OPPORTUNITIES FOR USING THE FINITE ELEMENT METHOD BY COMPUTER RESEARCH ON A ROTARY CRUSHER OF THE ECCENTRIC TYPE 800Dx75

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ABSTRACT: *An analysis is made of the strain-stress state of an eccentric rotary crusher. The finite element method, and in particular its application, is used. Minimum factors of safety are determined on the basis of results obtained for the regions of the highest stress under different operating conditions. These have been analyzed and the possibilities for improving the design performance parameters of an eccentric rotary crusher have been assessed.*

KEYWORDS: eccentric crusher, ore dressing, mining mechanization, finite element method

1. INTRODUCTION.

A rotary crusher of an eccentric type breaks the material by crushing and twisting. Smaller pieces are subjected to crushing, while the larger are subjected to twisting. In this sense, a rotary crusher takes an intermediate position between a jaw crusher and a cone crusher. It has the following characteristic features that in the majority of cases are its definite advantages in comparison to jaw and cone crushers: 1) Simple structure due to the simple kinematical scheme of its design. This results in less consumption of metal and labour, and in the long run to less expensive machines. 2) Very steep (almost vertical) chambers. Due to this the material is discharged from the crushing area very quickly as a result of which the machine operates at high revolution. The high revolution of the rotor brings about a dynamic effect in the demolition of the material, hence lower energy consumption during the crushing process. 3) Lower consumption of energy for overcoming the harmful frictional resistance is due to the lower number of friction surfaces the result of which is the higher efficiency of the crusher. 4) Opportunities for a total dynamic balance. The machine has a smooth run and very low vibrations in the whole frequency spectrum [1,2,3]. 5) Curvilinear outline of the shattering planes due to which its crushing space does not tend to self-clog as is the case with jaw crushers. 6) A considerably lower dynamics factor compared to that of the jaw crushers because of the bi-lateral crushing of the material. Owing to this, the details of a rotary crusher are more favourably loaded and the machine needs lesser flywheel moments. In the long run, the specific consumption of metal is lower, the machine is lighter and costs less. As far as the physical and mechanical properties of the crushed product are concerned, we can claim that rotary crushers of the eccentric type are equally good at breaking all materials that can be broken by either jaw or cone crushers. As far as the toughness of the product is concerned, this machine is equal to jaw or cone crushers, and is less sensitive to wet, clayey, and abrasive materials.

The granulometric characteristics of the crushed material are closer to those of the product obtained by means of a one crusher. The amount of plane and rod-like grains is reduced to a minimum. The machine has a high crushing grade that, in some cases, can reach up to 10-12 units. This type of machine is best used for medium and fine crushing of strong materials and such with medium strength and is applied in construction work, in mineral processing, in chemical industry, in metallurgy, in cement production, etc. It can also be employed in an automated industrial process that requires frequent adjustment of the parameters of the output product without interrupting the flow line [4].

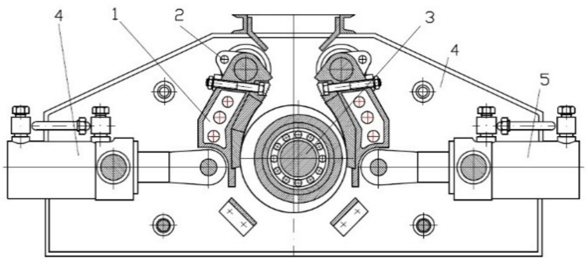

Fig.1. Rotary crusher

2. CAPACITY OF A ROTARY CRUSHER OF THE ECCENTRIC TYPE

As is the case with all crushers, with a crusher of this type the crushing power depends on the performance, on the physical and mechanical properties of the material, as well as on its granulometric characteristics. The physical ways of breaking with a rotary crusher of an eccentric type are very similar to those employed by jaw and cone crushers. Therefore, the most suitable equation is the following on obtained on the basis of Bond's hypothesis [4]:

$$N_{eng} = \frac{\kappa_{st}.w_0.Q}{100.\eta_m.\sqrt{D_t}}.\left[\sqrt{\frac{D_t}{d_t}-1}\right], kW, \tag{1}$$

where:

κ_{st} is the factor taking into consideration the stage of the crushing;

w_0 is energy consumption after Bond;

η_m is the efficiency of the drive mechanism;

D_t is the size of the square-shaped sieve mesh through which a t% of the input product passes;

d_t is the size of the square-shaped sieve mesh through which a t% of the crushed product passes;

Q is the mass performance of the crusher.

