

STRESS-STRAIN STATE OF MILL FOR COLD ROLLING OF TUBES

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Abstract

The stress-strain state of construction of the cold rolling mill for large diameter tubes was investigated. Mathematical model based on the use of numerical finite element method, and the special element, which is located on contact surface, is used to simulate the contact interaction. Computer simulation based on the proposed model allowed us to establish the causes of extremely low durability bearing and offer a fundamentally new solution, which enables in multiple times to increase uptime of the roll mill.

Key words: rolling mill, work roll, bearing, caliber, nonlinear problem, computer simulation

1. INTRODUCTION

Chelyabinsk Tube Rolling Plant is Europe's largest manufacturer of large diameter tubes. At the plant was approved the concept of development. This concept proposes the introduction of new technologies in the tube-rolling production, which can enhance the durability of the equipment. One of the main objectives of this concept is to increase the uptime of the work rolls, which are mounted in cold rolling mill.

The composition of the rolling mill includes the following components: a unmovable frame of gantry type, traverse, wedges, roller bearings, support rings, which mounted on the outer ring of bearings; work rolls. The working roll is placed in the roll cage, which is able to perform a reciprocating motion along the traverse at a given distance.

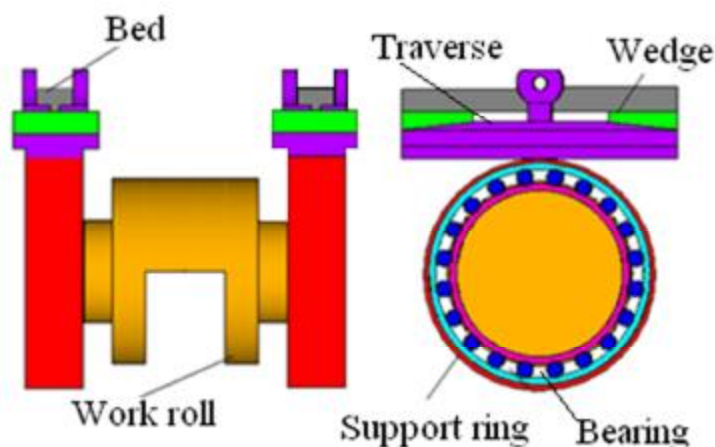


Fig. 1 Structural elements of tube rolling mill XIIT-250

In the process of technological cycle the rolling force is transferred from the caliber, mounted in the groove of work roll, on the work rolls, and from the work rolls on the unmovable traverse, through the multirowed rolling bearing. The current force is transmitted from cross member on the wedges, designed to adjust of the gap between rolls, and from the wedges on an unmovable frame. In accordance with the specifications, the maximum rolling force is 4 MN and is concentrated in a relatively narrow area, which includes a limited number of rolling elements. The presence of local zones of loading leads to rapid wear and subsequent destruction of the bearing. All design elements have a tendency to strain during loading, while between the contacting elements may appear a gap or, alternatively, a gap may be selected. Therefore, to determine the stress - strain state of the complex design needs to be addressed constructively nonlinear problem. The simplifying of the calculation scheme of design of the rolling mill due to the exclusion of individual elements have a significant impact on load distribution and stress state of all other elements. Accounting for the real geometry of the structure, leads to computational difficulties, which are caused by the solution of nonlinear differential equations. The analytic solution of the problem for the entire system is missing due the marked above reason. The numerical method, based on the use of modern computing technics, was applied for the solution of the problem. The algorithm of that method is based on the theory of finite elements [1-3] and allows to obtain the solution of the problem, taking into account the marked nonlinearities. The computational model of individual structural elements of the rolling mill was formed on the basis of the finite element, which has the shape of a pyramid with ten nodes. The concept of the contact surface is used to describe the interaction between two contacting bodies. On this surface are located the nodes of contact finite element. If the mutual displacement of the nodes of contact finite element, which located on mutually opposite contact surfaces, not exceed the value of the gap Δ , the force of interaction in the contact finite element is zero. When the gap is less than zero, in the contact element appears force of contact interaction and its magnitude is proportional to the relative displacement of element nodes. Using a computational model of the nonlinear finite element assumes the availability of data on the magnitude of the gap Δ . The gap between the structural elements that are in the contact zone can be measured by the probe. The distance between the nodes of the contacting surfaces is determined in accordance with the geometry of the bodies. A characteristic feature of solution of contact problem is the determination of the size of the contact area (elastic flattening). The size of this area depends on the load, the geometry of contacting bodies and the mechanical properties of materials. The dimensions of the contact area are much smaller then dimensions of contacting bodies. The width of a possible contact zone in the first approximation can be determined by the formula of Hertz. This fact is accompanied by a choice of sufficiently small size of the volumetric finite elements adjacent to the contact area. In forming the finite element model of contacting bodies the width of the contact zone, obtained by the formula of Hertz, should be increased. It is necessary to account for the influence of uneven load distribution along the length of the contact area.

