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# INFLUENCE OF JOINT GEOMETRIC PARAMETERS ON OPERATION OF CAGELESS ROLLER BEARINGS

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*Predominant influence of roller chamfer angle on operation of cageless roller bearings has been considered. Structural assembly of support elements of such types is proposed.*

Working capacity of roller bearing is defined by stability of kinematic connections of roller with racetracks during product operation. Infringement of these connections leads to termination of roller motion in a bearing and, as consequence, to jamming of rollers. In a great degree it is promoted by changes in geometrical parameters of mobile connections because of symmetry axes skew of roller, cases, and changes of face, ring and diametrical backlashes in a bearing. These and other geometrical parameters are connected with constructive and technological factors defining life cycle of roller bearing. In turn the technological factors, considering variety of external influences and breaking symmetry of kinematic connections in mobile connections, form structure and orderliness of cageless roller bearing assembly [1, 2].

The purpose of research consists in definition of causally investigatory connection between geometrical parameters of connection details, explaining mechanics of face splits and roller jams. The specified purpose is reached under condition of the following research problems solution:

- dynamic model development of operational interaction of roller faces leading to bearing jam;
- definition at a stage of chisel manufacturing techniques of structurally technological parameter by means of which it is possible to stabilize wear processes, destructions and jams of roller bearings.

In the beginning for realization of such assembly technology by means of which there is a successful detail adaptation of operated mobile connections, it is necessary to reveal causal investigatory connections of connection destruction processes leading to item malfunction. The most acute problem of cause definition of heavy load product malfunction has to do with drill chisels. Resource tests of drill cutter chisels with cageless bearings of the PGV type have established the fact, that in most cases chisel malfunction depends on working capacity of multi-row roller bearings of a chisel and is defined by the time when rollers are jammed in a bearing.

Based on existing technology the details multi-row roller bearings of drill chisels: cutter – 4, claw pin – 3, big and small rollers – 2 and 1 (Figure 1) are produced with high accuracy (under finish H9), and the size of diametrical, ring and face backlashes is provided by the method of selective assembly. As the long-term operating experience of chisels shows, a principle of detail dimensional parameters stabilization used in manufacturing techniques, does not eliminate the risk from sudden roller jam in connection. By applying the technology of disorderly and unstructured assembly the chisel becomes nonoperatable, before preplanned resource ends.

In Figure 1 geometrical parameters are specified (face backlashes on small and big roller tracks $\Delta_T^M$, $\Delta_T^Б$; diametrical backlashes on small and big roller tracks $\Delta_Д^M$, $\Delta_Д^Б$; ring backlashes on small and big roller tracks $\Delta_K^M$, $\Delta_K^Б$; chamfers of small and big rollers $h\times\mu$, $H\times\lambda$; diameters of small and big rollers $d$, $D$; axes of symmetry according to the big roller track of the pin, the big roller track of the cutter, the big roller, the small roller, the small roller track of the cutter, the small roller track of the pin А, Б, В, Г, Д, Ж), participating in mechanics of cageless roller bearing destruction. As a result of uncontrollable technological parameters of assembly processes it is not obviously possible to predict approach time of the event connected to critical value change of these parameters while in service.

Those designs of drill chisels are investigated in the work, in which the influence of some balls – 5 (Figure 1) can be excluded from the scheme of roller bearing intense condition. The radial backlash in a roller track is equal to 0,07...0,08 mm, whereas in a ball – 0,3...0,5 mm. For this reason balls do not perceive operational loading in a chisel pore.

The carried out statistical analysis of destructions kinds of hard loaded bearings of drill chisels exhausted their resource has revealed dominating influence of destruction in types of chips along the edge of rollers formed by crossing of surfaces of the chamfer cone and the face. Such kinds of destruction are reflected in the photo (Figure 2) where chips along the edge of each roller are clearly visible.

Chipped fragments of rolling bodies, falling on the roller track, also promote bearing jam. It might be the reason which leads to infringement of a drill chisel working capacity. The reasons which lead to destruction in the form of chips along the edge of rollers can be investigated by means of the model describing deformation-strained conditions of connections constructed on principles of the final elements theory.

