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DESIGN OF A NEW RADIAL SHEAR MILL BASED ON SIMULATION AND IMITATION MODELING

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At present time, the spectrum of virtual modeling and engineering analysis software systems has significantly expanded [1]. Developers improve these systems to solve more complicated problems, for example, nonlinear, multifactorial and multi-criteria dynamic processes of metal forming.

To solve these tasks, the universal CAE-systems of the so-called “heavy” class (MSC Marc, ANSYS, etc.) have been used, with the help of which equipment designs and technological equipment for various purposes are designed for strength and deformation calculations [2]. At the same time, the use of “heavy” CAE systems for solving various technological tasks is problematic due to the high resource requirements to the software and hardware complex, the need of highly qualified and specialized user knowledge, the complexity and duration of setting up of such systems for a specific task.

At the present work, a radial-shear mill (RSM) of a new design is proposed [3]. This mill is a complex of rolling and heating equipment, where by combining of hot screw rolling on RSM and pressing, rods of small diameters or wires of ferrous or non-ferrous metals with a fine-grained structure are obtained. RSM for continuous pressing of rods and wires contains a main drive, a working cage, a roll knot and a press matrix. The three-rolled working cage RSM consists of a stand, in the boring of which the working rolls units through 120 ° are mounted. Working rolls are mounted on pillows, torque, which is transmitted through spindles from electric motors. The cages of the new mill are designed with the possibility of the rolls location with different angles to the axel of rolling and tangential displacement relative to it for 18 mm. The rolls of this RSM have smooth and screw-shaped gripping and crimping areas, respectively, and calibrating cylindrical

areas.

The working cage of the proposed rolling mill is a multi-component machine, including rotating rolling rolls, bearings, stand, mounting mechanisms of the rolls, fastening devices and other units and details. Designing of such a machine is a very complicated and labour-consuming process, requiring a large amount of calculation and graphic work. It should be noted that the existing methods for calculating of the elastic deformation of the stands driving working rolls do not allow to take into account all the constructional and technological features of the hot-pressing process of bars [1]. In this regard, we, by the imitating modeling investigated the strength and vibration of heavily loaded elements of the new mill. The initial data for the calculation is the solid-state geometric form of RSM, the forces applied to them and the conditions of fixing.

To calculate the elastic deformation and vibration of heavily loaded elements of the mill cages, we selected the Patran Nastran [4] finite element analysis program and developed the RSM dynamic computer model. Patran Nastran computer modeling system allows to explore the kinematics, dynamics of mechanisms with the ability to calculate the deflection, vibration, stress-strain state (VAT) and thermal state, as individual links, and the mill as a whole.

During the construction of RSM in the Patran Nastran environment, we performed the following operations [1]: created geometric models of each detail and assembly of the mill units; selected the type of finite element analysis; created a grid of the finite elements of the new RSM cage details; chose the material of parts, according to their mechanical and physical properties; set the rheological properties of the material; formed kinematic and static boundary conditions; set the conditions to the coefficient of friction; depending on the technical characteristics of the RSM and the condition of constancy of the second volumes, set the law of rolls rotation of the investigated mill; on the basis of the finite-element method, volumetric VAT calculated the components of the stress tensor, Mises equivalent stresses, and also the elastic deformation and vibration of the heavy-loaded elements of the mill; in the form of distribution fields of the corresponding

parameters according to the volume of detail, presented the results of the calculation; made an assessment of the level of received displacements, vibration, elastic deformations and stresses in the volume of each part of the mills relative to the required criteria of hardness and strength and introduced appropriate changes to the design of the mill.

By the simulation modeling in the MSC Nastran environment, the above given indicators were calculated according to the following algorithm. In the KOMPAS program, according to the working drawings, three-dimensional geometric model of each part was created and the units of the working cage were assembled. The model was imported into the Nastran Patran preprocessor with accepted kinematic links. Selected the materials of details, their mechanical and physical properties. Formed kinematic and static boundary conditions, modeled the calculated mechanical scheme, including the load distribution over the surface of the rolled billet. We made a supply of torque to the rolls (torque was modeled using MPC elements of RBE2 type). Using the options Mesh Seed applied 6 and 8 knot volumetric finite element grid and determined the vibration and VAT. When developing a design scheme, condensed grids of finite element were used in places of expected stress concentration. Elastic bonds between the units of the cage were modeled by the spring-damping element CBUSH.

The stress state in heavily loaded mill parts was calculated by applying rolling force and impact of thermal stresses. The MSC.SuperForge software complex was used to calculate the energy-and-power parameters of bars pressing. The distributed force acting in the contact zone on the roll was taken according to the series of modeling results of copper alloy M1 pressing and taken equal to the maximum 320 kN for each roll 280 kN per matrix.

For the calculation of the working rolls put a load in the form of moment and force with which the billet affects to the rolls. After all the operations for setting the initial conditions of simulation modeling, the VAT of the mill design was calculated.

As a result of computer modeling, data were obtained using which one can

make a correction to the mill design on the stage of project. It was found that the maximum equivalent stress occurs in the necks of the roll and make up 499.5 MPa. The calculated maximum values of equivalent stresses in a heavily loaded roll and matrix do not exceed the maximum allowable value of the strength limit of steel 9X1 (880 MPa). The stresses in the stand of cage made up 45.412 MPa, which is also significantly less than the strength limit for the stand material (981 MPa for steel St40HS).