Figure 2 shows the mosaic, which characterizes the distribution of equivalent stresses in the contact zones of roller bearing. It is seen, that the area of largest equivalent stresses, which are marked red, is located in the contact zone between the roller and outer ring. The value of the maximum equivalent stress is equal to 548.6 MPa. In the opposite zone of contact between rollers and the inner ring, the similar stresses significantly below, their maximum value is 491.3 MPa. Equivalent stresses in the contact zones of the neighboring rollers with bearing rings considerably less, their maximum value does not exceed 122.1 MPa. With using of computer simulation, which was based on the proposed mathematical model, was revealed that the main reason for the destruction of a rolling bearing is the intense cyclically changing technological load. Changing the stiffness of the support ring does not give the desired result, but leads only to a relatively small increase in bearing life. Therefore, the basic idea of the new design of roll mill is to remove of bearings out of the working crate. Two parallel back-up

rolls were introduced in the construction of the roll mill instead of bearing. These back-up rolls are located between the bearing surface of the work roll and a bearing surface of rail. New construction of the working crate of rolling mill XIIT-250 is shown in Fig.3. Working crate includes in its membership: a motionless frame 1, on which is set crank drive 2; two upper rails, fixed on the unmovable frame; the two lower rails; two gear racks 5, which designed to drive in rotational motion of the working rolls, are also enshrined on the unmovable frame.

