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APPLYING DISCRETE ELEMENT MODELING TO THE ROTOR COMPONENTS OF A HORIZONTAL SHAFT HAMMER MILL

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Abstract: *In the present paper it's presented a DEM (Discrete Element Modeling) structural analysis that was used for the components of the rotor from a horizontal hammer mill. During operation they are subject to centrifugal forces caused by the rotation of the rotor. A detailed simulation study was conducted to investigate the deformations of the rotor components, the tensions that appear during operation and their variation transversely through dangerous zone.*

Key words: *Modeling, Simulation, DEM.*

1. INTRODUCTION

The literature shows that substantial effort has been expended in understanding the impact crusher performance in relation to machine configuration and operational conditions through experimental work and mathematical modeling ([9], [10]).

However, due to lack of detailed knowledge on velocity and energy distributions of collision inside a milling chamber, the mechanisms are still not clear.

The DEM was first proposed by Cundall and Strack (1979) to model the behavior of soil particles subject to dynamic loading [4].

Mishra and Rajamani [7] pioneered the application of DEM to grinding mills and demonstrated that despite the DEM simulations were two dimensional (2D), the technique was able to predict the power draw of mills with reasonable accuracy over a wide range of mill diameters.

Since then, the DEM technique has been widely applied to ball mills (Cleary [2], van Nierop [11]), centrifugal mill (Inoue and Okaya [6], Cleary and Hoyer [3]).

Meanwhile the DEM code has been extended from 2D to 3D, and the contact parameters involved in the DEM model have been studied and corrected to improve the simulation accuracy.

The DEM has also been applied to studies of impact induced particle breakage. Potapov and Campbell (1994) [8] found that ratio of the impact velocity to propagation velocity of the longitudinal (sound) waves in the material was a useful parameter that described the rate at which the kinetic energy of the collision was transferred to the strain energy of the particle.

Djordjevic [5] studied the PFC3D (particle flow code) that models the movement and interaction of particles by the DEM techniques was employed to simulate the particle movement and to calculate the velocity and energy distribution of collision in two types of impact crusher: the Canica vertical shaft crusher and the BJD horizontal shaft swing hammer mill.

2. METHODS

DEM method was used in the present paper in order to evaluate the protection against plastic failure of the components in a rotor of a horizontal hammer mill. Three alternative analysis methods are provided for evaluating protection against plastic collapse [1]:

(a) Elastic Stress Analysis Method – Stresses are computed using an elastic analysis, classified into categories, and limited to allowable values that have been conservatively established such that a plastic collapse will not occur.

(b) Limit-Load Method – A calculation is performed to determine a lower bound to the limit load of a component. The allowable load on the component is established by applying design factors to the limit load such that the onset of gross plastic deformations (plastic collapse) will not occur.

(c) Elastic-Plastic Stress Analysis Method – A collapse load is derived from an elastic-plastic analysis considering both the applied loading and deformation characteristics of the component. The allowable load on the component is established by applying design factors to the plastic collapse load.

Elastic stress analysis method

To evaluate protection against plastic collapse, the results from an elastic stress analysis of the component subject to defined loading conditions are categorized and compared to an associated limiting value.

(a) A quantity known as the equivalent stress is computed at locations in the component and compared to an allowable value of equivalent stress to determine if the component is suitable for the intended design conditions. The equivalent stress at a point in a component is a measure of stress, calculated fromstress components utilizing a yield criterion, which is used for comparison with the mechanical strength properties of the material obtained in tests under uniaxial load.

(b) The maximum distortion energy yield criterion shall be used to establish the equivalent stress. In this case, the equivalent stress is equal to the von Mises equivalent stress given by Equation (1) [1]:

$$S_e = \sigma_e = \frac{1}{\sqrt{2}}\left[(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2\right]^{0.5} \quad (1)$$

Limit-load analysis method

(a) Limit-load analysis addresses the failure modes of ductile rupture and the onset of gross plastic deformation (plastic collapse) of a structure.

(b) Displacements and strains indicated by a limit analysis solution have no physical meaning. If the User's Design Specification requires a limit on such variables, the procedures must be used to satisfy these requirements:

(c) Protection against plastic collapse using limit load analysis is based on the theory of limit analysis that defines a lower bound to the limit load of a structure as the solution of a numerical model with the following properties:

(1) The material model is elastic-perfectly plastic with a specified yield strength.

