ISSUES CONCERNING THE INCREASE OF THE UPTIME FOR THE CYLINDRICAL GEAR THAT IS PART OF THE ROTATING SYSTEM OF EsRc 1400 BUCKET WHEEL EXCAVATOR

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Abstract: This paper presents the technical solutions leading to an increased uptime of the outer cylindrical gear, part of the mechanical transmission that delivers the rotation of the upper platform of the bucket wheel excavator. The solutions adopted are justified by analytical and finite element calculations performed for the pinion and gear wheel.

Keywords: bucket wheel excavator, rotating mechanism, outer cylindrical gear, pinion, gear wheel, finite element

1. GENERAL CONSIDERATIONS

EsRc 1400-30/7 bucket wheel excavators is designed to work in lignite quarries, for excavating sterile or coal, ensuring concurrently the uploading of the material on large belt conveyors.

The rotating mechanism is a subassembly, component part of the excavator, which ensures the rotation of its upper platform along with the rotor equipped with buckets and cutting teeth that dislodge the material from the working face.

Obsolete constructive solutions of component subassemblies of the excavator, which, of course, are also morally outdated, lead to the need for their revamping, including of the rotating mechanism which includes the mechanical transmission which is the subject of this paper.

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2. CONSTRUCTIVE SOLUTION OF THE MECHANICAL TRANSMISSION

Figure 1 shows the solid representation of the mechanical transmission comprising the bearing attack pinion and wheel gear with upper tread of the ball bearings, which connects the drive group and the lower part of the rotating (pivoting) part of the upper structure the excavator.

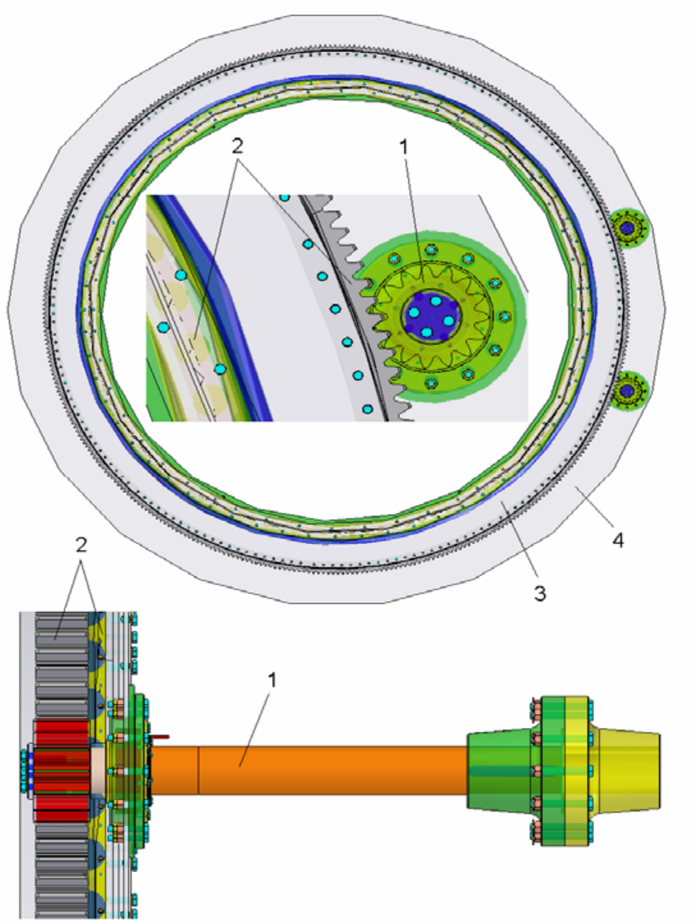


Fig. 1. Mechanical transmission comprising the pinion and the gear wheel:
1 - attack pinion with bearing, 2 - gear wheel with bearing, 3 - chassis, 4 - pivoting part

Figure 2 shows the overall construction of the cylindrical attack pinion with spurs together with the bearing for its attachment to the system. The pinion crosses throughout construction for the support of the platform and rests on the lower horizontal element of the platform support provided with an adjustable bearing.

