DESIGN OPTIMIZATION OF THE GEARBOX TO ELIMINATE MALFUNCTIONS

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Abstract: The paper deals with gearbox optimization. It is a two-speed bevel-helical gearbox, which is part of the cooling tower drive. The gearbox malfunctions often occurred during operation. For this reason, changes have been proposed to eliminate transmission failure. The changes concerned the bevel gearing as well as the helical gearing, the body of the gearbox and its lubrication.

Keywords: optimization, helical gear, bevel gear, gear box body

1 Introduction

In the production of equipment and machines with higher performance at reduced weight, greater demands are placed on the development of new technologies, which is closely related to the constant improvement of the structure. Tendencies of developed parts are to significantly increase the speed of movement of individual members of the mechanisms, rise of operating temperatures, increase in the load on components and other operating parameters [1 – 4]. These operating parameters bring more demanding theoretical problems. It requires scientific in the design of units, as well as in the design of individual parts of machines [5 – 7].

Each device or machine consists of a large number of components. We would not be able to responsibly design machine mechanisms and the whole machine if we could not recalculate, design and manufacture its elements [8, 9].

A machine part is a building element which, after production, is no longer machined by any further assembly operations and on the other hand, cannot be disassembled into simpler parts. We divide components into simple and complex ones [10 - 13]. The simple parts include, for example key, nut, screw: complex components include for example gearbox housing. The part is most often made of one piece of material.

The components are assembled into assembly units or groups. Such group forms a functional unit. These groups are divided into simple and complex ones [14, 15]. More complex groups can consist of several simpler groups, or of separate components.

An assembly group is called a mechanism when it is used in machines and devices to perform purposeful, useful and unambiguous movements. The mechanism transmits not only motion but also forces and transforms one type of motion into another [16]. The mechanism allows a part or group of parts to move along a predetermined path.

The gear is a three-piece mechanism consisting of a pair of meshing gear wheels and a stationary frame with the axles of both wheels. The gears form a rotating kinematic pair with the frame and the sides of the gearing meshing form the so-called general kinematic pair with a point or line touch [17, 18].

The gearing i.e., the system of meshing teeth on both gear wheels, must meet the basic requirement - to achieve a constant gear ratio between the two gear wheels. In addition, it must meet other functional requirements (high load-bearing capacity and efficiency, smooth and quiet operation, etc.) and production requirements (suitable and productive method of production, inspection and assembly) [19, 20].

The task of the transmission is to ensure the transfer of mechanical energy from the driving machine to the driven machine, while a certain parameter (speed, torque, type of movement, etc.) is modified.

The two-stage bevel gearbox ensures the transfer of mechanical energy from the driving machine to the driven machine. The gearbox is an important part of the cooling tower.

The cooling tower ensures the cooling of the turbine condensers, in which the steam is taken away from its no longer usable energy, while in the cooling towers steam and small water particles escape into the air.

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The gearboxes for cooling tower are required to be water, dust and temperature resistant. The article is devoted to the optimization of gearing and bearings of a bevel helical gearbox for cooling towers.

2 Requirements for bevel helical gearbox

The request was initiated from the contracting authority. Originally, the line had a two-stage bevel helical gearbox (Fig. 1) for cooling towers, which provide cooling for turbine condensers, which also include this gearbox. This gearbox transmits high power from the electric engine to the input shaft via the countershaft and drives the propeller in the cooling tower through the output shaft. This gearbox malfunctions regularly during operation. The input speed was equal to $n_1=1500\ \mathrm{min}^{-1}$, the total transmission ratio was equal to 11.2, the axial distance of helical gearing was equal to 315 mm.

Figure 1. Bevel helical gearbox.

