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THEORETICAL DETERMINATION OF THE TECHNOLOGICAL AND POWER PARAMETERS OF A CONE CRUSHER TO SET BOUNDARY CONDITIONS FOR A STATIC STUDY OF THE "ROLLING CONE"

IVAN STEFANOV MININ
and TEODORA VASSILEVA HRISTOVA

University of Mining and Geology
St. Ivan Rilski
1700 Sofia
Bulgaria
e-mail: minin@abv.bg
teodora@mgu.bg

Abstract

The main purpose of this research is to determine a suitable class fraction of the metal composition of the ore fragments and the maximum detection of the minerals in them which is accomplished through the crushing and milling processes. In these processes, the product is being shattered by the action of external mechanical forces. The shattering processes are most energy-intensive and expensive in most factories - 35-40% (sometimes up to 60%) of capital and operating costs. Also, crushing has lower energy absorption than milling. Therefore, the aim is to achieve optimal crushing and definitely little use of the milling processes. Despite the reduced energy consumption, the cost of maintaining the crusher components is high. In the present study, a new

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method for determination of the forces and reactions in a "KUBRIA" 210/35 cone crusher was used to calculate and to set the boundary conditions in the static examination of the main "rolling cone" knot by using "Solid Works" software.

1. Introduction

Capital equipment projects are typically managed on the basis of operating costs, technical obsolescence, performance improvement, functionality, and safety requirements. Most of the capital replacement solutions are taken according to engineering and economic safety requirements [1] and lack of fatigue. Equipment failures lead to a reduction in staff safety and temporary discontinuation of the production process, thus generating losses. To increase and ensure safety, it is necessary to determine the maximum allowable loads of the components of the equipment in order to reach the maximum operating life of the specific equipment. Hence, the maximum allowable load on a crusher for medium crushing was determined and the case study was done for a "KUBRIA" 210/35 cone crusher. This objective was due to the fact that several cases of machine shaft breaks occurred at the ore processing plant.

2. Construction Features of Crusher "KUBRIA" 210/35

In order to determine the locations with extreme tension zones and their safety factor, the crusher mechanism must first be considered. The shape of the crushing chamber is made of cones with non-steep forming and a parallel zone for the calibration of the crushed product. The material enters and passes through the crushing chamber formed between the movable 3 and the fixed cone 2 under its own weight as shown in Figure 1. The moving cone is driven by the cam 9 using of a conical gear 4, the input shaft 7, and a wedge belt pulley. The high-pressure hydraulic cylinder 5 by the piston 11 supports the weight of the cone and the vertical component of the crushing force. The self-aligning hydrostatic bearing 6 transmits the vertical reaction from the movable cone 3 of the support piston 11 and provides kinematic freedom and the definite movement of the rolling cone [2].

