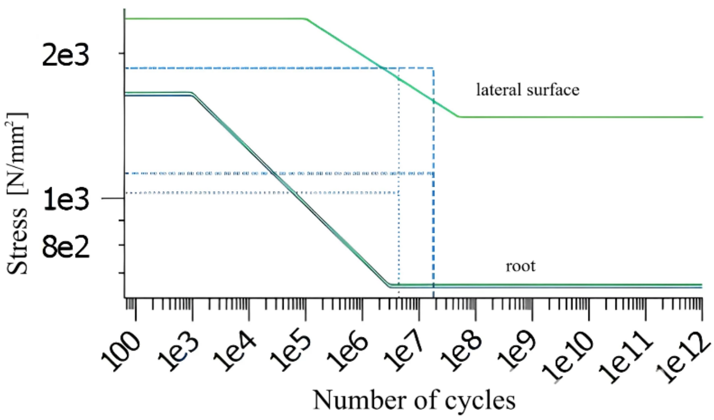


Fig. 9. Weller's curve

3. Reliability. This is a complex property of an object, which consists in its ability to perform specified functions while maintaining its main characteristics within the established limits. The graph (Fig. 10) shows the dependence of reliability on service life.

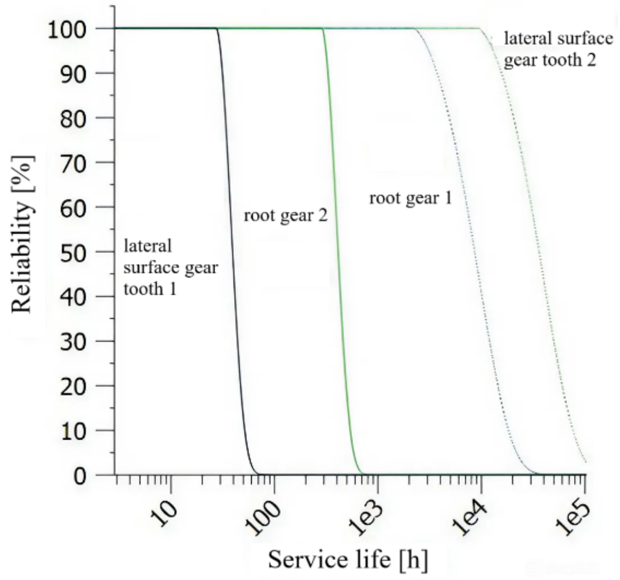


Fig. 10. Dependence of the reliability of the meshing elements on the service life

4. Safety factor. The graph (Fig. 11) shows the results of numerical modeling of the dependence of the safety factor on the service life. The wheel root and the gear root have higher initial safety factors.

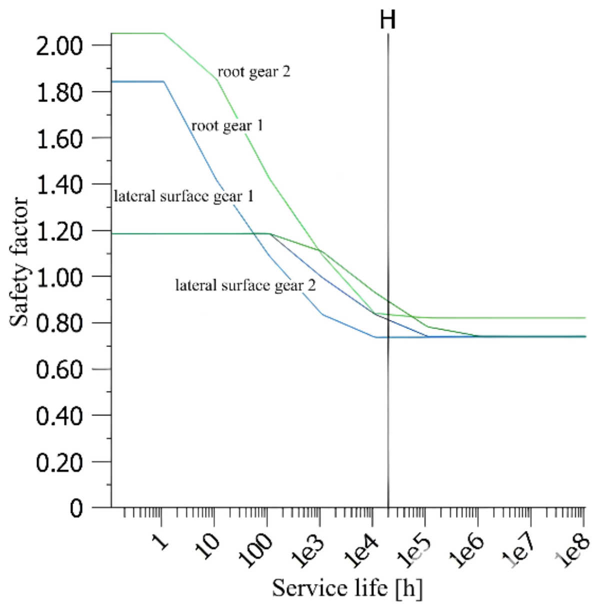


Fig. 11. Dependence of the safety factor of the gearing elements on the service life

Conclusions

In this paper, we designed a gearbox for an elbow orthosis. The design was carried out using the KISSsoft software package, which uses the ISO 6336 standard for calculating the strength of gears.

The obtained values of contact stresses and deformations in the gearing remain within the permissible limits. The maximum stress values are 183 MPa, and the strain is 0.867 nm, respectively.

According to the results obtained for the bevel gear, the highest stresses occur on the lateral surface of the tooth (2300 MPa). After 10^5 cycles, the stresses begin to decrease by about 30% and eventually amount to 1600 MPa.

The initial stresses in the gear tooth root are 26% lower than the stresses (1700 MPa), and after 103 cycles, the stresses begin to decrease. The minimum stresses are 500 MPa. It can be concluded that the lateral surface of the tooth is more stressed during the entire service life.

From the graphical representation of the temperature distribution at the contact point, it was found that the highest temperature was observed at the tooth edge and amounted to 98°C.

The first to lose reliability is the side surface of the gear tooth. The side surface of the wheel tooth will retain its reliability for the longest time.

Although the lateral surfaces of the teeth have lower initial safety factors, at the time of the planned service life, this value decreases by 30%, which is much less than the loss of safety factor in the roots of the wheel and gear.

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