(2) The strain-displacement relations are those of small displacement theory.

(3) Equilibrium is satisfied in the undeformed configuration.

Elastic-plastic stress analysis method

(a) Protection against plastic collapse is evaluated by determining the plastic collapse load of the component using an elastic-plastic stress analysis. The allowable load on the component is established by applying a design factor to the calculated plastic collapse load.

(b) Elastic-plastic stress analysis provides a more accurate assessment of the protection against plastic collapse of a component relative to the previous two methods because the actual structural behavior is more closely approximated. The redistribution of stress that occurs as a result of inelastic deformation (plasticity) and deformation characteristics of the component are considered directly in the analysis.

2. NUMERICAL ANALYSIS

First of all must be establish the work areas where tensions are maximal (area of interest). In this way will be removed from the geometric model parts do not have a direct influence on the model considered.

Field of interest is chosen and also the type of analysis that we want to achieve. Thus, for a structural analysis we choose parts that will be in motion and will be submitted more during operation.

The next step is to realize the geometric model will be subjected to structural analysis (Fig. 1).

It was used only a quarter of the entire geometric model to reduce computing effort. This does not affect the analysis because symmetry conditions were imposed on areas covered by the plans XY and XZ (fig. 2).

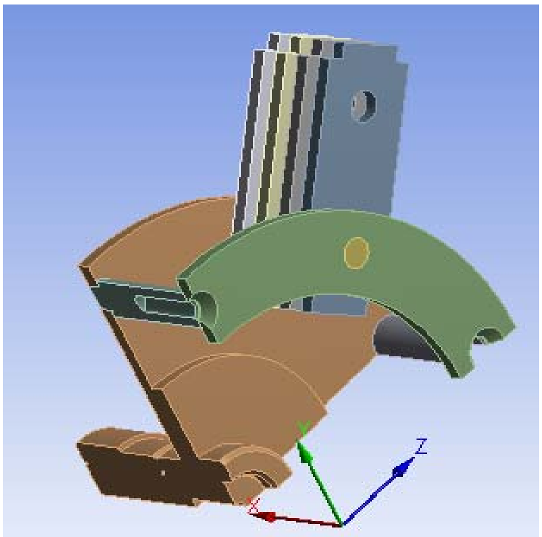

Fig. 1 Geometric model of the hammer mill subject to structural analysis

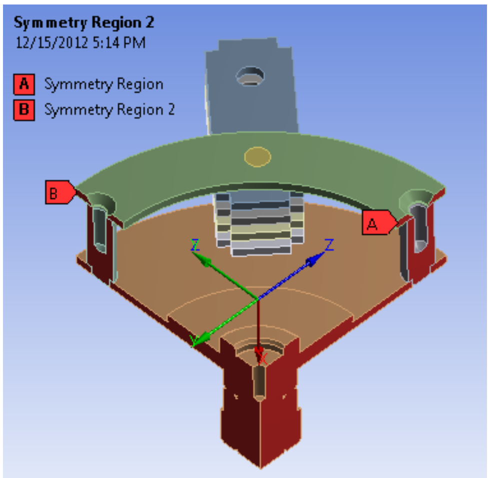

Fig. 2 Establishing the symmetry of the model

The next step is to choose the material of the components of the model used for parts. This is 1C45 (AISI 1045). Further, ANSYS set stress – strain curve of the material chosen.

The next step is the discretization of the model (fig. 3). For the structural analysis rectangular elements were used to the extent allowed by the geometry complexity. For the parts with a more complex geometry tetrahedral elements are used. In this way is more simple the division algorithm in finite elements of the model.

