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Analysis Of The Stressed State Of Bearing Elements Of The Support Of The Rolling Cutter Bit.

Vladimir A Pyalchenkov*, Vladimir V Dolgushin, Gennadiy A Kulyabin, and Lybov I Nikitina.

Industrial University of Tyumen, Volodarskogo str., 38, Tyumen, 625000, Russia

ABSTRACT

The durability of cutting structure and support units of rolling cutter bits depends on the magnitude of the forces acting on the bit elements. A considerable amount of analytical and experimental research has been devoted to the study of the regularities in the distribution of forces along the cutters of the cutting structure and the bearings of the rolling cones of the drill bits. In the article, the model of a rolling cutting bit is considered, which is an non-deformable roller housing fixed on three roller bearings on a deformable bearing pin. When studying the model, contact deformations of the bearing elements are taken into account, and the clearances in the bearings are assumed to be zero. The results of an analytical study on the load distribution between support bearings of a rolling cutter bit and the calculation of the contact stresses in bearings are presented, which their durability depends on. From the conditions of the rolling cone equilibrium and the conditions of joint deformation of the support elements, dependences are obtained, which allow determining the values of the reactions in bearings and those of contact stresses for different variants of applying an external load to the rolling cone. The results of the calculations allow us to conclude that the greatest contact stresses arise in a retaining drilling bit bearing. These results are confirmed by experimental studies on the photo-elastic model of the support unit of the rolling cone. The proposed technique can be used to optimize the design of the cutting structure and rolling cutter bit supports.
Keywords: drilling, drill bit, rolling cone, support, load, durability, contact stresses.

Corresponding author

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INTRODUCTION

The reliability and durability of drill bits depend on the force value acting on the rolling cone cutters directly affecting the rock. A considerable number of both analytical and experimental studies have been devoted to investigations of the regularities of the interaction of the cutting structure of rolling cutter bits with a bottom hole. The model of the interaction of the bit cutting structure with a rock is proposed in paper[1]. The bit cutter affects the rock, making a complex movement of slippage of the rolling cone along the bottom hole depending on the rotation parameters of the rolling cone and bit. Experimental studies of the interaction of separate elements of the bit cutting structure with the rock are performed according to the drill chart with a single cutter. The results of an experimental study of the interaction of the bit with the rock are presented in papers [2,3].In paper [4], the axial force acting on the bit from the side of the rock to be destroyed is determined. In papers [1, 5] analytical dependencies are proposed for determining the velocities of impact and motion in contact with the bottom hole of cutting structure elements of a rolling cone, as well as nonlinear dependences between the rotation angles of the crown around its axis and around the axis of the bit when working on a deformable bottom hole. Using the wear resistance as a criterion of optimization in this model, the author determines the optimal ratio of geometric parameters due to their variation [6].In paper [5] the model is considered, which is a set of interconnected modules for calculating the kinematics and bit dynamics, vibrations of the drill string, as well as the formation and deepening of the bottom hole during drilling. The model allows to determine the distribution of forces for any moment of drilling time and displacements of any point of the drilling tool for a given combination of the bit design parameters, drill string, mode and drilling conditions, beginning from the upper end of the drill string and ending with the cutter tops of the bit rolling cones. Determination of the reactions in the rolling cutter bit supports and the contact stress values in the support bearings under different variants of the interaction of rolling cone cutting structures with the bottom hole is an important practical task, which solution will allow optimizing the construction of both supports and cutting structure. Various mathematical models of rolling cutter bits for the analytical determination of the forces acting on support bearings of the rolling cones are proposed in the papers [5,7]. In paper [8], a technique, a set of devices and means of measurement were proposed for the experimental determination of the loads perceived by each rolling cone during their work at the bottom hole. The technique consists in the experimental determination of the load acting on each section of the rolling cutter bit model, followed by an analytical evaluation of the loads distribution between the support bearings. A number of experimental studies performed on various devices, for example papers [9, 10], are devoted to solving this problem. In this paper, a version of its analytical solution is presented.

METHODS

As a model, we will consider a rolling cone mounted on a bearing pin on three roller bearings and loaded with an axial force P applied at a distance R from the axis of the bit (Fig.1).