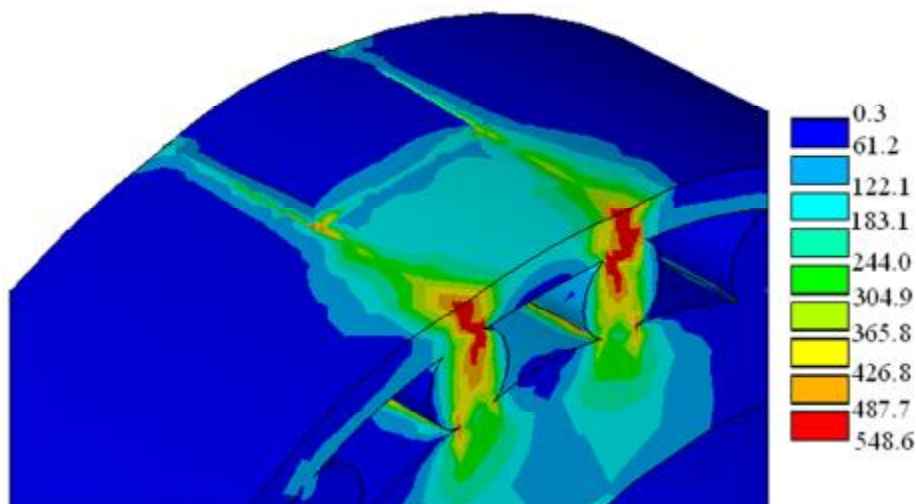


Fig.2 Distribution of equivalent stresses in the contact zones of roller bearing

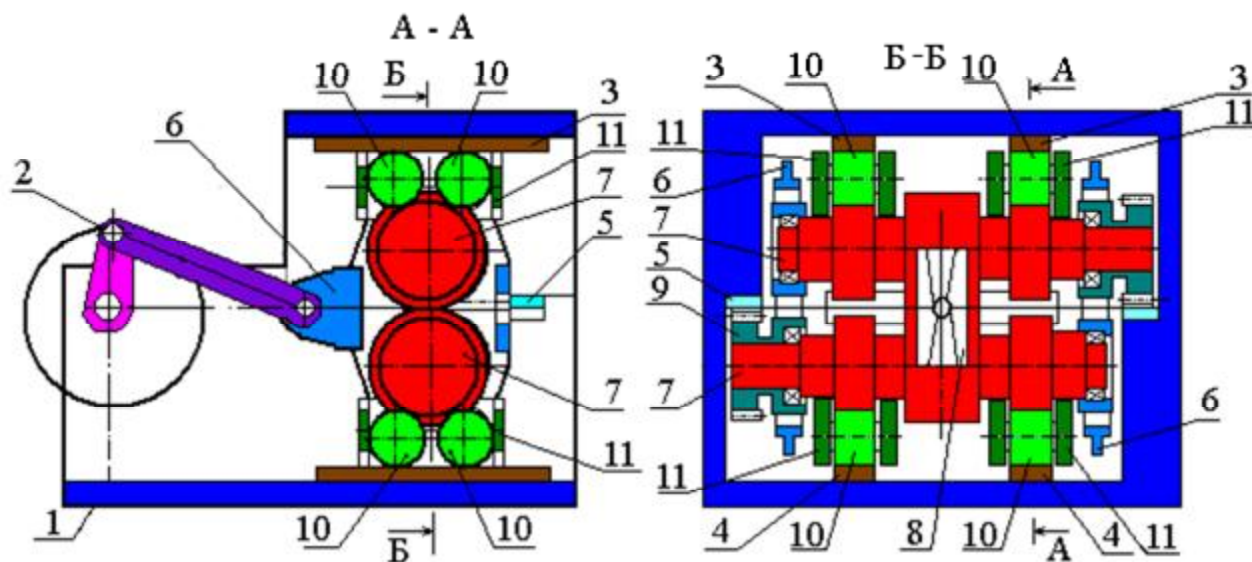


Fig.3 The new design of the working crate

Cassette with two working rolls 7 is attached to a crank drive. In the central part of each work roll is mounted caliber 8. The working rolls have the supporting surfaces, which located on both sides of the caliber. At the edge of the working rolls is located gear 9, which is in engagement with the gear rack 5.

Back-up rolls 10 have the opportunity for self-aligning supporting surface of work roll with one hand and with one of the supporting rails - on the other.

In the process of technological cycle crank mechanism drives the cassette with the work rolls. Due to the engagement gear mounted on the edge of the work roll, with the gear rack, the work rolls are reported rotational motion, as a result embodied in these gauges produce a rolling billet. The load, required for deforming billet is transferred from the caliber through the supporting surfaces of the working roll in the back-up rolls, and then on the support rails 3 and 4. Housing carriage 11, is not directly related to the cassette 6, allows the back-up rolls to hold the position of equilibrium between the rail and the supporting surface of work roll. Carriage design allows it to freely self-install under the influence of contact load and ensure an even distribution of pressure between the back-up rolls. This eliminates the undesirable effect of shifting of the longitudinal axis of the back-up rolls relative to the normal to the supporting surface of the rail. This fact contributes to further improve the accuracy of the size of rolled tubes.

2. RESULTS OF CALCULATIONS

The calculation scheme, which is shown in Fig.4, was used to calculate the stress-strain state of a new variant of design of rolling mill. Due to the symmetry of the construction of rolling mill and loading, its upper part, which includes a work roll, back-up rolls, rails and wedges, was considered. The wedges are needed to adjust the roll gap. The movable carriage, which are designed to fix the backup rolls were not considered in the finite element model. There are a few areas of contact in the above-described calculation scheme: between the working roll and the four back-up rolls, between back-up rolls and rails, between the rails and the wedges. Obtaining reliable results of the calculation of the stress-strain state requires consideration of all the contact zones of the analyzed structure.

Accounting for all areas of possible contact is accompanied by difficulties solving the problem of nonlinear elasticity. An analytical methods for solving the problems of this type does not currently exist.