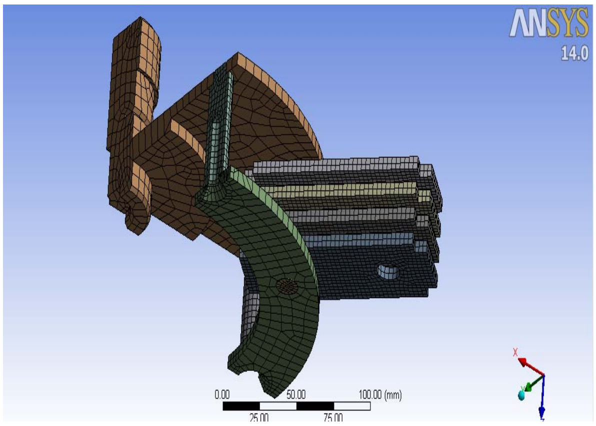

Fig. 3 Discretization of the model subjected to DEM analysis

For the rods and hammers smaller elements were used in order to capture the detailed stress and strain states.

Next were established the types of contacts between the component parts of the model geometry (fig. 4).

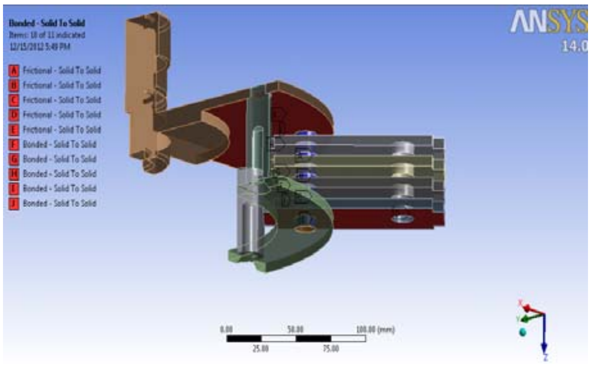

Fig. 4 Establishing contacts between component parts of the geometric model

Next the value of the rotation speed of the model (the rotor of the hammer mill) must be set. It is a static structural analysis, will consider only the centrifugal forces acting on the model.

3. RESULTS AND DISCUSSION

Using structural analysis for each item was determined the equivalent stress model and total deformations.

Hammers are subject to stretching due to the centrifugal forces that acts on them. The values of the equivalent stress occurring during operation varies between 0.047 - 102.5 MPa. The maximum values were recorded at the contact between hammer and the rods on wich they are articulated (fig. 5).

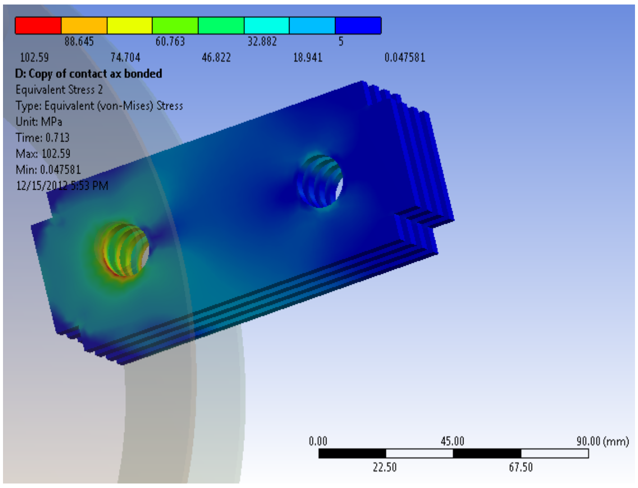

Fig. 5 Equivalent stresses that appear in the hammers

The total deformation recorded for the hammers is 0.67 mm as shown in the figure below:

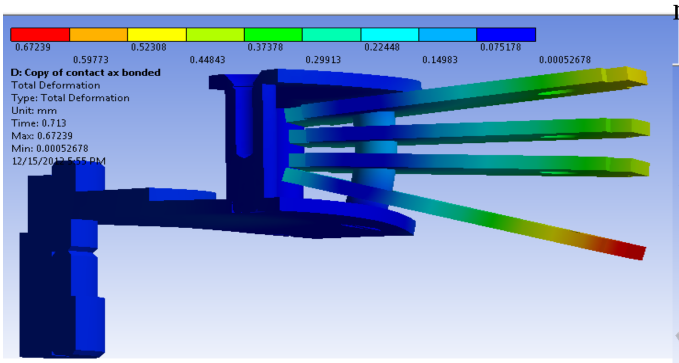

Fig. 6 Maximum deformation recorded for the hammers during operation

In the case of the rods on which are articulated the hammers, the maximum equivalent stress recorded had the value of 465, 29 MPa as shown in figure 7.