Intensive deterioration and chip of rollers along the edge and diameter changes the value of face, diametrical and ring backlashes concerning an initial condition of mobile connection, which leads to a roller turn of rollers in three-dimensional space of its own axis of symmetry B, and as consequence, to roller jam in one of three bearings of drill cutter chisel (Figure 3).

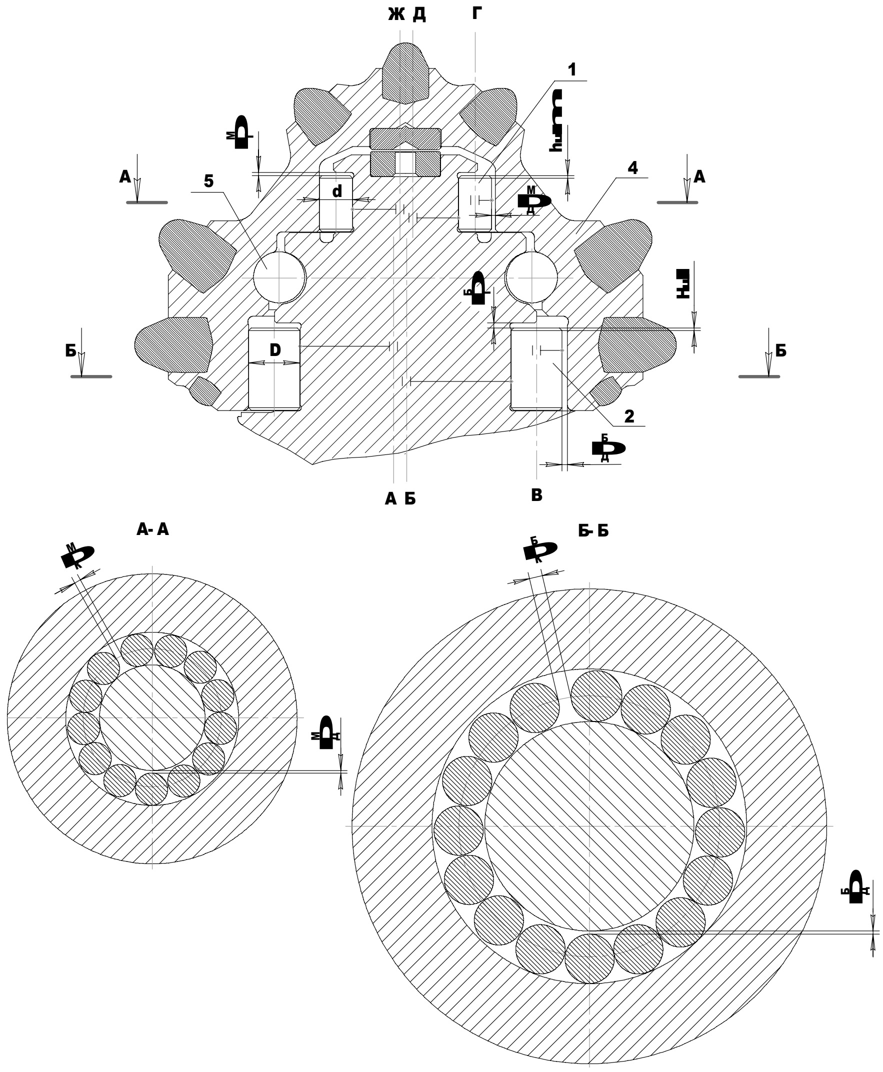


***Fig. 1.*** *Geometrical parameters of multi-row roller bearing of a drill chisel*

***Fig. 2.*** *Kinds of roller destruction*

***Fig. 3.*** *Jam of big rollers in a chisel cageless bearing*

The character of jammed rollers arrangement in used bearing and kinds of their deterioration specify connection of technological factor (change in position of symmetry axes B of rollers within the limits of ring, face and diametrical backlash areas) and design parameters of facets, with processes of rolling bodies destruction and bearing jam at interaction of roller faces with face surfaces of a drill chisel claw pin.

On the basis of stable character of bearing malfunction for the reasons of jamming (Figure 3) and roller destructions (Figure 2) the criterion of the maximal contact pressure the roller $K_\sigma$ estimating working capacity of mobile connection, is chosen.