The maximum power of the engine can be determined after the equation

$$N_{eng}^{\max} = \frac{\kappa_{st}.w_0.Q^{\max}}{100.\eta_m.\sqrt{D_t}}.\left[\sqrt{\frac{D_t}{d_t}}-1\right], kW. \quad (2)$$

3. DETERMINING THE MEAN VALUES OF THE CRUSHING MAGNITUDE

The forces that act on the components of the crusher are variable. They depend on the magnitude of crushing and display the same nature of variation as that power. Within a single cycle, the force of crushing varies from 0 to P_{max}.

The magnitude of the reacting forces at the suspension points of the moving jaw varies from zero to the respective maximum value, as well as the magnitude of the reacting forces at the bases of the eccentric bearings of the movable rotor.

What is considered as the most unfavourable case is the one when the distribution of the material within the crushing space is such that the resultant force of the loads is applied in the middle of the crushing jaw (fig.2).

Since a rotary crusher of the eccentric type has got two working chambers located at either side of the eccentric rotor, two volumes of material are crushed within one revolution of the eccentric shaft whereby two peaks of the resultant crushing force that exerts pressure on the rotor are obtained.

The work necessary for the crushing of a unit of volume located in any of the crushing chambers will be

$$A_1' = 2.e.P_{calc}\ , \text{J}. \quad (3)$$

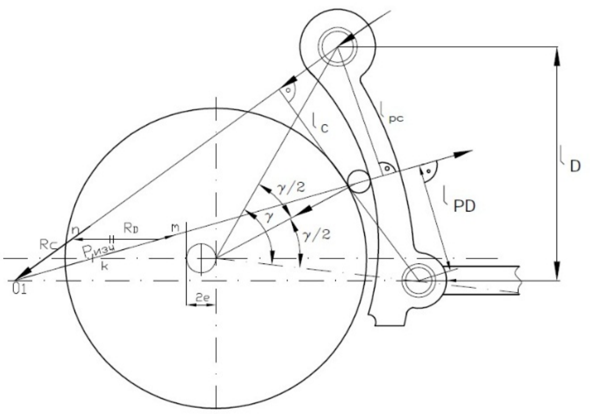

Fig.2 Crushing force and support reactions

Then, the total work per one cycle will be

$$A_1 = 2.A_1' = 4.e.P_{calc}\ , \text{J}. \quad (4)$$

This work will be performed by the engine in the course of a single working cycle (per revolution of the eccentric shaft). Therefore, we could put down

$$A_1 = \frac{1000.\eta_m.N_{eng}}{z}, \text{J}. \quad (5)$$

When calculating both equations (4) и (5), we obtain P $_{calculated}$

$$P_{calc} = \frac{1000.\eta_m.N_{eng}}{4.e.z}, \text{N} \quad (6)$$

The force P_{calc} is perpendicular to the working plane of the shattering jaw. The reacting force along the piston rod is directed along the axis of the rod. The forces of R_D, P_{calc} and R_C are counter forces, therefore they have a common point of intersection: point O where the directrices of the forces R_D and P_{calc} intersect.

Analytically, the support reactions are determined by the equations

$$R_C = P_{calc}.\frac{L_{PC}}{L_D}, \text{N}; \quad (7)$$

and

$$R_D = P_{calc}.\frac{L_{PD}}{L_C}, \text{N}, \quad (8)$$

Where L_{PC}, L_D, L_{PD} and L_C are the arms of the respective forces to the points C and D (fig.2) and can be determined by the 3D model of the crusher.

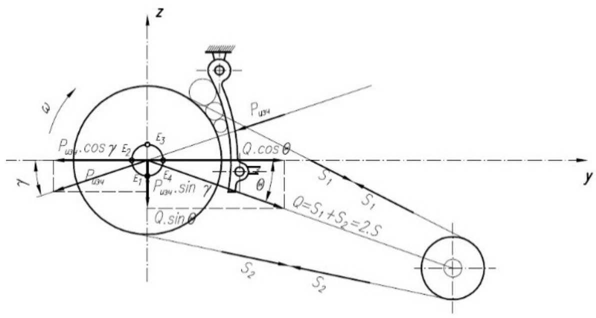

Fig.3. Forces in the belt gearing drive

3. DETERMINING THE FORCE IN THE BELT GEARING DRIVE

Torque of the crusher shaft.