Total deformation of the rods during operation reaches 0.12 mm as can be seen in figure 8.

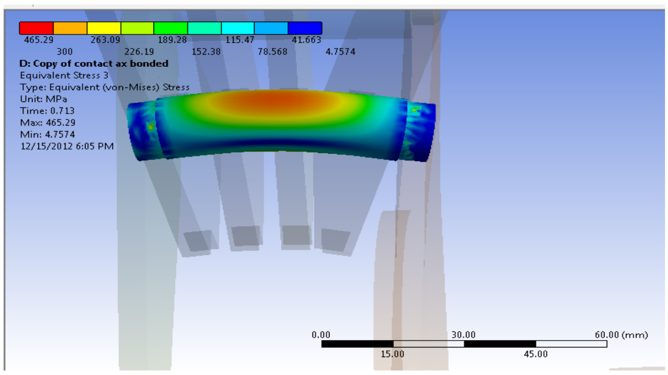

Fig. 7 The equivalent tension in the rods on which the hammers are articulated.

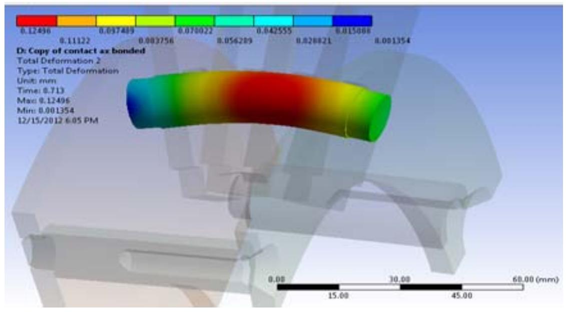

Fig. 8 Total deformation of the rods

Regarding the equivalent stresses that appears in the disc of the rotor (the one that holds together hammers package), the values range between 1.27 - 241.67 MPa (Fig. 9).

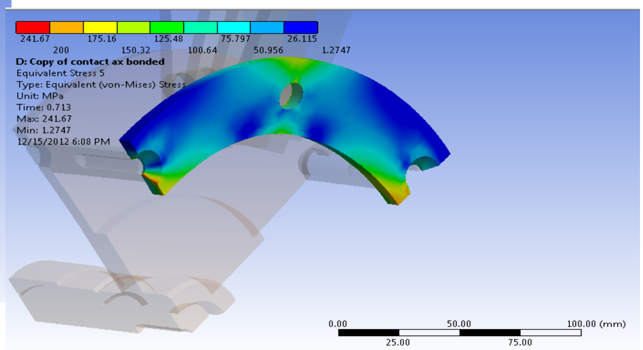

Fig. 9 The equivalent stresses that appears in the disc of the rotor

Was also made a linearized analysis of the equivalent stress that appears in the rods on which the hammers are articulated and in the rotor disc portion in the points where the amount of deformation energy has maximum. This analysis consists of a linearized analysis of tensions variation within the model after a given direction.

The recorded values of the tensions that appear in the rods take values between 27,572 - 578.78 MPa (fig. 10).

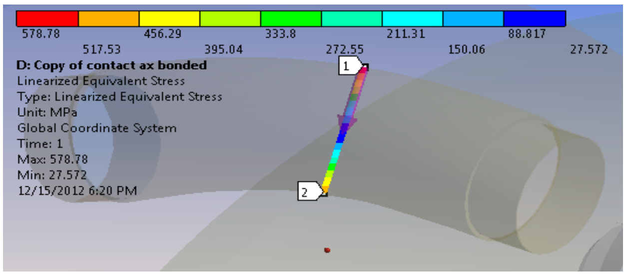

Fig. 10 The variation of the tensions that appear in the rods

According to the method of elastic analysis of the model, the software records the values of membrane tension (stretching), bending, bending and tensile stress, peak tension and total value in a multitude of points from point 1 to 2 transversely thru the rods. The values of the stress in each point, and their chart can be seen in figure 11.