The gearbox is designed to reduce the speed of the drive machine - electric engine, to the input speed of the driven machine in a constant gear ratio. The input shaft is positioned horizontally, the output shaft vertically upwards. The gearbox consists of a bevel gearing and a helical gearing mounted by means of bearings. The maximum engine torque must not exceed 1.7 times the rated torque on the transmission input shaft. This condition can be achieved in operation by regulating the current of the electric engine during starting (two-speed motors, frequency converter, electric engine starter, etc.) or by using a clutch with soft characteristic, it is not recommended to use an asynchronous engine drive with a short - circuit rotor, controlled by direct connection to the grid.

After removing the top cover of the gearbox, damage to the bearing cage was found (Fig. 2), which should ensure even spacing of the rollers around the circumference.

Figure 2. Damaged bearing on input shaft.

After replacing the damaged bearing, the correctness of the meshing of the bevel gears followed (Fig. 3), where the meshing area is determined by the color respectively testing paste. Testing paste is characterized by good adhesion to metal or sliding surfaces, does not dry out and smudge. The teeth are painted with the paste, when the wheels of the gear turn at the point of meshing, the paste is wiped off, and clear imprints appear on the sides. The measurement did not show incorrect meshing or any unevenness on the meshed surfaces of the gears.

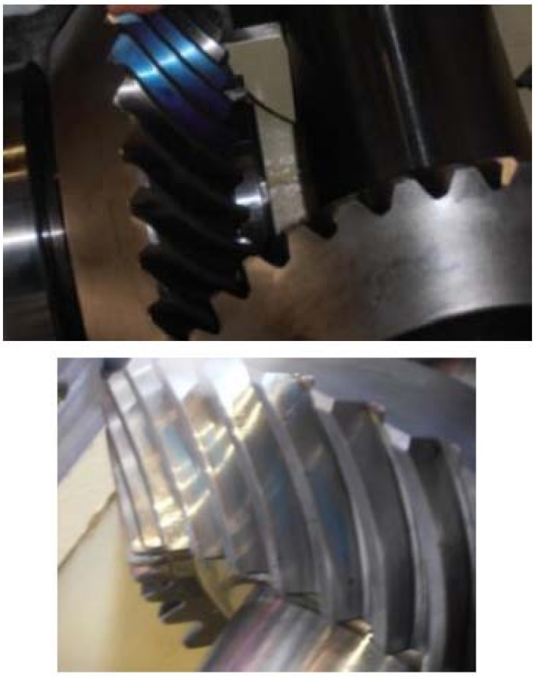

Figure 3. Check of the correct meshing.

The gearbox was lubricated by dipping the wheels in lubricating oil. To eliminate the failure rate, a strength check of the bevel and spur gears was performed, on the basis of which the optimization of the gearbox was designed.

3 Design of transmission optimalization

At the beginning, a new modified material for bevel gears was chosen. Specifically for the bevel pinion, the material steel 16 220.4 was chosen - surface hardened gearing on the sides. The heat treatment of the wheel was designed as cementing and hardening. The tensile strength of this material is 880 MPa, the yield strength is 635 MPa. The material of the bevel face gear wheel was chosen to be from steel 14 220.4 - surface hardened gearing on the sides. The heat treatment of the wheel was designed as cementing and hardening. The tensile strength of this material is 785 MPa, the yield strength is 688 MPa.

For the second spur gear, the material was chosen to be the same as for the first gear i.e., for the helical pinion material, material 16 220.4 - surface hardened gearing on the sides and for the helical gear wheel 14 220.4 - surface hardened gearing on the sides.

The modulus m_{nm} at the mean conical distance was determined from the greater value of bending or contact stress according to [21].

The calculation of the modulus from bending stress:

$$m_{nm} = f_F \cdot \sqrt[3]{\frac{K_F . M_{kI} \cdot \left(\frac{1}{\psi_L} - 0{,}5\right) . \cos^2 \beta_m}{z_1 . z_C . \sigma_{F\,limb}}} \qquad (1)$$

where f_F is auxiliary factor for calculating the gear modulus [mm], K_F is coefficient of additional loads for bending calculation, M_{k1} is torque on the pinion, [Nm], Ψ_L is conical value of relative width, β_m is gearing helical angle, [°], z_1 is pinion number of teeth, z_c is planar wheel number of teeth, σ_{Flimb} is bending fatigue stress of material [MPa].