At the first stage of mathematical 3D model construction it is necessary to present connection of a roller with a racetrack of a claw paw in an electronic kind using applied packages of software products CAD CAM and interactive system of design automation engineering analysis and manufacturing – Unigraphics NX.

At the next stage constructed mathematical 3D models are imported to the program ANSYS which represents a multi-purpose package of designing and analysis of the deformation-strained connection condition. In the above-stated program efforts which act on rollers during chisel operation are applied to the imported mathematical model of a roller. These efforts aspire to move the roller around on its own axis of symmetry. Further the responses of the system to the enclosed efforts in the form of pressure distribution are considered.

For management of roller destruction process and their jamming at the stage of roller bearing fabrication it is necessary to reveal the design parameter of the roller which is responsible for its further work and to offer the design and technological solution on risk decrease of a chisel malfunction for this reason. The angle of a roller facet has been chosen as design parameter.

At initial angle value of a roller facet $\lambda$ equal to 45°, set by the designer, the maximal pressure arising in a place of roller contact with face surfaces of a roller claw pin $\max_{(45)}K_\sigma$=893 MPa (Figure 5).

Let's change the value of facet angle variable parameter in the model having increased its value by 15°, i. e. we shall make it equal to 60° and analyze the model. The maximal pressure which has occurred at roller faces, has made $\max_{(60)}K_\sigma$=812 MPa, which is 10 % less than in the case of a facet with 45° angle.

In Figure 4 the schedule of maximal pressure change $K_\sigma$ from the corner of roller facet is presented.

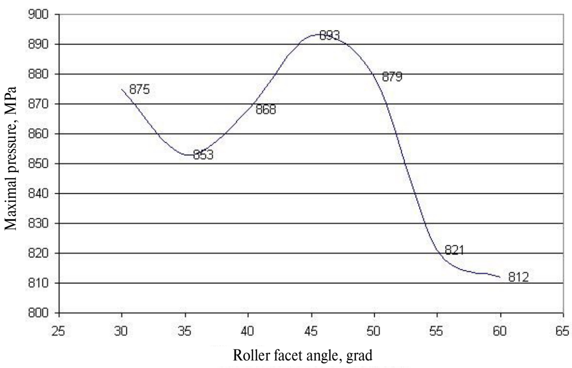


***Fig. 4.*** *Dependence of the maximal pressure occurring in a contact place of roller and face surfaces of a claw pin from the corner of roller facet*

Then we shall simulate a roller with different facet angles. Let's make it 30° from the top, 60° from the bottom we shall analyze the model. The maximal pressure which has occurred in a place of contact, has made $\max_{(30\times60)}K_\sigma$=732 MPa, which is 18 % less than in the case of initial facet with an angle 45°, set by the designer.

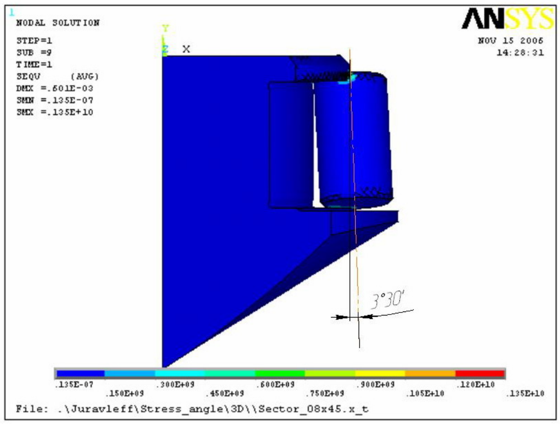


***Fig. 5.*** *Pressure intensity at interaction of a roller with bearing details*


***Fig. 6.*** *Jammed bearing of a mining drill chisel*

Apparently from Figure 5, at interaction of a roller with facet angle equal to 45° according to the design documentation with face surfaces of a claw pin it is pushed out from the surface of a racetrack. In this case there is a skew of roller axis B in relation to A and B axes of symmetry on the corner 3°30′, which leads to substantial increase of specific pressure upon a pin face surface and intensive deterioration of these surfaces (see traces of