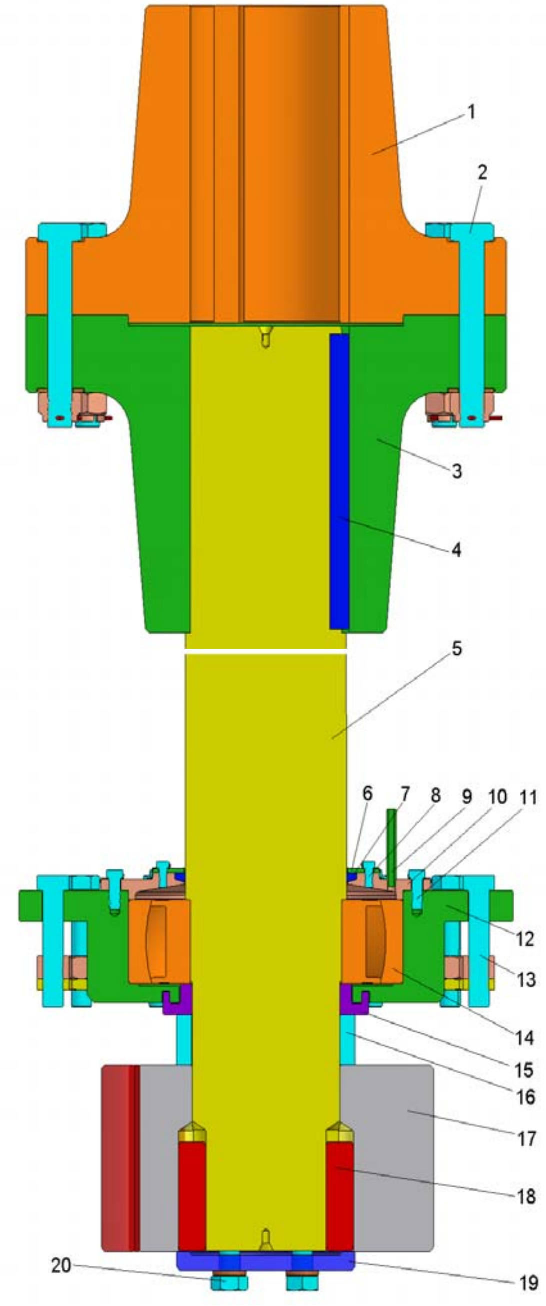


Fig. 2. Overall construction of the bearing attack pinion:
1 - flanged semi-coupling mounted on the shaft of the cylinder-conical reducing gear of the driving group; 2 – M 36×240 special screw; 3 - flanged semi-coupling mounted on the shaft of the pinion; 4 - 18×11×380 parallel wedge; 5 - pinion shaft; 6 - two pieces cover for the sealing ring made of felt; 7 - felt ring; 8 – M 12×20 screw;
9 - G ¼ pipe for the forced anointing of the bearing; 10 - bearing cap; 11 - M16×40 screw; 12 - support of the bearing;
13 - M30×140 screw with normal and low nut for fastening on the rotating side of the excavator; 14 - oscillating radial bearing with spherical rollers on two rows, series 22244; 15 - labyrinth flanges; 16 - spacer ring; 17 - attack cylindrical pinion with spurs z = 15, m = 27; 18 – 4 Ø40×140 cylindrical pins for torque transmission;
19 – cover to fasten the pinion to the shaft;
20 – 4 M 30×70 screws with Grower washer for fastening the cover.

Figure 3 presents the crown gear that is part of the gear for transmitting the motion to the rotating structure of the excavator; it includes nine segments. Division diameter of the crown gear is 10206 mm.

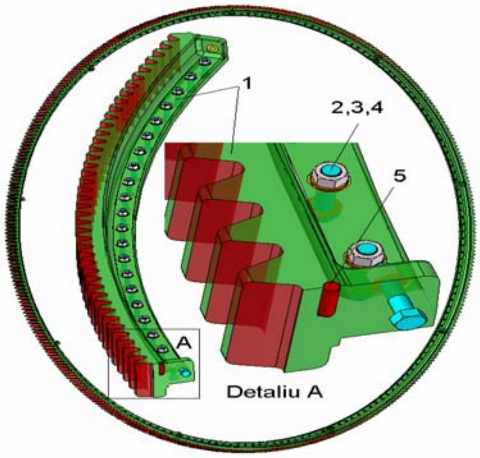


Fig. 3. 3D model of the crown gear z = 378, m = 27:
1 - segment of crown gear, nine pieces;
2 – M 24×95 special screw (distance on Ø 25 H7), 207 pieces;
3 – plane washer; 4 – M 24 nut; 5 - Ø20×70 cylindrical pin

3. ANALYTICAL CALCULATION OF THE TRANSMISSION COMPRISING THE PINION AND CROWN GEAR

Analytical calculation of the outer cylindrical gear of the transmission relies on the documentation for the manufacturing of the pinion and of the gear wheel and its final aims is to determine the variation of the safety factors depending on the uptime of the gear; subsequently, one may rethink new solutions to increase safety in operation for the analyzed mechanism.

The calculation involves: geometric calculation of the gear, which includes calculation of geometric elements and of measurement dimensions and of verification of spurs, calculating the resistance of cylindrical gear, which includes checking the fatigue by contact of the spurs sides and verification to fatigue by bending the spurs.

The conclusions emerged from the verification calculation are summarized in the two groups of diagrams in Figures 4 and 5, which expresses the dependence between the safety factor for contact and bending fatigue and uptime, for different hardness and roughness of the spurs sides.