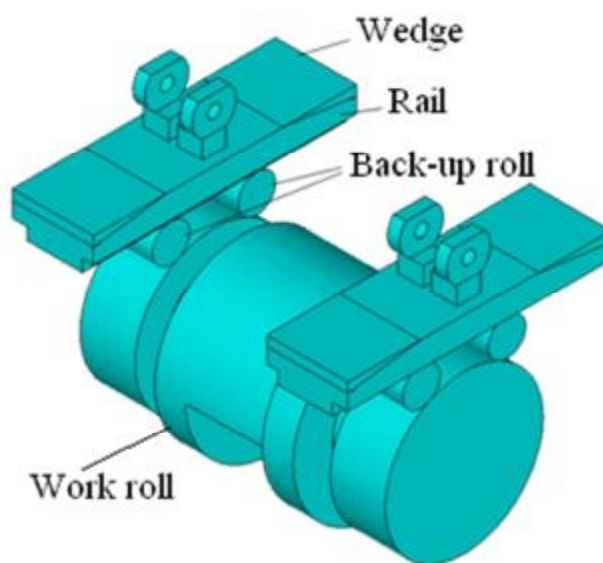


Fig.4 The calculation scheme of a new variant of the rolling mill

In some works, devoted to the study of the stress-strain state of the contact interaction, is used a simplified approach, which not lets to receive the reliable results. Using finite element method allowed us to obtain a solution of this complex problem in closed form, without the prior division into its constituent parts. Mathematical model of the structure, which above-described, is based on the using of volumetric finite element with ten nodes and a special contact element, designed to simulate areas of possible contact. The finite-element model of the analyzed structure is shown on Figure 5. The model includes in its membership 366112 nodes, 246760 volume finite elements, 10608 contact elements. The sizes of finite elements and the distance between nodes in the marked areas of possible contact, is much less, than in other parts of the model. This fact is important for obtaining reliable results of the calculation. Too little distance between nodes finite element model leads to a very large number of finite elements and, consequently, a large volume of computer memory to solve the problem. Initial distance between the nodes contact element on the contact surface was determined by the Hertz theory, then this distance was changed, and was assessed the influence of the number of nodes of the model at results of the solution. The optimal model for further calculations has been determined as a result of computer simulation.

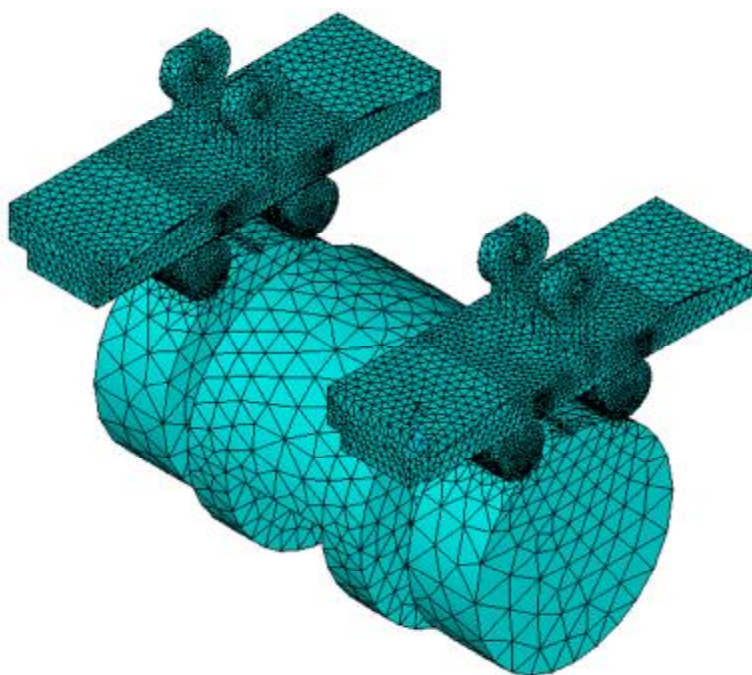


Fig.5 Finite-element model of the analyzed structure

The surface load, which is transmitted from the caliber on the work rolls, has been applied on the horizontal plane of the groove, which is intended for its attachment, and was determined by dividing the force of rolling on the area of its horizontal plane. The horizontal displacement of all nodes, located in the horizontal plane of symmetry of the back-up rolls, in the direction perpendicular to its longitudinal axis were equal to zero. Displacement of all nodes of finite element model, located in the vertical plane of symmetry of the work roll, in the direction of its longitudinal axis, has been equal to zero, due to symmetry of loading. Displacements of all nodes, located on the horizontal plane of the wedge, in vertical direction have been equal to zero. It was assumed, that the value of coefficient of friction, in the areas of possible contact, was equal to 0.14.

Mosaic of vertical displacements of parts of the analyzed structure is shown on Figure 6. It is seen, that the strain of work roll is accompanied by a pronounced bending. The region of the largest vertical displacements, marked in red color, observed in the middle of the work roll, in the zone of load application from the caliber. The maximum value of vertical displacement is equal to 0.551 mm. The similar magnitude of maximum vertical displacement in the original structure of the rolling mill, which was obtained from the numerical calculations, is equal to 1.038 mm. This fact indicates that the introduction into the design of rolling mill back-up rolls, allowed to increase its rigidity, and this, in turn, was led to an increase in the accuracy of the pipe. The vertical displacements in areas of contact work roll with back-up rolls, which were hindered its vertical displacement, is much less. Their values vary along the length of the contact area in interval from 0.054 mm to 0.338 mm. The displacement of areas of the working rolls, which are located on its support surface between the back-up rolls, more, than in the contact area. This fact can be explained by that the local displacement in the contact zone, caused by the interaction of contacting bodies, is influenced at the resulting value of displacement. The local flattening of the work roll in the contact zones with the back-up rolls is the result of the interaction, marked above. The analogical flattening occurs in the contact zone between the back-up rolls and the traverse due to the fact that the sizes of area of contact between these bodies are small.