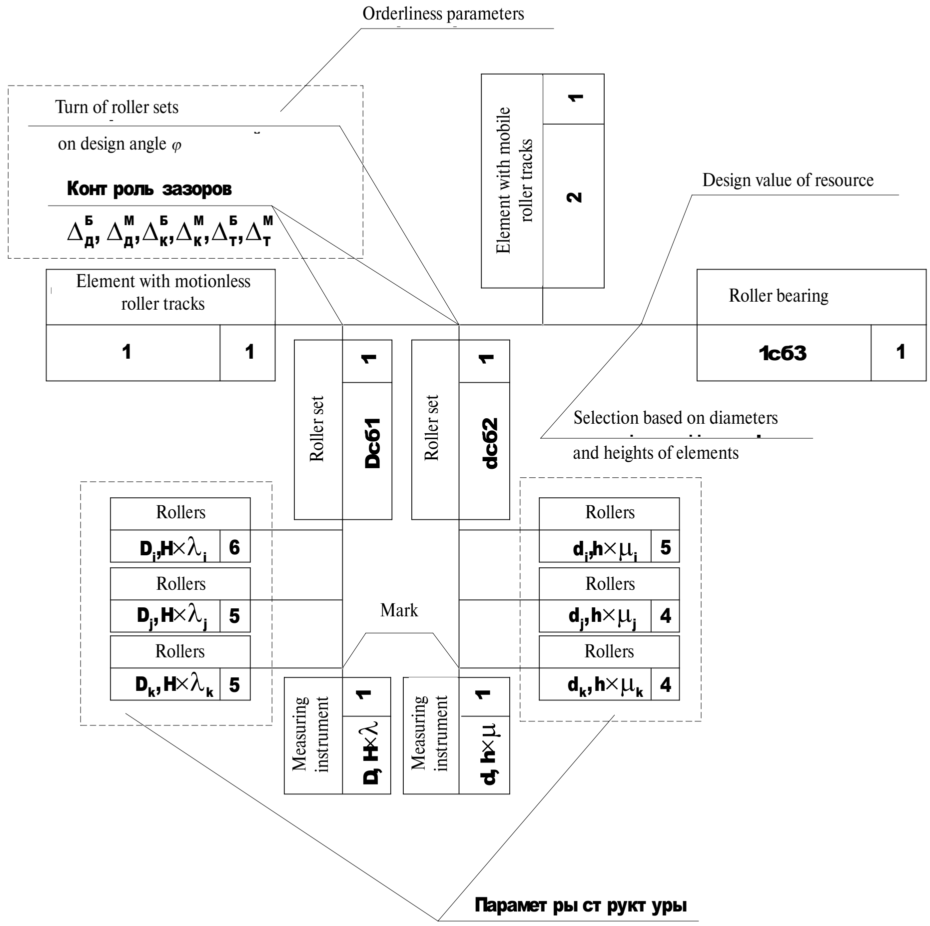


***Fig. 7.*** *The scheme of the central ordered assembly of multi-row roller bearings: i, j, k is designation of selective group*

deterioration, Figure 3). Besides, the skew of roller axis causes displacement of roller face surface which value blocks values of face backlash $\Delta_T^Б$. If displacement is more than diametrical backlash in a roller bearing $\Delta_Д^Б$, then the roller will rest against a cutter roller and will create at this contact the additional moment of forces strengthening the effect from a skew which leads to roller bearing jam and drill chisel failure as a whole.

Reliability of results obtained by mathematical modeling proves to be true by concurrence to data of field tests. At roller jam (Figure 6) their extraction from racetrack surfaces of a chisel claw pin is visible, and the value of a skew is equal to 3°40′.

The carried out researches of the intense condition of roller bearings have revealed the following properties mobile connections in a roller bearing:

- dominating influence of facet parameters along the face surface of rollers contacting to a motionless fillet of a claw pin and mobile cutter of a drill chisel, leads to asymmetry of destruction processes;
- asymmetry of the maximal value of roller plastic deformation is defined by displacement of max $K_\sigma$ towards mobile face of chisel cutter relatively to a plane of roller symmetry.