The following aspects result from the presented charts:

1. For a 35 kW drive power, a hardness of sidewalls of 53...56 HRC for alloy steel for improvement and a roughness of 3.2 mm, the pinion has a very short lifetime. For higher values of lifetime, both at the strain by contact fatigue, and by bending, the safety factors situate below acceptable values;

2. For the same drive power and roughness of sidewalls, but for a hardness of 56...63HRC for cementing steels, the lifetime of the pinion increases quite a lot, the values of safety factors being above acceptable values, even after 10 000 hours of operation.

The final conclusion is that it is necessary to reconsider the mark of material, as well the technology used for the manufacturing of the pinion, i.e. it is necessary to adopt a cementing steel in order to provide a hardness of the spur sidewalls of 56...63 HRC and their roughness should be at least 3.2 mm.

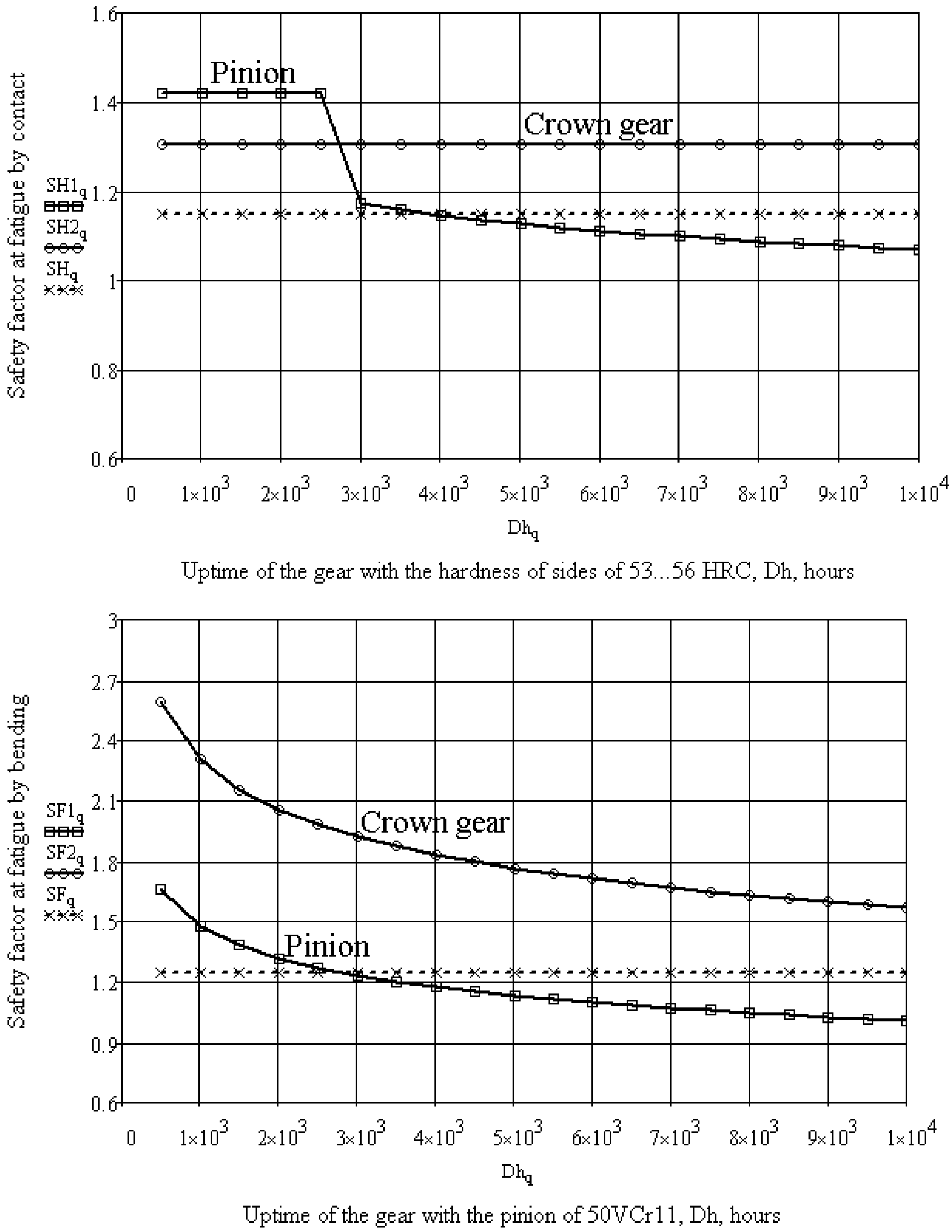


Fig. 4. Safety factors for the current gear at P_m=35 kW and R_a=3,2 µm

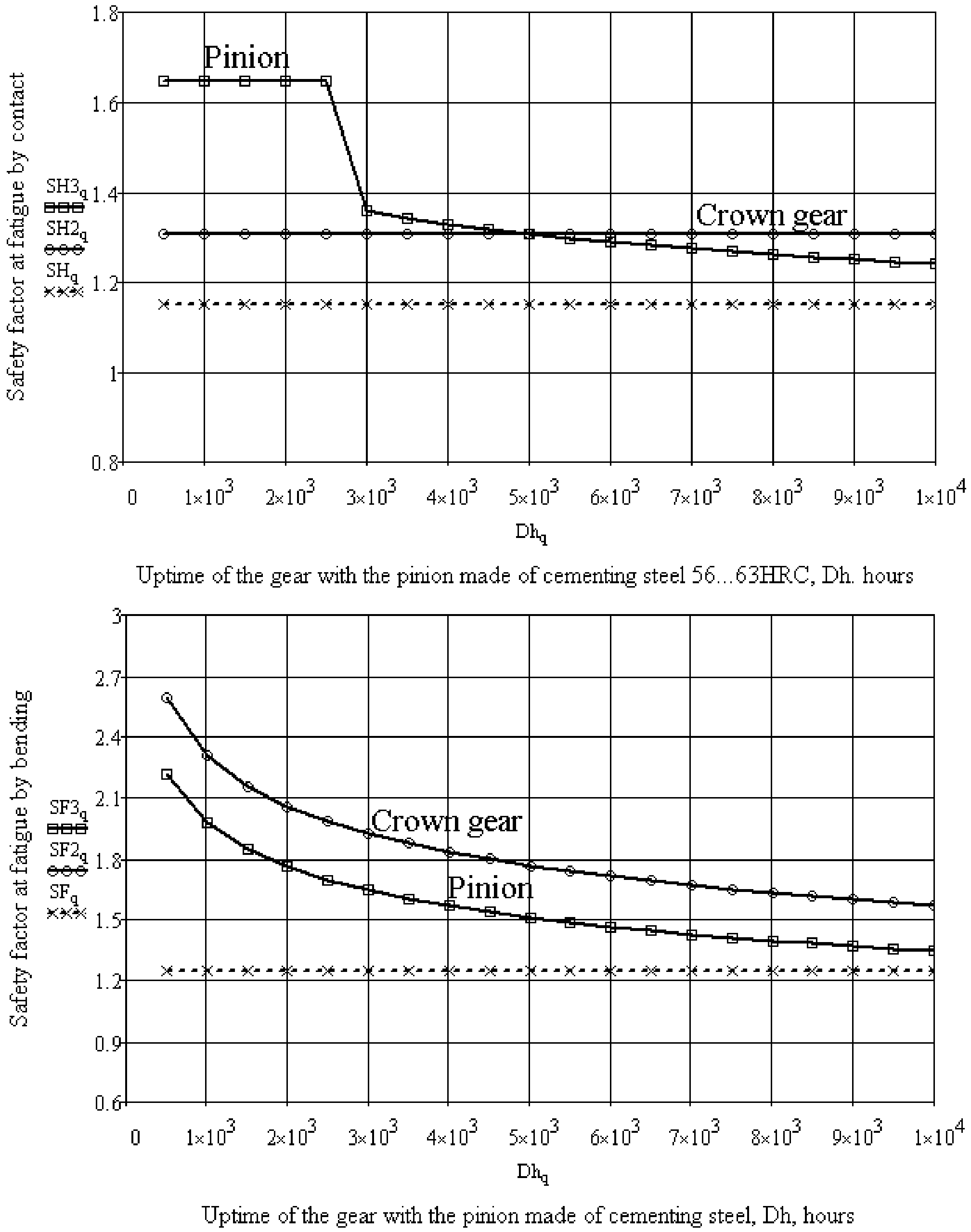


Fig. 5. Safety factors for pinion made of cementing steel at P_m=35 kW and R_a=3,2 μm

4. CALCULATING THE TRANSMISSION COMPRISING THE PINION AND CROWN GEAR BY THE METHOD OF FINITE ELEMENTS

4.1. Attack pinion gear

To determine the strains and shifts for the attack pinion gear, finite element analysis model was fitted in the flange semi - coupling without translation in the area of the bearing. Figure 6 presents the variation of equivalent stresses in the attack pinion. Loading capacities are the torque M_t=123150 Nm and the tangential force F_t=606490 N and radial force F_r=225850 N. For these applications there has resulted a maximum stress in the area of stress concentrator produced by the sudden increase of diameter near the bearing. A stress of 600.54 MPa resulted in the pinion at the lower part of the spur, which is close to that determined by analytical means, i.e. 561.96 MPa. Deviation occurs because of the way of discretization the geometrical model and the analytical calculation is made with the normal force applied on the sidewall of the spur.

Figure 7 shows the variations of normal and tangential stresses and of shifts occurring in the pinion.

