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

Iliev Zhivko, Perenovski Nikolay, Assistant Professors, University of Mining and Geology – Sofia

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,
therefore 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]:

\[
N_{\text{eng}} = \frac{k_{\text{st}} \cdot W_0 \cdot Q}{100 \cdot \eta_{\text{m}} \cdot \sqrt{D_1}} \left[ \frac{D_1}{d_i} - 1 \right], \text{kW}
\]
where:
- $k_{st}$ is the factor taking into consideration the stage of the crushing;
- $w_0$ is energy consumption after Bond;
- $\eta_m$ 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

$$N_{eng}^{max} = \frac{k_{st} W_0 Q^{max}}{100 \eta_m \sqrt{D_t}} \left[ \frac{D_t}{d_t} - 1 \right], kW. \quad (2)$$

$3. DETERMINING THE MEAN VALUES OF THE CRUSHING MAGNITUDE$

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

$$A_t' = 2.6 P_{calc}, J. \quad (3)$$

Then, the total work per one cycle will be

$$A_t = 2.6 A_t' = 4.6 P_{calc}, J. \quad (4)$$

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

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

When calculating both equations (4) and (5), we obtain

$$P_{calc} = \frac{1000 \eta_m N_{eng}}{4.6 z}, N \quad (6)$$

$3. DETERMINING THE FORCE IN THE BELT GEARING DRIVE$

Torque of the crusher shaft.

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

where:

Fig.2 Crushing force and support reactions

Fig.3. Forces in the belt gearing drive
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).

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

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

\[
N_{trq} = 90\text{kW} \text{ is the power of the engine;}
\]

Force of periphery
\[
P = \frac{2.1M_{trq}}{D} = 2.1849 \text{ kN}
\]

\[
S_{tr} = \sigma_{trr} \cdot F \cdot 10^3 = 1.2 \cdot 0.009181 \cdot 10^3 = 1.1 \text{kN}
\]

where:
- \( F = 0.000918 \text{m}^2 \) is the belt section;
- \( \sigma_{trr} = 1.2 \text{MPa} \) is the strain of the preliminary stretching of the belt;
- \( Q_{\max} = 3 \cdot \sigma_{trr} \cdot z \cdot F \cdot \sin \frac{\alpha}{2} \) [kN] is the force over the shafts of the belt gearing drive;
- \( \alpha = 147^\circ \) is the angle of range of the belt pulley;
- \( z = 8 \) is the number of belts;
- \( Q_{\max} = 3.12 \cdot 8 \cdot 0.009181 \cdot \sin \frac{147}{2} \cdot 10^3 = 26 \text{kN} \)

\[
\tan \theta = \frac{P}{2 \cdot \sigma_{trr}} \cdot \frac{\sin \frac{\alpha}{2}}{2} = 0.516
\]

\( \tan \theta \) is the deviation of the force from the line connecting the centres of the two shafts \( \theta = \arctan 0.516 = 27.3^\circ \).
The stresses after von Mises can be expressed by means of the three principal stresses, in accordance with formula (9):

$$\sigma_{VON} = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}}$$  \hspace{1cm} (9)

The boundary value of the stress \( [\sigma_{AL}] \) can be expressed 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):

$$FOS = \frac{[\sigma_{AL}]}{\sigma_{VON}}$$  \hspace{1cm} (10)

5. RESULTS

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 Mises) in the shaft are within the range of 26,196,110 N/m² 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.
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].

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.

8. REFERENCES:

[1]. Цанев Ц. Върху някои кинематични и конструктивни особености на роторна трошачка ексцентриков тип., Рудодобив, бр. 9, 1973г.
[2]. Цветков Хр., Цанев Ц. Кинематика на роторна трошачка ексцентриков тип при