$$M_{trq.} = 9554.\frac{N_{eng.}}{n_{shaft}} = 9554.\frac{90}{465} = 1849\ \text{N.m}$$

where:

$N_{trq.} = 90kW$ is the power of the engine;

Force of periphery

$$P = \frac{2.M_{trq.}}{D} = \frac{2.1849}{0,964} = 3836N$$

$S_{str} = \sigma_{str}.F.10^3 = 1,2.0,000918.10^3 = 1,1kN$,

where:

$F = 0,000918m^2$ is the belt section;

$\sigma_{str} = 1,2MPa$, is the strain of the preliminary stretching of the belt ;

$Q_{max} = 3.\sigma_{str}.z.F.\sin\frac{\alpha}{2}$, [kN] is the force over the shafts of the belt gearing drive;

$\alpha = 147^0$ is the angle of range of the belt pulley;

z =8 is the number of belts;

$$Q_{max} = 3.1,2.8.0,000918.\sin\frac{147}{2}.10^3 = 26kN$$

$$tg\theta = \frac{P}{2.S_{str.}}.ctg\frac{\alpha}{2} = 0,516$$

$tg\theta$ is the deviation of the force from the line connecting the centres of the two shafts $\theta = arctg0,516 = 27,3^0$.

4. STUDY OF THE ROTOR OF AN ECCENTRIC CRUSHER USING THE FINITE ELEMENT METHOD (FEM)

The first stage of the research was to create a 3D CAD model of the object studied. That was the case of the structure of the rotor of an eccentric crusher illustrated in fig.4. The simulation was carried out in the Solid Works medium with the highest possible detail of the output data. Fig.5 is a drawing of the cross section showing the overall dimensions of the machine.

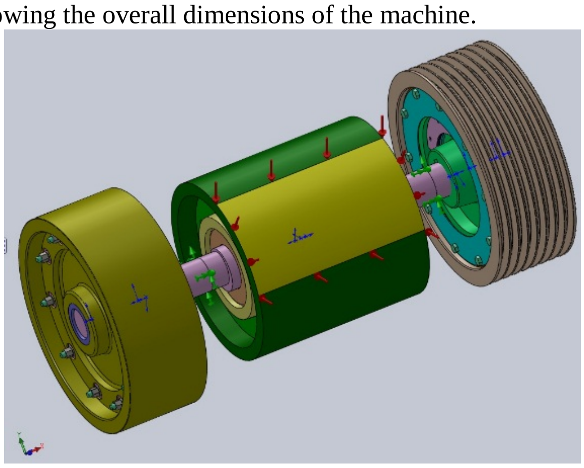

Fig.4. A 3D model of the rotor of an eccentric crusher type 800Dx75

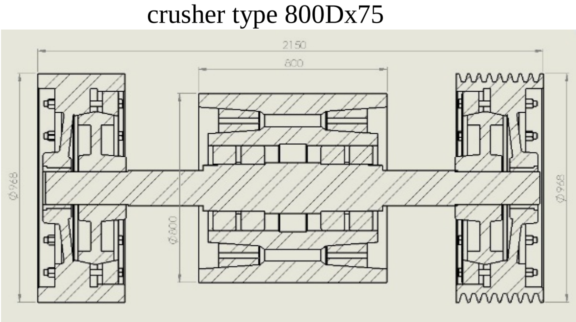

Fig.5. Overall dimensions of the rotor of an eccentric crusher type 800Dx75

The next stage was the precise planning of the study itself with the aim of obtaining the most accurate results. For this reason the research process was characterised by the following features:

- Choice of a software product, appropriate and at the same time powerful enough, for the purposes of studying with the finite element method. What was determining for our choice was the opportunity of the Cosmos Works product to carry out the so called "object discretisation", i.e. forming a lattice of finite elements with such a huge (multi-component) assembled unit (Fig.6).