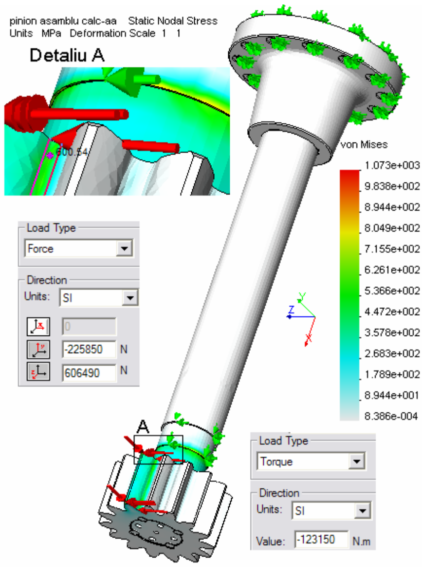


Fig. 6. Variation of equivalent stresses in the pinion

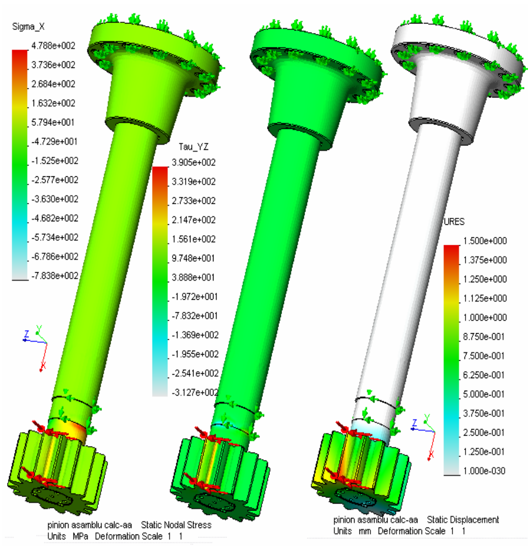


Fig. 7. Variations of normal and tangential stresses and of shifts occurring in the pinion

From this figure it results that the maximum shift occurs at the peak of the spur and towards the lower part of the pinion, with a maximum of 1.5 mm.

This aspect is shown by the wear of the pinion, as shown in Figure 8.

Fig. 8. Wear of the attack pinion

4.2. Crown gear

Finite element analysis model was fitted into holes bored holes Ø25H7 and end hole. Loading capacities are the torque M_t=123150 Nm and the tangential force F_t=606490 and radial force F_r=225850 N.

Figure 9 shows the variation of equivalent stresses in the segment of crown gear. For these stresses there resulted a maximum strain in the area of the stress concentrator due to the counterbore of the end hole. A tension of 732.5 MPa resulted in the pinion at the lower part of the spur, which is close to that determined by analytical means, i.e. 622.96 MPa.

Figure 10 shows the variations of normal and tangential stresses and the shifts of the segment of crown gear.

Maximum shift occurs at the tip of the spur and towards the lower part of the segment of crown gear with a maximum of 0.62 mm, and this aspect confirms the wear of the pinion.

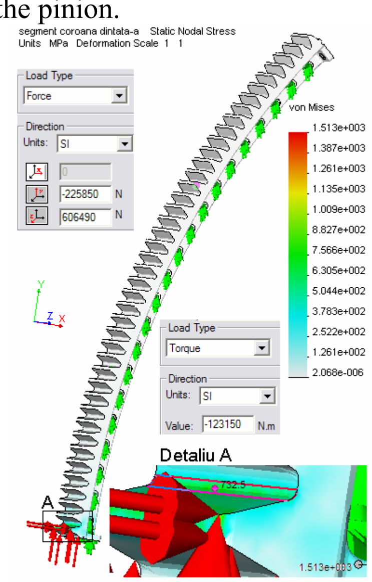


Fig. 9. Variation of equivalent stresses in the segment of crown gear

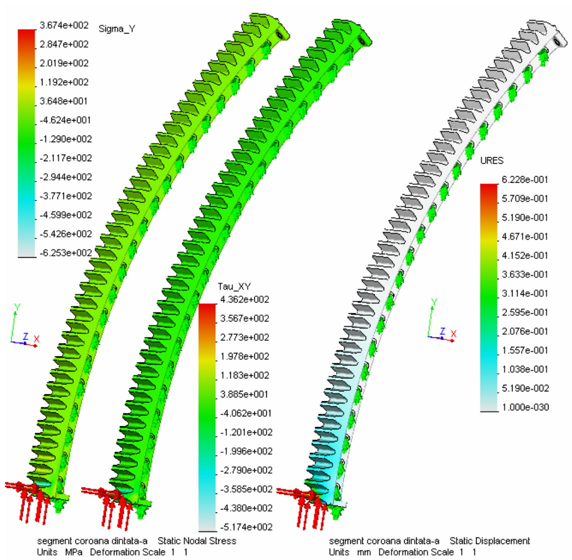


Fig. 10. Variations of normal and tangential stresses and of shifts occurring in the segment of crown gear

5. CONSTRUCTIVE IMPROVEMENT OF THE GEAR

Since the power of the electric motor that drives the rotating mechanism increased from 35 to 45 kW, there was performed a verification calculation for contact fatigue of the sidewalls of spurs and for bending fatigue of spurs at the power of 45 kW.