Modulus calculation from contact:

$$m_{nm} = f_H \cdot \sqrt[3]{\frac{K_H . M_{kI} . \cos \delta_1 . \cos^5 \beta_m \cdot \left(\frac{1}{\psi_L} - 0{,}5\right) . ({u_1}^2 + 1)}{{z_1}^2 . z_C . {u_1}^2 . \sigma 2_{Hlim}}} \qquad (2)$$

where f_H is auxiliary factor for calculating the pinion pitch circle, K_H is coefficient of additional loads for contact calculation, δ_1 is half apex angle of the pinion rolling cone [°], u_1 is transmission ratio of first bevel stage, σ_{hlim} is contact fatigue stress of material [MPa].

According to this standard, the modul of bending for spur gear is determined by:

$$m_n = f_F \cdot \sqrt[3]{\frac{K_F M_{k1}}{\psi_m z_1 \sigma_{FP}}} \qquad (3)$$

where f_F is the bending coefficient for bevelled teeth, M_{k1} is the input shaft torque, ψ_m is the tooth width coefficient, z_1 is the number of pinion teeth, σ_{FP} is the permissible bending stress for the disappearing load.

The modul value of the contact stress for spur gear was determined by [21]:

$$m_n = f_H \cdot \sqrt[3]{\frac{K_H M k_{k1} (i+1)}{\psi_m \cdot z_1^2 \cdot i \cdot \sigma_{HP}^2}} \qquad (4)$$

where f_H is the coefficient for bevelled teeth subjected to contact, M_{k1} is the input shaft torque, ψ_m is the tooth width coefficient, z_1 is the number of pinion teeth, i is the ratio number, σ_{HP} is the permissible contact voltage for the disappearing load.

For the first stage of bevel gearing the transmission ratio was determined to be 3.55 and for the second stage, helical gearing transmission ratio was determined to be 3.15.

Modulus of the first stage for bevel gearing with eloid shape was calculated from bending and contact equations, and then the normalized value of the modul m_{nm}=7mm was chosen, with the modul on the outer front plane is m_{te} = 11.076mm.

For the second stage of helical gearing, the modulus was determined from the bending and contact equations on the basis of which the value of normalized modulus was chosen to be m_n=8mm.. The proposed modules were used for calculation of all needed gear wheels dimensions.

Lubrication of the bevel and helical gearing for the first and second stage upper bearings is ensured by means of pressured oil, which is integrated into the gearbox by a pump. The purpose of the pressure oil is not only to lubricate but also to cool the first and second gears of the gearbox. Lubrication of the bevel pinion bearing is by immersion in the oil filling and at the same time by injecting oil from the distribution from the pump.

Cooling of the gearbox is provided by an axial fan, which is located on the input shaft of the gearbox where the heat transfer to the surrounding area as well. The shaft seal of the gearbox is secured by rotary o-ring seals (Fig.4).

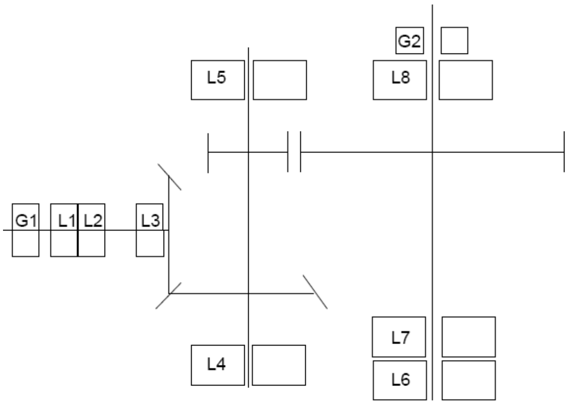

Figure 4. Scheme of gearbox bearings and seals.

