Static analysis and strength calculation of drive shaft of large-scale cone crusher

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Abstract. The paper analyzes the causes of failures of coarse crushers "KKD 1500-180" at JSC "Almalyk MMC". It has been established that the most common cause of these failures is a drive shaft failure. Static analysis of the strength of the drive shaft was performed using Autodesk Inventor 2020 CAD software. A dangerous shaft section was found, and recommendations aimed at improving its operability were proposed.

1 Introduction

At the present stage of the development of innovative technologies [1-15], it is necessary to conduct research aimed at improving the reliability of technological equipment [16-29]. Improvement of settlement and analytical and design solutions. One of the urgent tasks of modern mechanical repair production is to assess the load and deformations of various parts that are subject to frequent restoration, repair or new manufacture.

Extensive experience in operating large-size cone crushers "KKD 1500-180" at JSC "Almalyk Mining and Metallurgical Combine" indicates frequent equipment downtime due to emergencies associated with the loss of operability of large-sized parts. One of the problematic parts of this equipment is the drive shaft.

2 Analytical calculations

To identify the most loaded sections of the crusher drive shaft, a static analysis and strength analysis were performed using Autodesk Inventor 2020 software. Strength criterion is one of the most significant operability criterion of machine parts [30-40].

The initial data used for conducting static analysis:

- drive shaft of a step-type crusher with diameters from 158 to 200 mm and a length of 3121 mm;
- mass of the drive shaft, m = 710 kg;
- electric motor power P = 355 kW;
- rotation speed n = 1500 rpm.

Rotational motion is transmitted from the electric motor through the clutch to the shaft, then the crusher's drum operates through the bevel gear.

Torque is determined by the formula:

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$$M_r = \frac{P}{\omega} = \frac{30P}{\pi n} = \frac{30 \cdot 355}{3.1416 \cdot 1500} = 2.26 \text{ kNm} \quad (1)$$

where ω – is the angular velocity.

To carry out a static analysis, we determine the forces appearing on the shaft during rotation.

We calculate the forces acting from the clutch:

$$F'_\tau = \frac{M_r}{r_c} = \frac{2260}{0.25} = 9040 \text{ N} \quad (2)$$

$$F'_n = 0.4F'_\tau = 3616 \text{ N} \quad (3)$$

$$F' = \sqrt{\left(F'_\tau\right)^2 + \left(F'_n\right)^2} = 9736 \text{ N} \quad (4)$$

We determine the forces on the shaft section under the gear:

$$F''_\tau = \frac{M_r}{r_g} = \frac{2260}{0.405} = 5580 \text{ N} \quad (5)$$

$$F''_n = 0.4F''_\tau = 2233 \text{ N} \quad (6)$$

$$F'' = \sqrt{\left(F''_\tau\right)^2 + \left(F''_n\right)^2} = 6010 \text{ N} \quad (7)$$

We constitute the equations of forces equilibrium and determine the prop reactions *R*' and *R*'' (figure 1):

$$\sum M'_y = R' \cdot 1.805 - mg \cdot 0.9025 + F''_n \cdot 0.3325 - F'_n \cdot 2.536 = 0 \quad (8)$$

$$R' = 1190 \text{ N} \quad (9)$$

$$\sum M''_y = -R'' \cdot 1.805 + mg \cdot 0.9025 + F''_n \cdot 2.1375 - F'_n \cdot 0.731 = 0 \quad (10)$$

$$R'' = 7591 \text{ N} \quad (11)$$

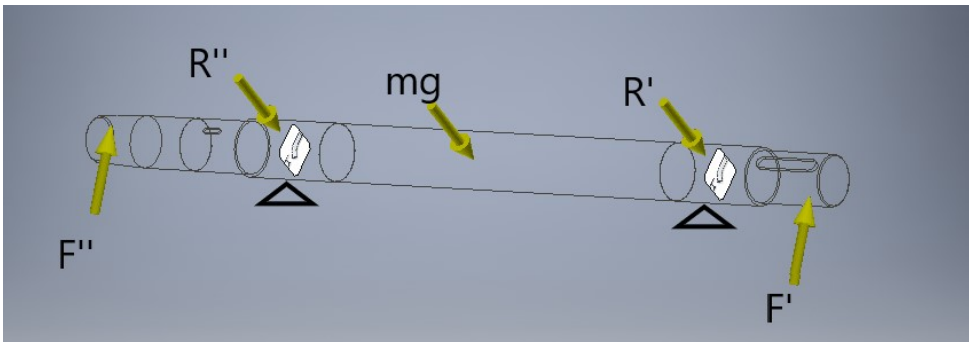

Fig. 1. Forces acting on the shaft.

We will verify the correct determination of forces:

$$mg + R'' + R' - F'' + F' = 0 \quad (12)$$

$$6965 + 7591 + 1190 - 6010 + 9736 = 0 \quad (13)$$

The equilibrium condition is satisfied, therefore, we can further use the calculated forces acting on the shaft for static analysis.

3 CAE calculations

The result of calculating the stresses according to Mises is presented in figure 2.