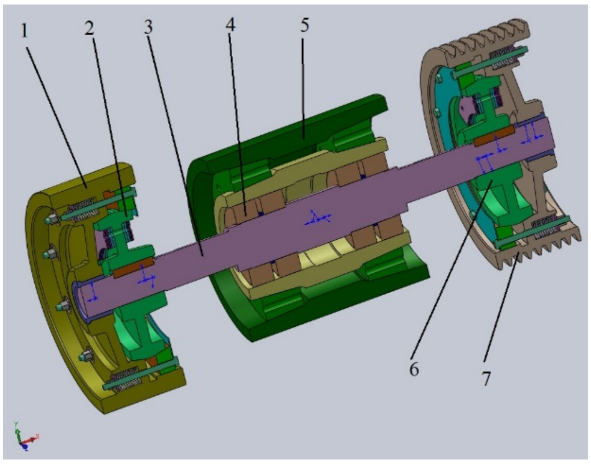

Fig.6. Cross-section of a 3D model of the rotor of an eccentric crusher type 800Dx75

Fig.6 shows a cross-section of the 3D model of the rotor that consists of the following elements: 1 - flywheel head; 2, 6 - friction clutch; 3 – shaft; 4- bearings; 5 – rotor; 7 – belt pulley.

The position of the bearings is limited by the option Fixed. This is necessary due to the fact that the structure needs to be immobile so that the static analysis could be carried out. The crushing force is given as load distributed over area, i. e. pressure. The area being 0.363 m^2 and the force acting over the area being given, the pressure obtained amounts to 716, 252 N/m^2.

- Defining the conditions for power load in terms of magnitude and distribution (Fig.7).

- Analysis of the strained and stressed conditions

In the course of our work, only a few of the opportunities of the software product were used for studying by applying the finite element method, namely:

* Equivalent stress according to the theory of von Mises.;

* Safety factor after the theory of von Mises-Henky;

* Static strain (absolute and equivalent).

According to the theory of von Mises-Henky, the yield limit is determined by the ratio of the magnitude of the equivalent stress after von Mises σ_{VON} to the magnitude of the allowable stress $[\sigma_{AL}]$

$$\sigma_{VON} \geq [\sigma_{AL}]$$

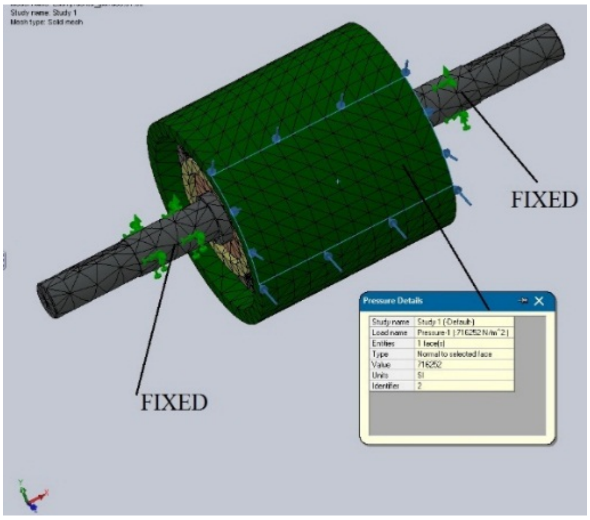

Fig.7. Boundary conditions imposed on the model and its discretisation (netting) within the Cosmos Works medium

The stresses after von Mises can be expresses by means of the three principal stresses, in accordance with formula (9):

$$\sigma_{VON} = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_1 - \sigma_3)^2}{2}} \tag{9}$$

The boundary value of the stress $[\sigma_{AL}]$ can be expresses as a function of the yield stress or the rupture stress of the material. Hence, the safety factor according to von Mises, called FOS (or: factor of security), will be calculated in accordance with formula (10):

$$FOS = \frac{[\sigma_{AL}]}{\sigma_{VON}} \tag{10}$$

5. RESULS

The results obtained for the stresses in the shaft and the rotor are illustrated in fig. 8.

Fig. 8 shows that the maximum stresses (after von Misses) in the shaft are within the range of 26,196,110 N/m^2 or 26.196MPa, which, according to formula (8), gives a safety factor of FOS=8.42 (fig.9).

The existing stress concentrators are in the preceding shaft shoulders (where the carrying elements, i.e. the bearings, are mounted). These results are within the norm and are admissible.

Fig.10 illustrates the strained displacement of the shaft and it is obvious that its maximum is along the periphery of the case of the rotor and amounts to 0.035mm.

The critical values of FOS=8.42, at which the stress is close to the boundary between elastic strain and plastic strain, as well as the alternating character of the load suggest an assessment of the fatigue strength of the structure, too, and this can be carried out in future dynamic tests of the machine.