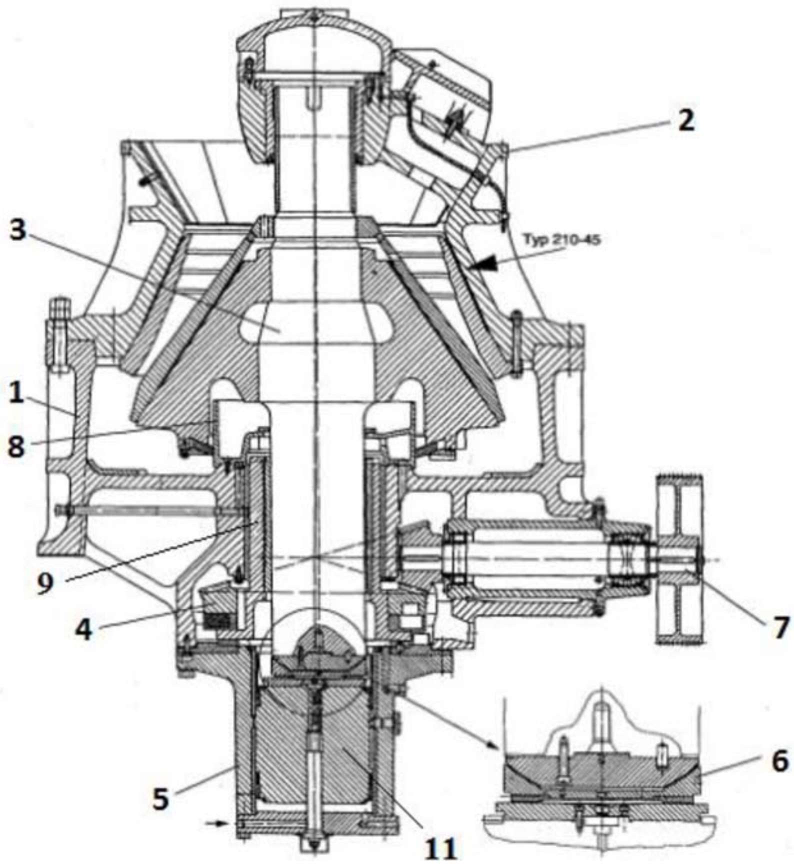

Figure 1. Conical crusher for medium crushing with two-sided movable cone type "KUBRIA" 210/35 [2].

3. Methodology for Determining the Basic Technological Parameters of the Crusher

Critical regions depend on the stress level which is created by the forces applied to them. Also, the determination of these critical zones is based on the methodology of the basic technological parameters and forces [3]. These are discussed below:

3.1. Determination of the angle of hookup

The condition for selecting the angle of the grasp of cone crushers is the same as for jaws. It is given with the expression as:

$$\alpha_1 + \alpha_2 \leq 2\rho, \tag{1}$$

where ρ is the angle of friction $(\rho = arctg\,\mu)$ and μ is the coefficient of friction between the ore and the lining of the cones. The angle of grabbing was chosen to be much larger in comparison with the jaw crushers and it was considered $\alpha \approx 26^\circ$.

3.2. Frequency of vibration of the movable jaw

The condition to obtain a crushed product with a maximum particle size equal to the width of the parallel zone is that each particle is at least once smashed in that area. This requirement implies that the time for one oscillation is less than the time of movement of the material in the parallel zone [4]. The frequency of vibration of the movable cone, which coincides with the eccentric rotation frequency and satisfies the above condition, is determined by the formula:

$$z = \frac{1}{2} k \sqrt{\frac{g.(tg\alpha_1 + tg\alpha_2)}{2s}} s^{-1}, \tag{2}$$

where:

α_1, α_2 are the angles of inclination of the formation of the cones;

g is the Earth's acceleration;

s is the movement of the vibration cone of the crusher in meter;

κ is a coefficient that measures the retardation of the material in the discharging zone due to the presence of friction forces and is equal to 0.5/0.9.

The flicking frequency of the cone per minute can be determined by the formula:

$$n = 60.z,\ \mathrm{min}^{-1}. \tag{3}$$

3.3. Productivity volume

If the time for one vibration of the moving cone is equal to the time that the particle travels through the length of the parallel zone, then each rotating of the crusher eccentric will unload rotary material with the form of a ring. Therefore, the production volume will be calculated as follows:

$$Q_\nu = 3600\pi.z.k_P.D\frac{(2b+s)s}{2(tg\alpha_1 + tg\alpha_2)},\ \mathrm{m}^3/h, \tag{4}$$

where z is the frequency of vibration of the movable cone, k_P is a coefficient of the bulk density of the ore, D represent the diameter of the movable cone, parameters of s, m are the action of the moving cone, and b is the width of the discharging port of the crusher.

Since the crusher has a range of regulation, $b_{\min} \le b \le b_{\max}$, it is necessary to calculate the Equation (4) twice under different conditions including minimum and maximum values of b.

3.4. Productivity mass

This parameter may be calculated by using Equation (5):

$$Q = \rho.Q_\nu,\ t/h, \tag{5}$$

where ρ is the density of the crushed product.

3.5. Determining the required power of the driving engine using Bond's hypothesis

The formula for determining the engine power according to Bonds hypothesis is presented as follows:

$$N_{im} = \frac{k_{cs}.Q.W_0}{10^2\eta_M\sqrt{D_t}}\left[\sqrt{\frac{D_t}{d_t}} - 1\right].\mathrm{kW}, \tag{6}$$

where:

k_{cs} : the crushing stage coefficient;

W_0 : the specific energy consumption of the ore, kWh/t;

D_t : the maximum size of the incoming pieces in the crusher, m;

η_M : is the mechanical efficiency of the motor of the crusher;

d_t : the maximum size of the crushed pieces, m.