The gearbox is tightly closed and can operate in dusty, humid and chemically harmless environments that do not degrade the oil filling and do not contaminate the sealing ability of the rotary seals. The gearbox housing is divided, and the connection surfaces are sealed with a sealant. The interior surfaces of the gearbox are painted with oil-resistant paint.

The inside of the gearbox is connected to the surrounding atmosphere by an air vent, which is in the pouring plug, to which it is necessary to connect an air hose with an outlet outside the diffuser.

Heating of the oil is achieved by means of a heating spiral (Fig.5). The temperature and quantity of oil in which the heater is immersed must be such that the temperature of the heater does not exceed a flash point of 10 °C. The heater must be switched off and cooled before removal.

Figure 5. Heater.

The purpose of the heating coils is to heat the oil in the gearbox to operating temperature, which makes the gearbox ready for 100% load at any given time.

If the temperature is lower than +5°C, the thermostat will switch on the heaters. Heaters are switched off after the oil reaches temperature of +10°C .

Finite element analysis of the gearbox was performed. The condition for the successful mastering of this issue is the creation of a computational model for solving the problems of static deformation analysis by the finite element method. The first step is to create geometric 3D model of gearbox body. The next step was to define the material properties of gearbox body. The material from which the gear wheel was made was replaced in the study for solving FEM tasks, by using material constants which characterize the material. Basic material properties such as Young's modulus of elasticity, the modulus of rigidity and Poisson's number were used. A finite element type of the Tetra type was used (Fig. 6).

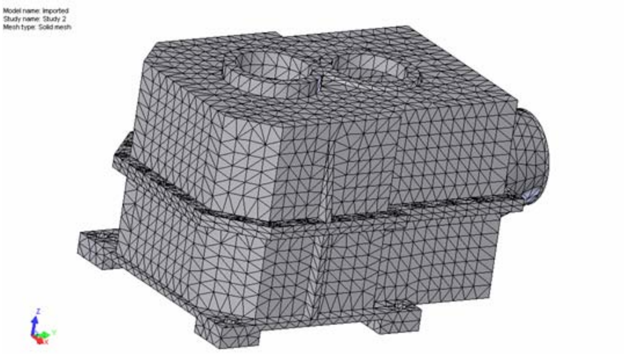

Figure 6. Finite element creation.

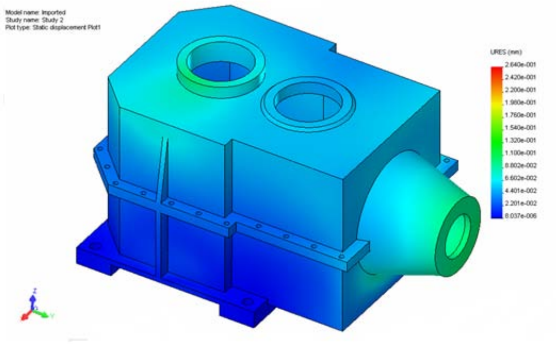

Figure 7. Deformation analysis.

Deformation analysis of the gearbox body confirmed the correctness of the design of the modifications. The gearbox housing complies.

4 Conclusion

The two-speed bevel gearbox ensures the transfer of mechanical energy from the driven machine to the driven machine. The gearbox is an important part of the cooling tower. To eliminate the fault, an analysis was performed, on the basis of which the optimization of this gearbox was proposed. The design of bevel gears and spur gears, the design of bearings, the design of special accessories for gearboxes intended for cooling towers, the design of gearbox lubrication and the analysis of the gearbox housing using the finite element method were performed. The proposed modifications have led to the elimination of failures.