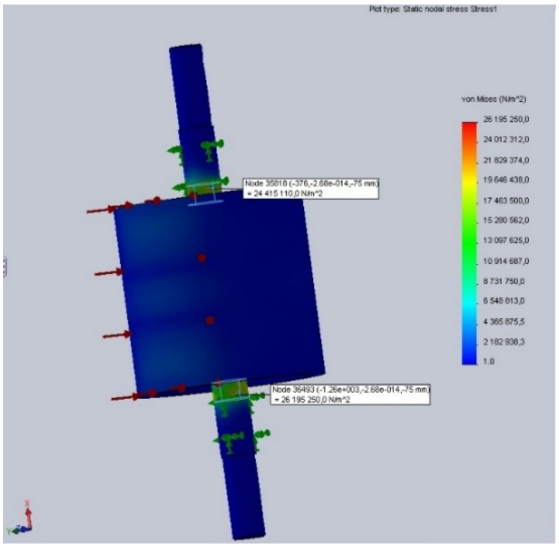

Fig.8. Distribution of the equivalent stresses on the CAD-CAE model

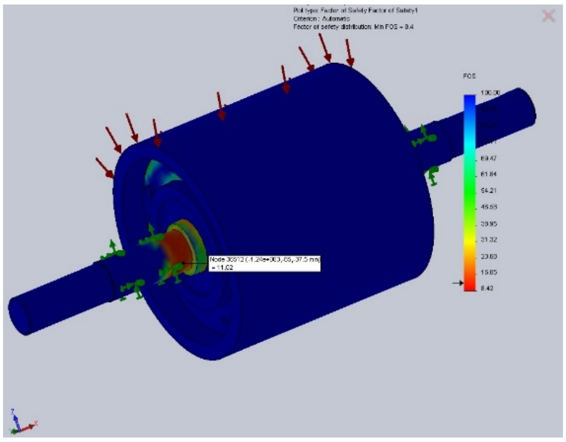

Fig.9. Distribution of the safety factor (FOS) over the drive shaft and the rotor of the crusher

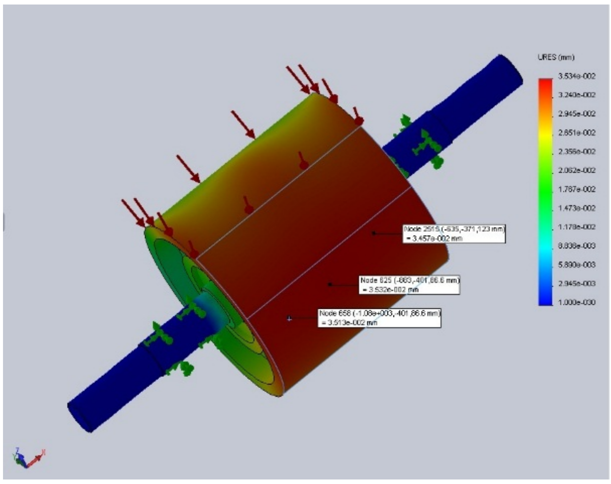

Fig.10. Distribution of the static displacement of the shaft and rotor

6. RESULT ANALYSIS AND CONCLUSIONS

The following conclusions can be drawn on the basis of the analysis of the results obtained:

- The data from the strain-stress state of the studied drive shaft and rotor make it possible to determine the magnitude and to visualise the distribution of the areas of high mechanical stress;
- Based on the results obtained, structural changes can be made, the distribution of stresses can be optimised, and the peak values can be reduced. In this way, the safety of the eccentric rotary crusher can be enhanced and the machine's operating cycle extended.

7. MODAL ANALYSIS

The conditions for the appearance of resonance phenomena in the mechanical system have unfavourable consequences and this presupposes a modal analysis. The results from the frequency analysis of the "shaft-hub-case" system of the eccentric rotary crusher are visually displayed in fig. 11, fig. 12, and fig. 13.

Fig. 11 shows the first harmonic frequency of twist of the whole system of "shaft- hub-rotor". This is the first frequency displacement of the rotor case: 89.6[Hz].