4. Methodology for Determining the Basic Crush Power Parameters

The list of the resistance forces acting on the cone (Figure 2) are as follows:

P_t : a distributed crushing force whose equal acting is perpendicular to the forming cone and is applied at a distance of 1/3 of the length measured from the lower end;

P_X : the reaction in the upper bearing of the moving cone applied in the middle of the bearing perpendicular to its axis;

P_E : an eccentric node reaction applied in the middle of the bearing perpendicular to the axis of the cone;

P_Y : a spherical bearing reaction directed along the axis of the cone, perpendicular to the spherical surface;

G_K : the weight of the cone applied in its mass center.

It must be assumed that these forces line in a plane passing through the axis of the cone and conveying an angle γ with the direction of the eccentricity measured in the direction of the eccentric rotation. The normal bearing surface reactions also give rise to the corresponding frictional forces T and F_T. The average values of the forces and moments under the established operating mode of the crusher will be determined according to the following dependencies:

4.1. Determination of the reduced engine torque relative to the eccentric axis

This moment was determined as follows:

$$M_E = \frac{3000.\eta_M.N_{im}}{\pi.n}, \text{ kNm,} \tag{7}$$

where N_{im} is the power of the engine driving the crusher, η_M is the mechanical efficiency of the motor of the crusher, and n is the theoretical rotation speed of the eccentric shaft.

4.2. Determining the reaction of the eccentric

This reaction was determined by using Equation (8):

$$P_E = M_E \frac{M_E - G_E.\mu_3.R_3}{e_1.\sin\gamma + \mu_1(R_1^{"} + R_2^{"})}, \text{ kN,} \tag{8}$$

where G_E is the weight of the eccentric, μ_3 and μ_1 are the coefficient of friction in the heel-formed bearing and the coefficient of friction in the eccentric, respectively. Parameter of e_1 representing the eccentricity, R_3 is the radius of the eccentric cup, and symbols of $R_1^{"}$, $R_2^{"}$ are the internal radius and external radius, respectively (Figure 2).

4.3. Determining the crushing strength

This variable may be calculated as follows:

$$P_T = \frac{P_E.l_E - G_K.a}{l_1.\sin\beta + l_2\cos\beta}, \text{ kN,} \tag{9}$$

in which:

G_K is the weight of the movable cone, l_E is the distance from the upper bearing to the center of the eccentric, β is the angle of the movable cone, $a = e_1\frac{l_3}{l_E}$ represent the intermediate eccentricity, and symbols of l_1, l_2, l_3 are the length values which are shown in Figure 2.

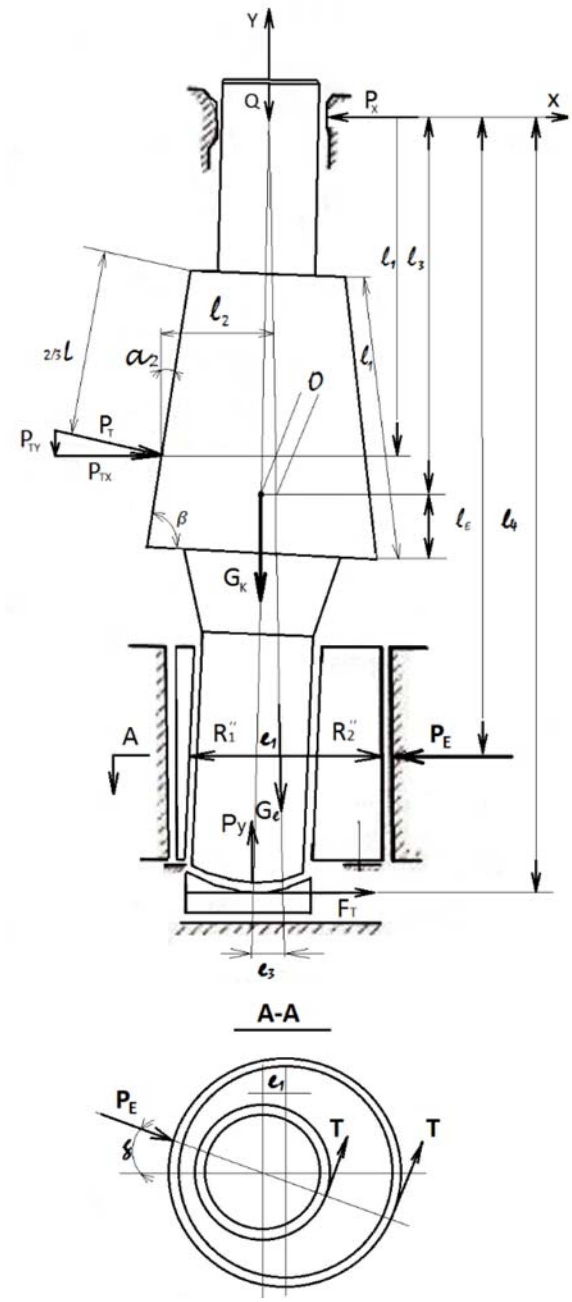

Figure 2. Kinematic scheme of the crusher “KUBRIA” 210/35 [2].