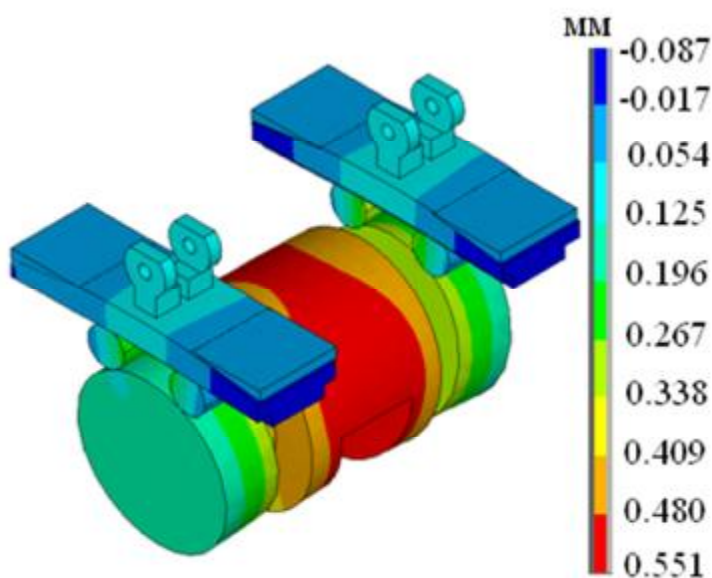


Fig.6 Mosaic of vertical displacements

Distribution of equivalent stresses in work rolls is shown in Figure 7. On the left side of this figure the distribution of equivalent stresses in the work roll is shown on the isometric view. On the right side of this figure is shown the pattern of distribution of equivalent stresses on the top view. It is seen, that the areas of largest equivalent stresses is located on the support surface in the contact zone with the back-up rolls. The stresses are distributed extremely uneven along the length of the contact bodies. Their maximum is observed in edge area, which is located closer to the place of installation of caliber or to the vertical plane of symmetry of the work roll and is equal to 378.56 MPa. The equivalent stresses on the opposite side of the contact zone significantly less, their maximum value is equal to 86.3 MPa. The width of the contact zone varies as well and has a maximum value at the edge, which closer to the vertical plane of symmetry.

The picture of the distribution of equivalent stresses in the rail is shown in Figure 8. Two bright bands, which correspond to zones of contact between the rail and the back-up rolls is clearly seen on this figure. The values of equivalent stresses in these areas are much higher than similar values in the other parts of the rail. Distribution of equivalent stresses along the length of this zone of contact is uneven also. However, this unevenness is less pronounced, in contrast to the previously considered unevenness in the contact zone between the work roll and backup rolls. The maximum value of equivalent stress is equal to 408.36 MPa and is observed at the left edge of the contact zone, which is closer to the caliber. On the opposite edge of the contact zone similar value of equivalent stress is equal to 283.21 MPa.