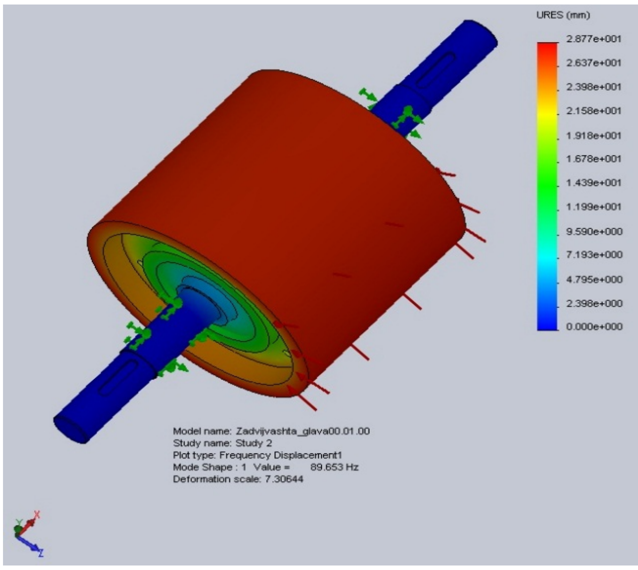

Fig.11. Results obtained for the first natural shape of the "shaft- hub-rotor" system

Fig. 12 shows the second harmonic frequency of the whole system of "shaft- hub-rotor" which is the second displacement frequency of the rotor: 329[Hz].

Fig. 13 shows the third harmonic frequency of the whole system of "shaft- hub-rotor" which is the third frequency displacement of the shaft and rotor: 340[Hz].

The chosen frequency of rotation of the rotor after the belt gearing drive is 465[min^{-1}]=7.75[sec^{-1}]=7.75[Hz].

Bearing this in mind, we have established that there is no danger of resonance to any element of the "shaft-hub-rotor" system of the crusher's structure that might be due to oscillations transmitted by the drive.

Therefore, the natural frequencies that we have established above only matter when other external induced mechanical oscillations appear whose frequencies are similar or equal to the natural ones.

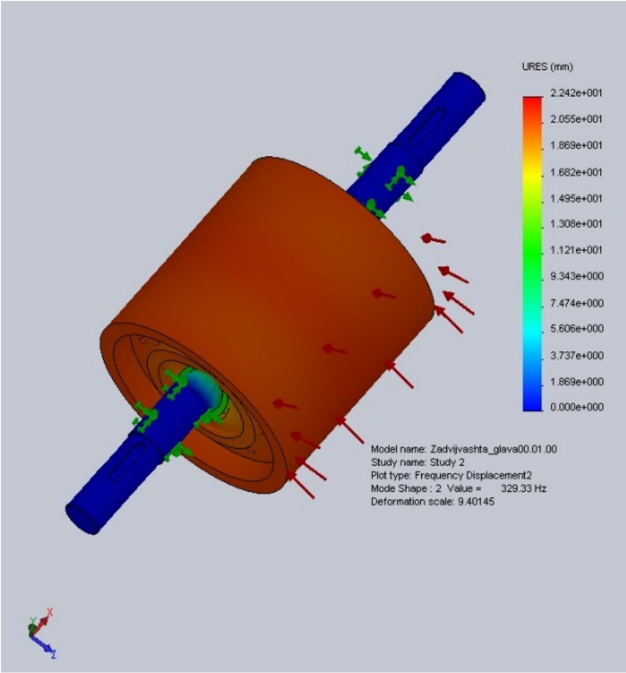

Fig.12. Results obtained for the second natural shape of the "shaft- hub-rotor" system

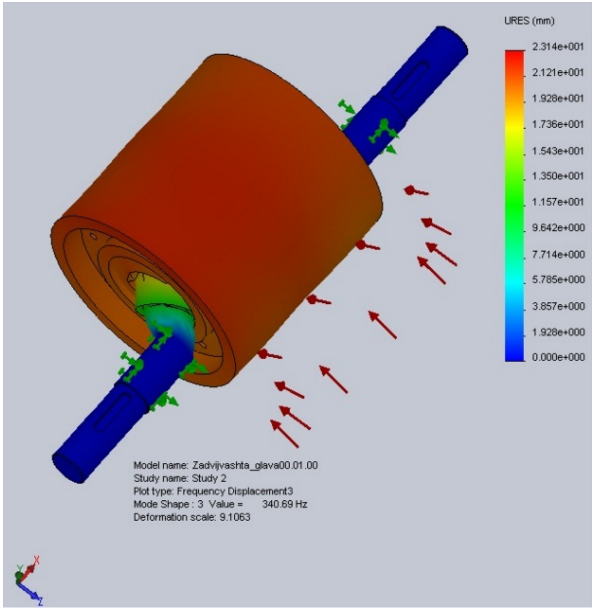

Fig.13. Results obtained for the third natural shape of the "shaft- hub-rotor" system

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