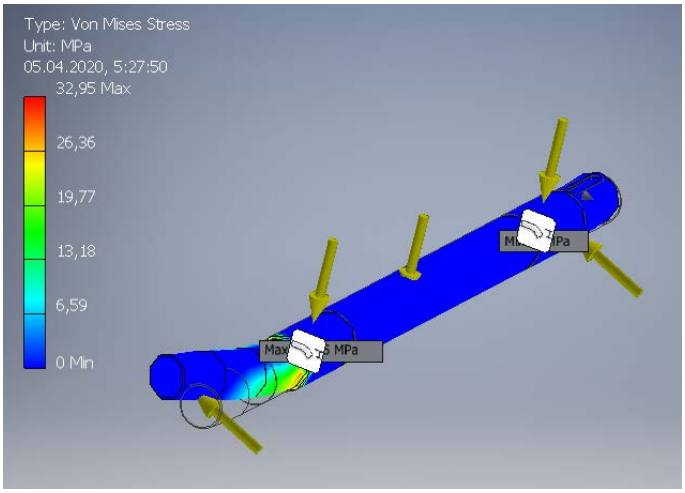

Fig. 2. Von Mises stress.

Figure 3 shows the maximum displacement of drive shaft.

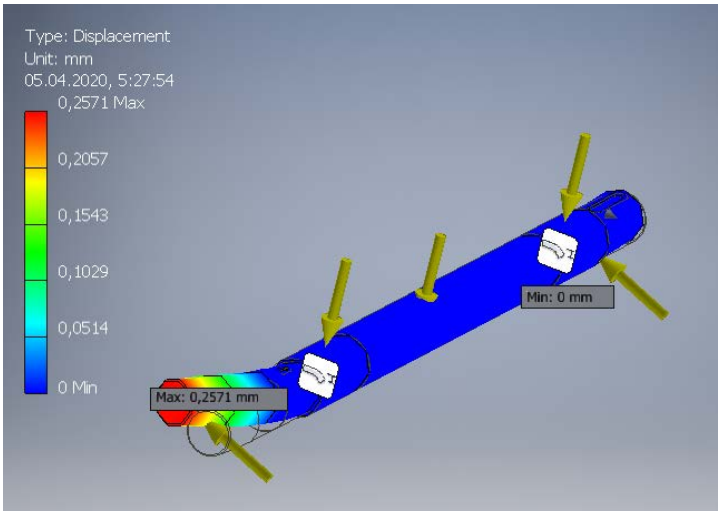

Fig. 3. Displacement.

Figure 4 shows the equivalent deformation.

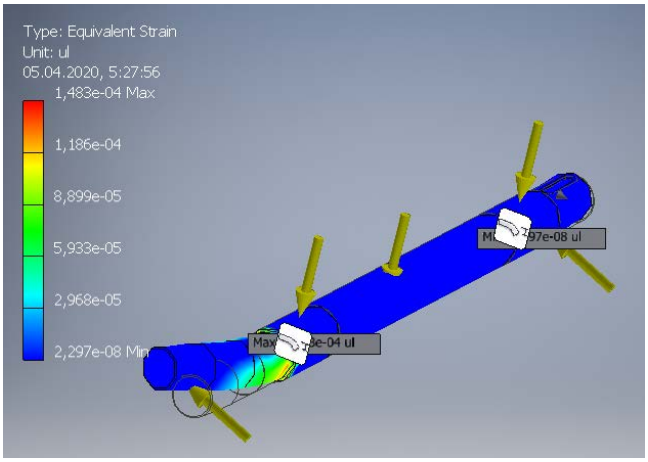

Fig. 4. Equivalent strain.

The numerical results of the analysis performed in Autodesk Inventor 2020 are presented in the table 1.

Table 1. Crusher drive shaft static analysis results

Name	Minimum	Maximum
Volume	90494800 mm^3	
Mass	710,384 kg	
Von Mises Stress	0,00450251 MPa	32,9513 MPa
1st Principal Stress	-5,99087 MPa	32,5363 MPa
3rd Principal Stress	-39,0433 MPa	2,60024 MPa
Displacement	0 mm	0,257126 mm
Safety Factor	10,6217 ul	15 ul
Stress XX	-32,4949 MPa	32,3698 MPa
Stress XY	-6,23733 MPa	11,9029 MPa
Stress XZ	-5,47142 MPa	17,3469 MPa
Stress YY	-12,2677 MPa	6,76512 MPa
Stress YZ	-1,73897 MPa	2,00225 MPa
Stress ZZ	-12,3199 MPa	6,42019 MPa
X Displacement	-0,0415655 mm	0,0416311 mm
Y Displacement	-0,00227041 mm	0,0892219 mm
Z Displacement	-0,000196482 mm	0,23855 mm
Equivalent Strain	0,0000000229683 ul	0,0001483 ul
1st Principal Strain	0,0000000102571 ul	0,000155541 ul
3rd Principal Strain	-0,00016906 ul	-0,0000000181391 ul
Strain XX	-0,000150454 ul	0,000154466 ul
Strain XY	-0,0000402308 ul	0,0000767734 ul
Strain XZ	-0,0000352906 ul	0,000111888 ul
Strain YY	-0,000035485 ul	0,0000374621 ul
Strain YZ	-0,0000112163 ul	0,0000129145 ul
Strain ZZ	-0,0000464921 ul	0,0000438258 ul

4 Summary

The paper presents the results of a study aimed at improving the operability of crushers using CAD. In accordance with the results obtained, it was found that the neck of the shaft, in the place where the liner is installed, is a dangerous section.

It is proposed to change the shaft design, increasing the diameter of the shaft in its dangerous section and thereby increasing its strength.

It is recommended to significantly reduce interruptions in the operation of crushers, since when starting and stopping the machine there are uneven peak loads that reduce its strength.

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