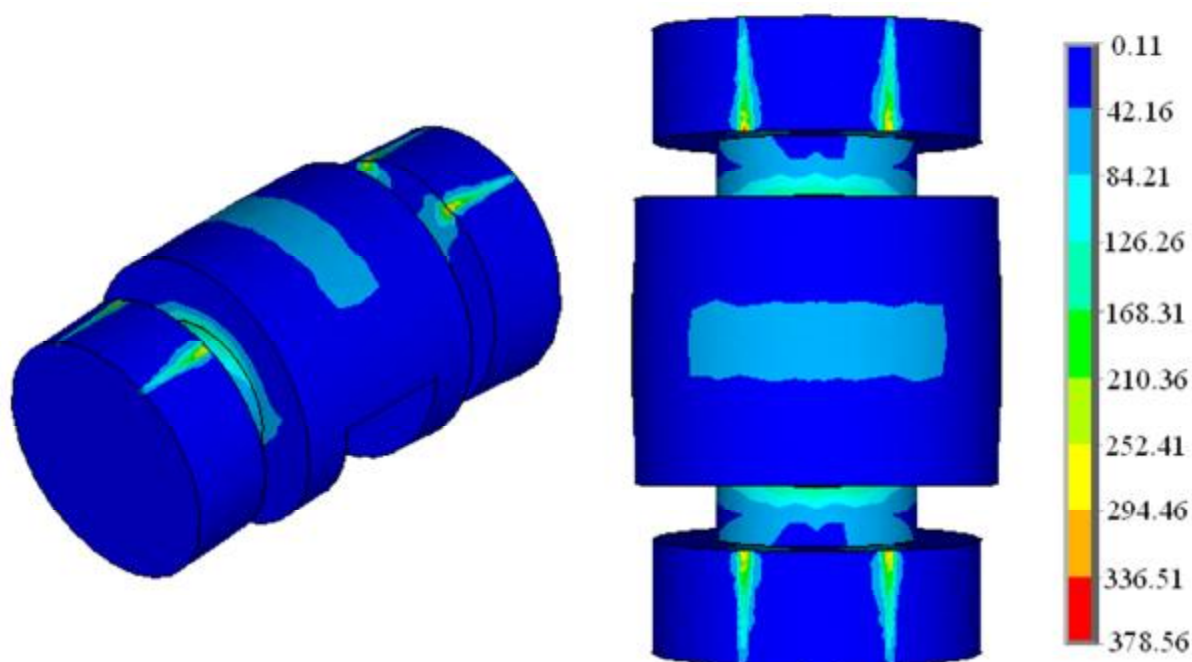


Fig.7 Distribution of equivalent stresses in work rolls, MPa

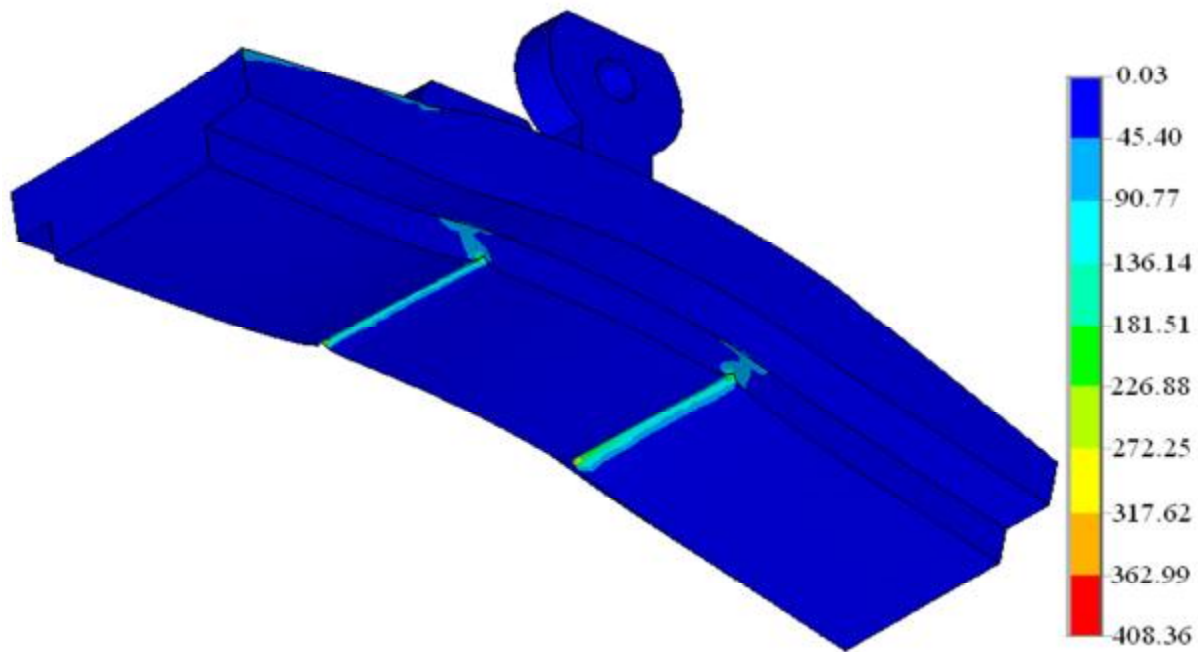


Fig.8 Distribution of equivalent stresses in the rail, MPa

CONCLUSION

Application of new actuator in the mill of XIIT-250 allowed for three years, more than in 20 times to lower expenses for support of its working capacity, firmness of basic plates has repeatedly raised, fields of deviations of average values of wall thickness of the rolled tube have decreased to 60 %. The use of two back-up rolls instead of rolling bearings has allowed three times to reduce the amount of deflection of rail compared with the original variants of the design and thereby improve the accuracy of the rolled tube.

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