The results are quantified by interpreting the diagrams of dependence between the safety factors at contact fatigue and bending fatigue depending on the lifetime of the pinion and crown gear for different brands of materials and different roughness of the sidewalls; these diagrams are shown in Figures 11...13.

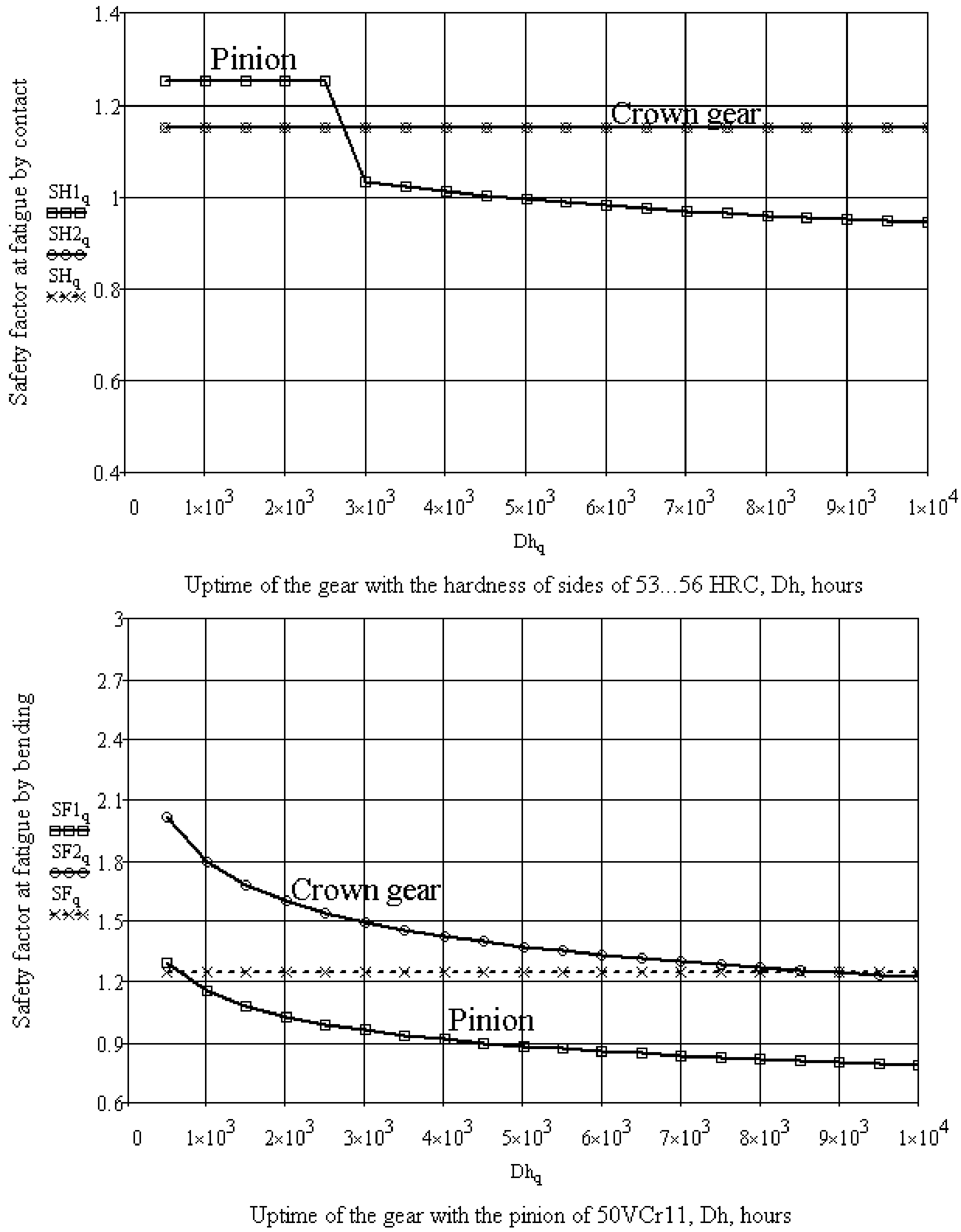
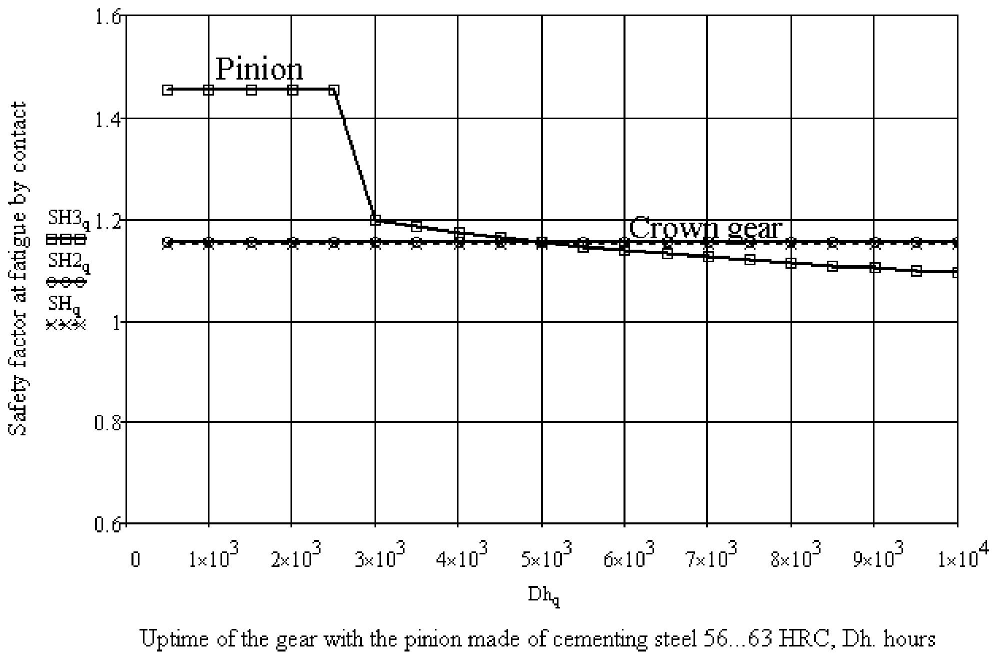