The greatest displacement under the action of load is equal to 0.000434 mm and occurs on the necks of rolls. This value is within the tolerance for the diametrical dimensions of bars with a diameter of up to 20 mm. For the stand, the gratest displacement is under the load of 0.000315 mm and occurs in its upper part, and for the RSM matrix this displacement is equal to 0.000093 mm.

The calculation of stress tensor components showed that when rolling on a new mill, stretching main stresses arise mainly in the necks of rolls. The magnitude of the main maximum, medium and minimum stresses in the necks of working rolls do not exceed 631.9, 263.8 and 69.95 MPa, respectively. The main maximum stresses in the stand of the cage are stretching and do not exceed 26.84 MPa. It should be noted that the main medium and minimum stresses arising in the stand of cage, in most cases, are compressive and do not exceed respectively 7.12 and -43.97 MPa.

The results of computer modeling evidence that the RSM cage has a sufficient safety margin. Estimated hardness of the mill design provides rods and wires with diametrical dimensions of high accuracy.

It should be noted that the determination of the maximum strength of heavily loaded parts of the RSM cage was carried out by pressing rods at room temperature. According to the project RSM is intended for hot pressing of rods and wires. We believe that during hot pressing all heavy duty parts with a 5-10 divisible coefficient margin of safety will withstand the applied load.

The calculation results showed that the projected RSM roll knots have low hardness in the horizontal plane. This is due to the lack of supports in the stand

cages, excluding the displacement of working rolls in the horizontal plane, as well as with non-horizontal location of the mill drive spindles. As a result, even small gaps between the bearings, pads and casement of stands, caused by the landing tolerances and wear, lead to horizontal displacements of the vertical axial plane of the working rolls, i.e. the working rolls occur in an unstable position, and their axes can be warped. This leads to negative consequences: increased axial forces occur in the roller unit, and the size of the roll gap undergoes to unpredictable vibrations, which reduces the accuracy of rolling.

By the calculation it was found that the pillows are elastically deformed in the vertical and horizontal planes and rotate at a small angle relative to the axis of rolling. Elastic displacement in the direction of the load action for pillows located on the drive side of the roll for 1.41 times greater than for pillows located on the opposite side of the roll.

The calculation results showed that the use of hydraulic cylinders to reduce gaps leads to the decrease in both the size and area of elastic deformation of rolling bearings. In this case, the elastic deformation of the bearings vary from 0.0008 to 0.0009 mm. The outer rings of the bearing have the maximum deformation, while with a decrease in the diameter of the rolls, the region of the greatest elastic deformation of the bearing shifts to the inner sides of the roll neck. This is due to the change in the scheme of load application in the RSM roll knot.

In accordance with the conducted research, were developed measures for the modernization of the RSM cage. Showed, that it is necessary to install a hydraulic pressure device in the cage to regulate the inter-roll gap of the working rolls. To ensure uniform distribution of the load and increase the service life, we suggested to replace the bearings. In the pillows of the cage, instead of a double row spherical roller bearing, we suggested to use a radial-resistant conical double-row roller bearing with a greater carrying capacity. The proposed project solutions will reduce the level of deformation of the cages.

It is known that during the RSM operation, resonant oscillations occurring when the own frequency of structure and the frequency of external forces coincide

are especially dangerous. That is why, it is important to determine the frequency of external forces. One of the main exciter of resonant frequencies in the construction of rolling mills is the working rotation frequency of working rolls drive, which distribute vibrations to the mill.

Researches established that with an increase in rolling speed, a slight increase in dynamic loads occurs at the main knots of the cages and in the drive line on the mill of a new design. The mechanical vibrations of the cage knots and drive lines become a cause of a small vibration occurrence. Researches have shown that, relatively small in size, forced oscillations appear in the cages of the new mill, which are excited by external periodically varying forces and speed of rolling.

It should be noted that although dangerous resonant vibrations do not occur in the matrix, stand and roller knots of the RSM, we attempted to change the design of this mill in such a way as to bring its natural frequencies beyond the working range and estimate the resonant vibrations and get a large margin before appearance above marked vibrations. Therefore, in the design of the new mill we introduced a hydraulic dampers for clamping the matrix and roller knot.

It should be noted that during the pressing of rods or wires in the proposed RSM, the change in metal form occurs under alternating shear deformation conditions. In this case, the billet is twisted around its axis, which leads to the movement of each metal particle in the deformation zone along a helical (helicoidal) trajectory. Such a turbulent movement of the metal leads to an intensive study of the structure in the peripheral and central zone of the billet. All this leads to an increase in the quality of bars or wires.

Conclusions:

1. It is shown that the application of the program of finite element analysis Patran Nastran is an effective tool for studying the influence of the stress-deformation state and vibration of various rollers on the mill strength and the geometric dimensions of the rolled billet.

2. On the base of modeling results, it was proved that the stress values occurring in the details of cage structure of the radial-shear mill in the process of

pressing rods and wires do not exceed the maximum allowable stress.

3. It is proved that when pressing rods and wires on the RSM, dangerous vibrations do not fall to the operating range of the external loads, so the structure of the new mill is quite good from the point of view of vibration strength.

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