Literature:

1. Miková, L. et al.: *Motion control of nonholonomic robots at low speed*. International Journal of Advanced Robotic Systems, 2020. Volume: 17 Issue: 1, p. 1-12.
2. Kuric, I. at al.: *Technical diagnostics at the department of automation and production systems*. 2019, Advances in Intelligent Systems and Computing , Volume: 835, p. 474-484.
3. Kaššay, P., Urbanský, M.: *Torsional natural frequency tuning by means of pneumatic flexible shaft couplings.* Scientific Journal of Silesian University of Technology.Series Transport, 2015, Vol. 89, p. 57-60. ISSN 0209-3324.
4. Juzek, M. et al.: *Non-Parallelism of Toothed Gear Shafts Axes as a Source of Gear Housing Vibration*. Transport Means - Proceedings of the International Conference, 2017, p. 601-606.
5. Kostka, J. et al.: *Stability Loss Analysis for Thin-Walled Shells with Elliptical Cross-Sectional Area*. Materials, 2021,14/19, pp. 1-22.
6. Jakubovicova, L. et al.: *Numerical analysis of stiffener for hybrid drive unite.* MATEC Web Conf., Volume 157, 2018, Machine Modelling and Simulations 2017, p. 1-9.
7. Virgala,I et al.*: Reconfigurable Wheel-Legged Robot*. MM Science Journal, MM Publishing, 2020, June, pp. 3960-3965, ISSN 1803-1269.
8. Kelemen, M. et al.: *A novel approach for a inverse kinematics solution of a redundant manipulator*. Applied Sciences, Basel , 2018, 8/11, pp. 1-20, ISSN 2076-3417.
9. Kopas, P. et al.: *Fatigue characteristics of welded high strength steel in the low cycle region of loading*. MATEC Web of Conferences, 2019. Volume: 254, p. 1-10.
10. Frankovský, P. et al.: *Application of holographic interferometry in the analysis of stress states in a crack root area*. Applied Optics. The Optical Society, 2020. 59/13, pp. 170-178, ISSN 1559-128X.
11. Sapietova, A. et al.: *Dynamic and Stress Analysis of a Locking Mechanism in the Ansys Workbench Software Environment*. Advances in science and technology – research journal. 2019, Volume: 13, Issue: 1, p. 23-28.
12. Maláková, S. et al: *Meshing Stiffness-A Parameter Affecting the Emission of Gearboxes*. Applied sciences, 2020. Volume: 10/23, p. 1-12.
13. Kelemenová, T. et al.: Specific Problems in *Measurement of Coefficient of Friction Using Variable Incidence Tribometer*. Symmetry, Basel, Multidisciplinary Digital Publishing Institute, 2020, 12/8, pp. 1-20, ISSN 2073-8994.
14. Sivák, P. et al.: *Influence of different strain hardening models on the behavior of materials in the elastic–plastic regime under cyclic loading.* Materials. 2020, 13/23, pp. 1-20, ISSN 1996-1944.
15. Homišin J, et al.: *Electronic Constant Twist Angle Control System Suitable for Torsional Vibration Tuning of Propulsion Systems*. Journal of Marine Science and Engineering. 2020, Voume 8, no 9:721, pp. 1-17.
16. Sapieta, M. et al.: *Detection of Elastic Deformation in Metal Materials in Infrared Spectral Range.* Materials 2021, 14(18), 5359; https://doi.org/10.3390/ma14185359.
17. Kopas, P. et al.: *Fatigue characteristics of welded high strength steel in the low cycle region of loading*. MATEC Web of Conferences, 2019. Volume: 254, p. 1-10.
18. Sapietková A et al. *Simplified computation methodology for contact forces on tapered rolling bearing with flexible parts.* Scientific Journal of Silesian University of Technology.Series Transport, 2018, 99, pp. 177–182.
19. Urbanský, M, Homišin, J., Kaššay, P., Krajňák, J.: *Measurement of air springs volume using indirect method in the design of selected pneumatic devices.* Acta Mechanica et Automatica. 2018, Vol. 12, no. 1, p. 19-22. ISSN 1898-4088.
20. Jakubovicova, L., Sapietova, A., Moravec, J.: *Static analysis of transmission tower beam structure.* MATEC Web Conf., Volume 244, 2018, pp. 1-9.
21. Slovak national standard STN 01 4686 Strength calculation of gears, Praha, (1975).

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