4.4. Determining the spherical bearing reaction

The spherical bearing reaction value was obtained by employing Equation (10):

$$P_Y = P_T . \cos\beta + G_K, \text{ kN}. \tag{10}$$

4.5. Determining the reaction in the upper bearing of the rolling cone

The reaction in the upper bearing of the rolling cone was determined as follows:

$$P_X = P_T . \sin\beta - P_E, \text{ kN}. \tag{11}$$

4.6. Determination of the oil pressure in the hydraulic cylinder of the crusher

The oil pressure may be calculated by using Equation (12):

$$p_0 = \frac{P_Y + G_Б}{\pi R_Б^2}, \text{ MPa}. \tag{12}$$

In the above equation, $G_Б$ and $R_Б$ are the weight and radius of the piston, respectively.

5. Computer Modelling

The device of the junction shaft -cone is complicated due to the presence of rotational movements. The classical approach to stress determination is the Finite Element Method (FEM), whose modern program implementation is through the three-dimensional parametric CAD-CAE system “Solid Works Simulation”. The FE technique is a numerical method for assessing engineering solutions [5, 6]. The working step can summarize the following:

(1) There are prerequisites for carrying out a study to identify the causes that cause mechanical damage in the loaded parts of the crusher.

(2) The computer research should be carried out as a linear structural analysis of the deformation, the tension state of the object under adequate load pattern and limitations of degrees of freedom for the worst possible working modes.

(3) Analyzing the design of the studied drive gears in order to improve the operational resource and reduce the risk preconditions for mechanical damage.

(4) The study neglected details that are unrelated to the carrying capacity of the construction. Schematically, the content of the methodology under discussion can be represented as such:

• An object is selected and a three-dimensional CAD model was created;

• The extreme parameters of mechanical loading for different operating modes of the selected node-moving cone are determined analytically by experts;

• Selection of the CAE software system, analysis, correct definition of boundary conditions (degrees of freedom), and loading to obtain the graphical and numerical results that characterize the deformational-tension state of the subject;

• Analysis of the results with regard to the distribution of extreme stress values, FOS, geometry, and topology of the details, physical-mechanical parameters of the materials used, working mode parameters, etc [7].

The analysis of the results was carried out in the following two categories: First is concerning the stress, the results were derived by the equivalent stress according to von Mises [2], which is expressed by the three main stresses as follows:

$$\sigma_{\text{von}} = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_1 - \sigma_3)^2}{2}}, \tag{13}$$

where σ_{von} is the equivalent von Mises stress and σ_1, σ_2, σ_3 are the first, second, and third major stresses, respectively. And factor of safety (FOS), this criterion is based on the Mises-Henky theory, according to which the FOS represents the relationship between the magnitude of the admissible stress corresponding to the elastic limit:

$$\text{FOS} = \frac{\sigma_{limit}}{\sigma_{von}} \geq 1. \tag{14}$$

Next, for the finite element method (FEM) application, it is necessary to analyze analytically the parameters of the jaw crusher for the most unfavorable and severe operating conditions with regard to the mechanical loading. For the purpose of simulating the machine detail load with the Solid Works Simulation software, the shaft-rolling cone was selected. The main model used in this research and assembled node-shaft is shown in Figure 3.

Figure 3. Full assembly of the component used in this research.

Setting boundary conditions is the second step in parameterizing the FEM study. The fixed surface is those parts of the shaft which are mounted in the eccentric cup and the suspension bearing, respectively. At the lower part of the shaft, there is a limitation in the heel-formed bearing (Figure 3) and in a certain area of the armor the force of crushing acts equally as the force of pressure (Figure 4). The crushing strength was calculated according to the methodology developed and was obtained 6841kN. For the assessment of the engineering solution, the three-dimensional model is divided into small parts of simple elements, interconnected with common points (nodes). The method determines the behaviour of the model by combining the information obtained from all the forming elements. The model meshing is one of the most important steps in the numerical study. A large number of elements implies a higher accuracy of the results but also increases the length of the computation process. Conversely, with a small number of endpoints, the calculation time decreases, but this is a prerequisite for network failures and inaccurate results. Figure 5 shows the model of the movable cone with the crusher shaft after discretization and Figure 6 shows the distribution of the equivalent stress in this unit at a normal load of the movable cone which is set to 6081kN.