These facts established by means of model, allow developing the ordered assembly of roller bearings in view of optimum structure properties of detail interaction. In Figure 7 the scheme of central assembly of multi-row roller bearings of the cageless type in view of optimum structure of detail interaction [3, 4] is presented.

As a result of the carried out researches the following has been revealed:

1. Constructive optimum parameter of rollers of big and small roller tracks should be a variable corner of roller facet (30° and 60°).
2. The complete set of rollers at their installation onto roller tracks should be carried out by the method of structural orderliness, i. e. the face of a roller with smaller angle of a facet should contact to a cutter fillet, and with greater angle of a facet with face of a pin.
3. Presence of constructive changes of facets will allow applying the structure into assembly process of roller arrangement in a bearing relatively to their plane of symmetry and stabilizing the process of roller support detail interaction, providing stability of kinematic connections.

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## MATHEMATICAL MODEL OF MECHANICAL PARTS INTERCONNECTED ELECTRIC DRIVES OF HEAD AND LIFT IN ROCK DIGGER


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*The need for development of new approach to modelling lift and head electric drives of quarry digger in the process of digging has been justified. The differences of the mathematical model suggested from the traditional approach are shown. The disadvantages of existing control systems in digger electric drives are revealed, the method of their removing is proposed.*

The important link in technological chain of open-pit mining for minerals is rock excavation. At the same time, recently there has been a decrease in the basic technical and economic parameters of quarry equipment utilization, including quarry diggers. It can be partially explained by insufficient reliability of operated machines. Search of ways to further increase technological level of career diggers demands mathematical model development adequately describing their basic working processes.

In order to describe dynamic processes taking place during work of quarry diggers, kinematic schemes of its drives are represented in the form of multimass mechanical systems. At the same time, the approach according to which full design schemes of drives are simplified up to two-mass [1, 2] is rather widespread. Parameters of such design schemes are thought to be constant. However, in actuality, design scheme parameters of digger drives is changed during the work. Thus, application of the standard approach leads to mistakes at loading calculation of specific joints in quarry diggers and complicates researches addressed to increase their reliability.

The most complex operation made by quarry digger is the process of digging, from the point of view of external loadings occurrence. In this connection, description of quarry digger work exactly in this mode is rather actual problem.

During digging two electric drives take part simultaneously: lift and head, therefore at creating of the described mathematical model it is necessary to consider both kinematic features of electric drives and their mutual connection. Thus the greatest role will be played by change in parameters of electric drives, caused by change in geometrical position of ladle and lever.

The simplified design scheme of interconnected electric drives of lift and head is shown in Figure 1. On this scheme the following designations are accepted: $J_н$ is the total reduced inertia moment of the first mass of head drive, including inertia moment of engine rotor, reducer and head drum; $c_н$ is the reduced rope rigidity of head mechanism; $J_п$ is the total reduced inertia moment of the first mass of lift drive, including inertia moment of engine rotor, reducer lift drum; $c_п$ the is total reduced rope rigidity of lift mechanism; $m_р$, $m_к$, $m_п$ are the masses of lever, ladle and rock respectively; $M_н$ is the electromagnetic moment of head engine, reduced to speed of head drum; $M_п$ is the electromagnetic moment of lift engine, reduced to speed of lift drum; $\omega_н$, $\omega_п$ are the angular speeds of the first weight of head and lift drive respectively; $F_{сн}$, $M_{сп}$ is the force and resistance moment of head and lift drive respectively; $v_к$ is the linear speed of ladle head; $\omega_к$ is the angular speed of ladle and lever.

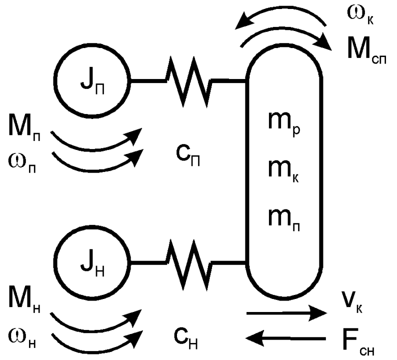


***Fig. 1.*** *Design scheme of lift and head electric drives in a quarry digger*

It is necessary to consider the design of ladle suspension and lever mounting for definition of digger parameters de-