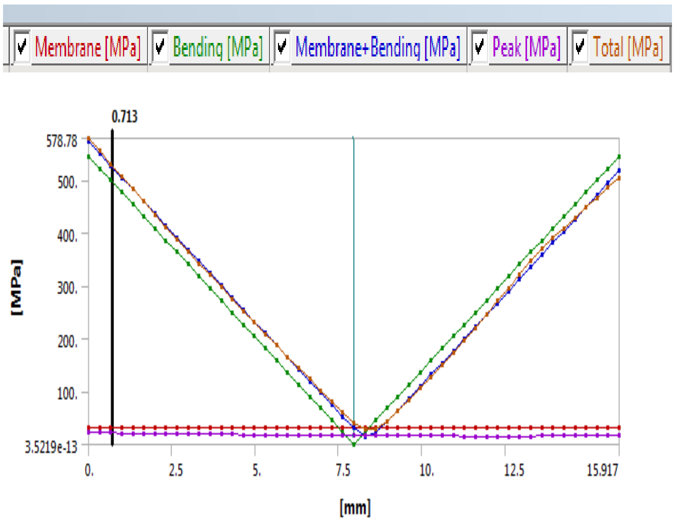

Fig. 11 Graphical representation of the variation of equivalent stress that appears in the rods of the rotor after 1 – 2 direction.

The same type of analysis of the tensions variation was carried out for the disc rotor. Chosen direction is 1 - 2 shown in figure 12, and was selected in the area where stresses and strains have maximum values. Variation of these stresses on the 1-2 direction is shown in figure 13. The values of the recorded tensions range between 109.22 - 311.03 MPa.

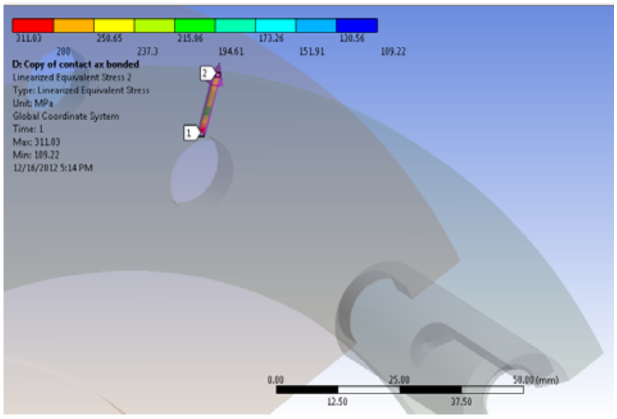

Fig. 12 Variation of the equivalent stress in the rotor disc in the direction 1 - 2.

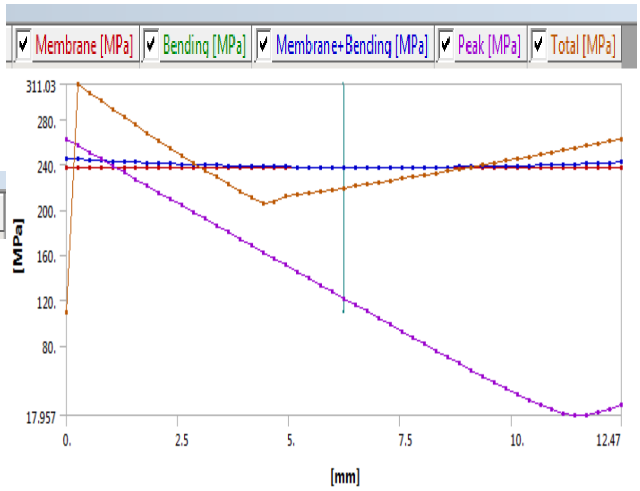

Fig. 13 Graphical representation of the variation of equivalent stress that appears in the discs of the rotor after 1 – 2 direction.

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Aplicarea metodei de analiză cu element finit pentru componentele morii cu ciocane cu arbore orizontal

Rezumat: *În lucrarea de față este prezentată o analiză structurală DEM (Discrete Element Modeling) la care au fost supuse componentele rotorului morii cu ciocane cu arbore orizontal. În timpul funcționării acestea sunt supuse forțelor centrifuge ce apar ca urmare a rotației rotorului. O simulare detaliată a fost realizată pentru a investiga deformările componentelor rotorului, tensiunile ce apar în timpul funcționării şi variația transversală a acestora prin zona periculoasă.*

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