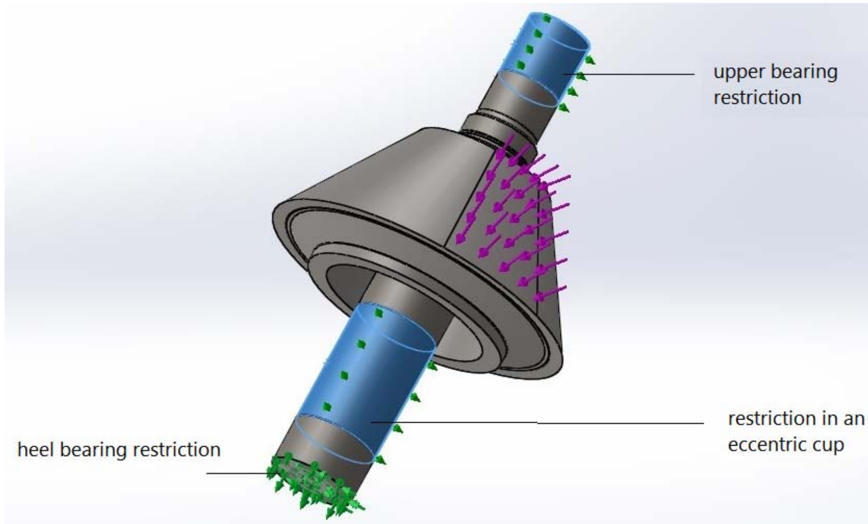

Figure 4. Fixed surfaces and constraints of the movable cone and the crusher shaft.

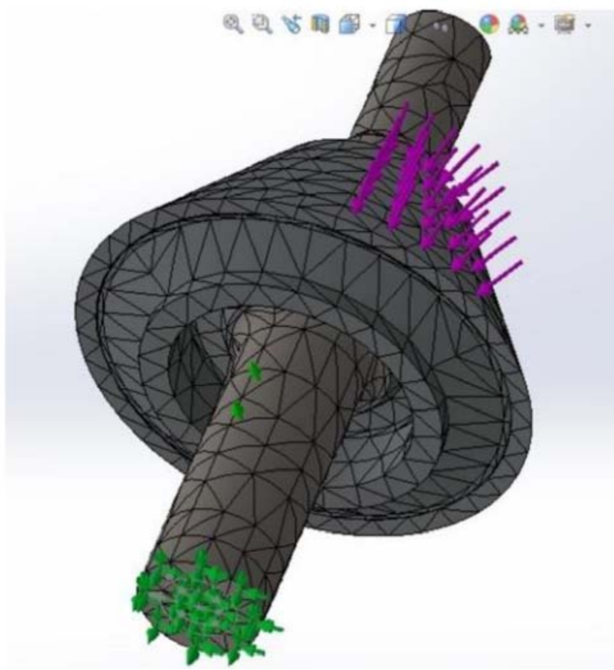

Figure 5. Discretization of the model with the applied crushing force.