Fig. 11. Safety factors for the current gear at P_m=45 kW and R_a=3,2 μm



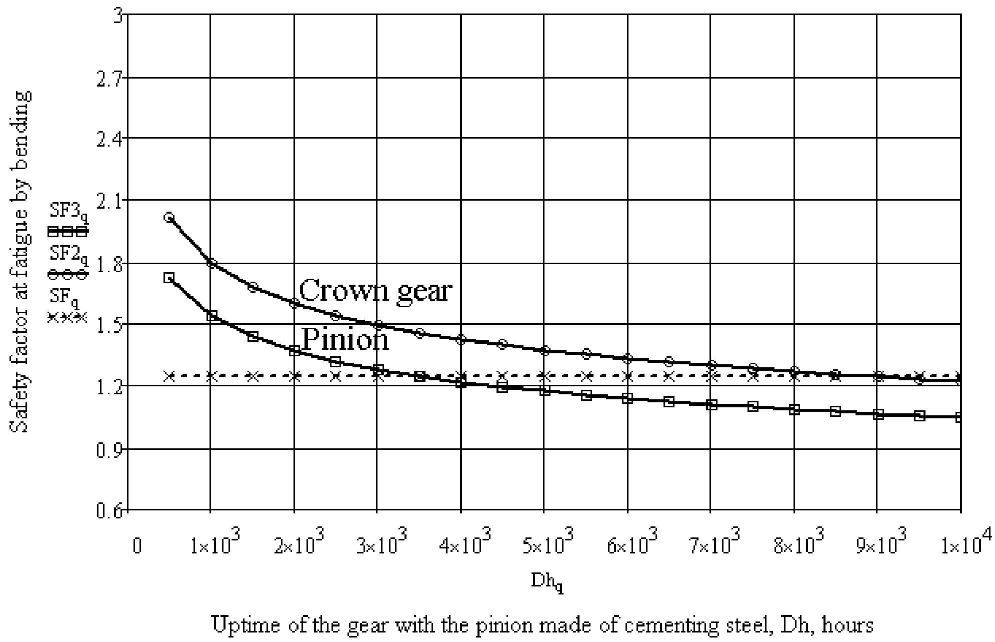
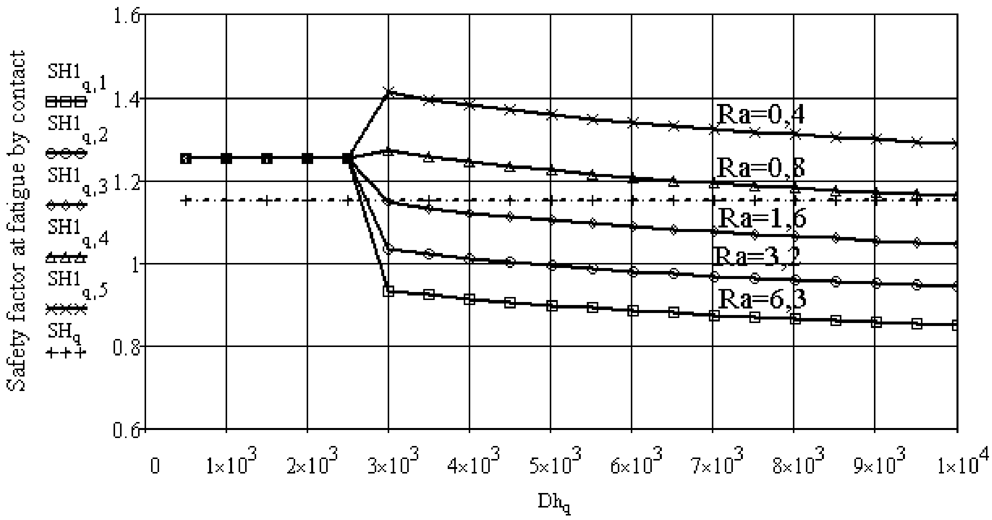
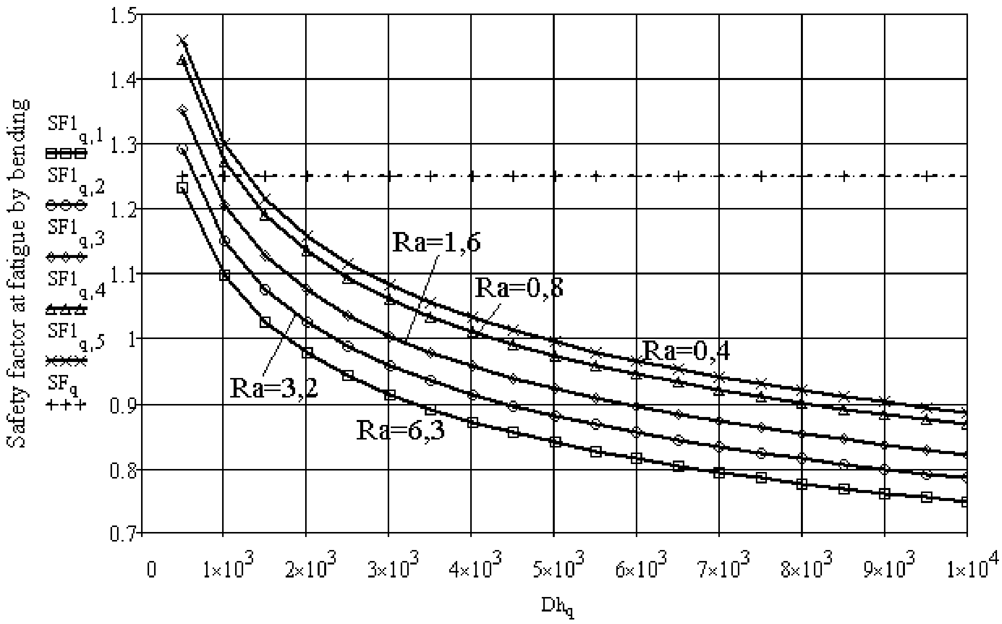


Fig. 12. Safety factors for the pinion made of cementing steel at P_m=45 kW and R_a=3.2 μm



Uptime of the gear with the hardness of sides of 53...56 HRC, Dh, hours



Uptime of the gear with the pinion of 50VCr11, Dh, hours

Fig. 13. Safety factors for the pinion made of cementing steel at P_m=45 kW and R_a=0.4 … 6.3 μm

Analyzing the diagrams resulting from the calculations, we can draw the following conclusions:

- for the brand of material used for improvement, which provides a hardness of the sidewalls of 53...56 HRC, material used for the pinion, and at a roughness of 3.2 mm, the safety factor, both for contact fatigue and bending fatigue, situates below the permissible value, aspect that leads to the need to reconsider the brand of material;
- for the same type of material used for improvement, even if correction of sidewalls is made so that very low roughness are obtained, the safety factors situate of below the permissible value, especially during the test to bending fatigue of the spur;
- the pinion should be made of an alloy cementing steel that provides a hardness of sidewalls of 56...62 HRC and the roughness of spurs should be 1.6 mm at least. In these conditions, it is possible to provide a lifetime for the pinion up to 6000 hours, without showing pronounced wear.

6. CONCLUSIONS

After analyzing the mechanical transmission from the construction of rotating mechanism of the rotor excavator, one can conclude that the current structure is not appropriate, both for the power of 35 kW, but mostly for the power of 45 kW. Calculations, both by analytical and finite elements means, led to the conclusion that it is necessary to adopt an alloy cementing steel to ensure hardness for the sidewalls of spurs 56...62 HRC, at a roughness of 1.6 mm at least, which guarantees a lifetime of 6000 hours.

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