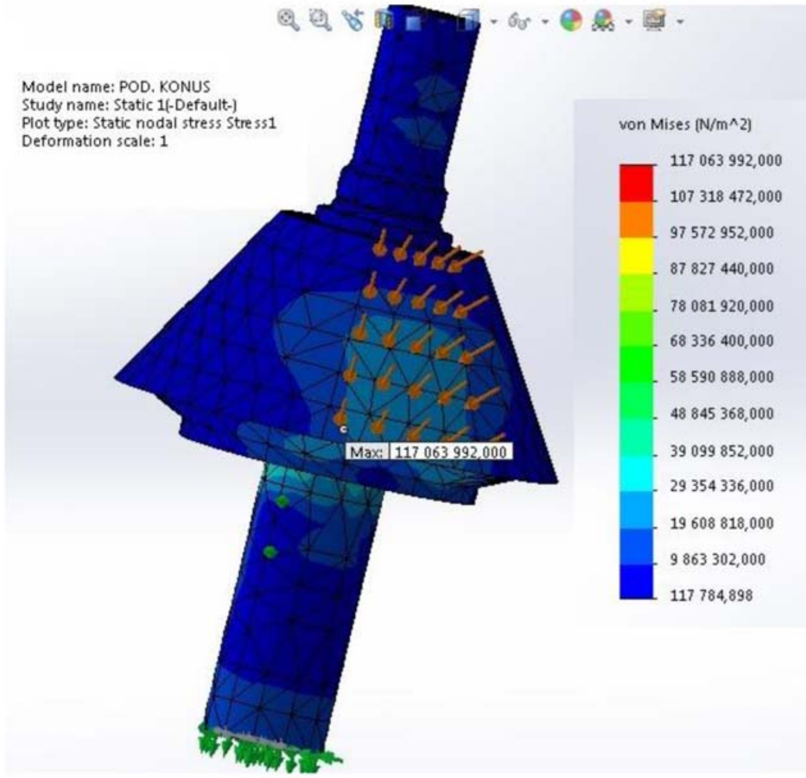

Figure 6. Distribution of the equivalent stress in a node movable cone-shaft at a 6081kN.

In Figure 7, the node is rotated so that the most stressed area located on the tread of the shaft emerging from the cone is seen. The value of this stress is about 117MPa.

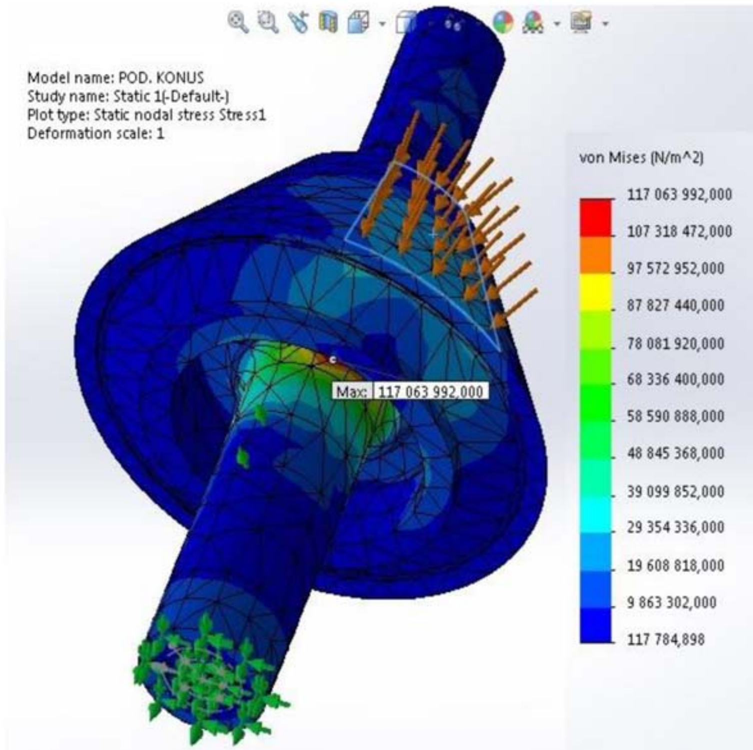

Figure 7. Zone of maximum stress at the component.

Figure 8 shows the distribution of the safety factor of steel and Figure 9 displays the distribution of the safety factor for the destruction of the steel as the minimum values are in the place of the maximum stress.

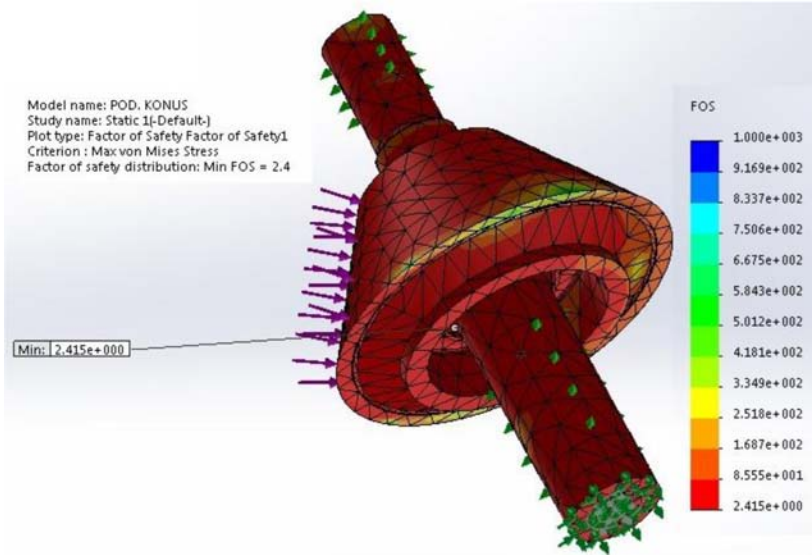

Figure 8. The distribution of the steel safety factor for pulling with a minimum value of FOS = 2, 42.

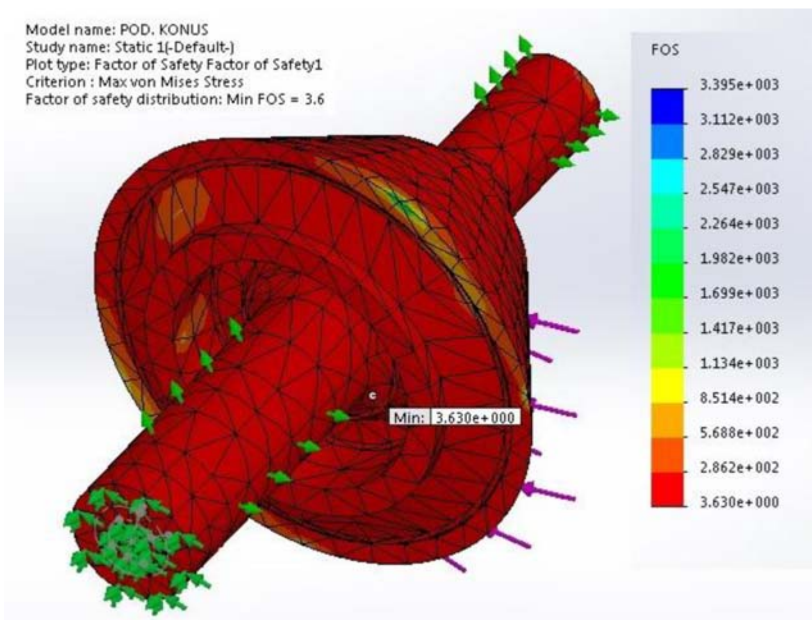

Figure 9. The distribution of the safety factor for the destruction of steel with a minimum value of FOS = 3, 63.

6. Conclusion

Through applying a new method for determining the critical region which has maximum stress and using the three-dimensional parametric CAD-CAE system "Solid Works Simulation", for the operation of the crusher the consequent facts were established:

(1) The high values of the crusher's axial junction response are evidenced by its rapid wear, problems in the operation, maintenance of the eccentric, and hydraulic system of the machine.

(2) The resulting high stresses in the machine shaft are close to the endurance limit, as evidenced by the many cases of cuts in the shaft in the step below the bottom of the cone.

(3) The repair of the crusher requires very high qualification of the service personnel due to the complexity of the units of which it is composed. In the other side, because of very high cost and from this point of view, this type of crushers is not suitable for the conditions of the processing.

(4) To achieve the efficiency and flawlessness of the technological process, the falling of noncrushing objects into the crushing space must not be allowed. Non-crushing objects include excavator teeth, parts of the chassis mechanism, bulldozers, pieces of metal, and so on. This can be avoided by placing metal detectors.

(5) In order to achieve economic efficiency, it is necessary to analyze the technological regime of the processing factory according to the border conditions of the facilities.

References

[1] H. Liao, E. A. Elsayed and L. Y. Chan, Maintenance of continuously monitored degrading systems, European Journal of Operational Research 175(2) (2006), 821-835.

DOI: https://doi.org/10.1016/j.ejor.2005.05.017

[2] Crush Operation Manual for "KUBRIA" 210/35, Krupp Fördertechnik GmbH, German, 2002.

[3] I. S. Minin, Technique and Technologies for Processing of Minerals, Sofia, Avangard prima, 2012.

[4] A. S. Donchenko, Reference Mechanic Ore Dressing Plant, Nedra, Moskva, 1975.

[5] T. S. Damyanov, Computer Modeling of Mechanical Systems, UMG-Sofia, 2009.

[6] P. Nedyalkov, FEM modelling and characteristics research of cylindrical journal bearing for cone inertial crusher /KID-300/, RECENT Journal 15(2) (2014), 112-116.

[7] B. Grigorov, Modeling and Designing with Solid Works, Sofia, 2002.

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