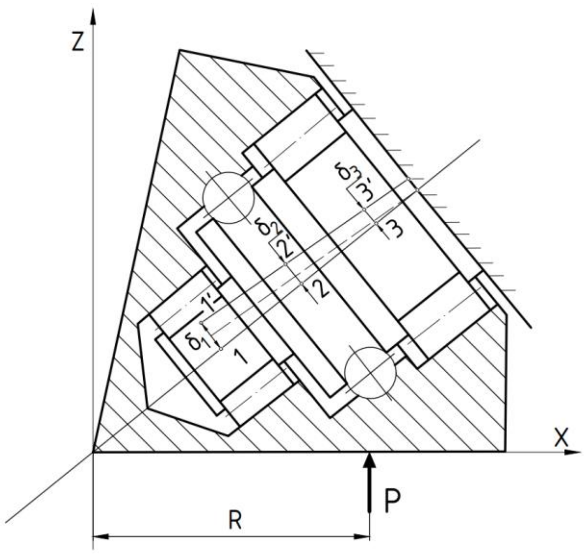

Fig 1: Design scheme of a rolling cone unit

Since the axial moment of the cross section resistance of the rolling cone body is a multiple of the axial resistance moment of the bearing pin cross section, it can be assumed that the deformation of the rolling cone body will be insignificant in comparison with the deformation of the bearing pin and the contact deformations in the bearings and can be neglected. The support bracket (with the exception of the bearing pin) will also be considered to be non-deformable. In order to determine the reactions values in the bearings, let us consider the condition of the rolling cone equilibrium. The cutter must be in equilibrium under the action of an external force P and reaction forces in the bearings. Let us assume that the peripheral and end roller bearings perceive only the radial loads and retaining ball bearing perceives the axial and radial ones (Fig. 2).

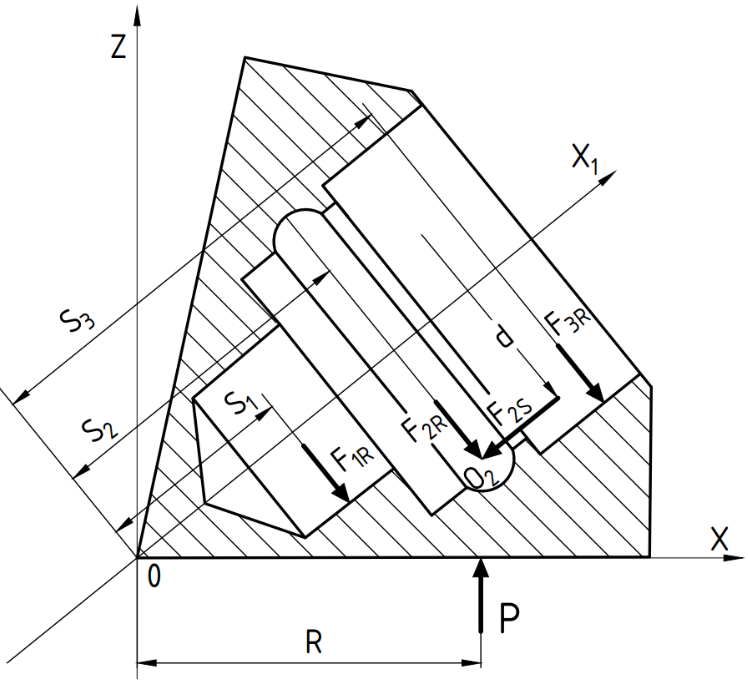

Fig 2: Scheme of forces acting on the rolling cone

For a given forces system, it is possible to form three analytic equilibrium conditions:

$$\sum Z_{1i} = 0; P \sin\varphi_0 - F_{1R} - F_{2R} - F_{3R} = 0, \quad (1)$$

$$\sum X_{1i} = 0; P \cos\varphi_0 - F_{2S} = 0$$

$$\sum M_{2i} = 0; P(R - S_2 \sin\varphi_0 - \frac{d_2}{2}\cos\varphi_0) + F_{1R}(S_2 - S_1) - F_{3R}(S_3 - S_2) = 0$$

Since equations (1) are insufficient for the determination of all unknowns, the given system is statically indeterminate and for its solution it is necessary to compile additional equations considering the system deformation. We assume that the radial clearances in all bearings are equal zero, since in the presence of gaps, the degree of static indeterminacy of the system increases, and its solution becomes much more complicated. Before the load is applied, the details of the support do not deformed and the axes of the bearing pin and the rolling cone coincide. Let us consider the bearing pin as a cantilever beam, rigidly pinched by one end. When applying the load the bearing pin is deformed, and its axis occupies a new position. Since the rolling cone is not rigidly connected to the bearing pin, their axes do not coincide after deformation (Fig. 1).And if the axis of the bearing pin is curved after the deformation, then the rolling cone axis remains straight due to earlier assumptions. From the condition for the similarity of triangles (Fig. 1), the expression can be obtained:

$$\frac{\delta_1 - \delta_3}{S_3 - S_1} = \frac{\delta_2 - \delta_3}{S_2 - S_1}, \quad (2)$$

It is convenient to express the formula (2) as follows:

$$\delta_2(S_3 - S_1) = \delta_3(S_2 - S_1) + \delta_1(S_3 - S_2), \quad (3)$$

From the displacement scheme of the rolling cone (Fig. 1) it follows relative to the bearing pin that:

$$\delta_1 = \delta_{1y} + \delta_{1k}, \quad (4)$$
$$\delta_2 = \delta_{2y} + \delta_{2k}, \quad (5)$$
$$\delta_3 = \delta_{3y} + \delta_{3k}, \quad (6)$$

where:

$\delta_{1y}, \delta_{2y}, \delta_{3y}$ - displacement of points 1,2 and 3 as a result of elastic deformation of the bearing pin;
$\delta_{1k}, \delta_{2k}, \delta_{3k}$ - displacement of points 1,2 and 3 as a result of contact deformations in a small roller, ball and large roller bearing.

RESULTS

Let us consider the deformation of the bearing pin as a cantilever beam clamped in the support bracket (Fig.3).

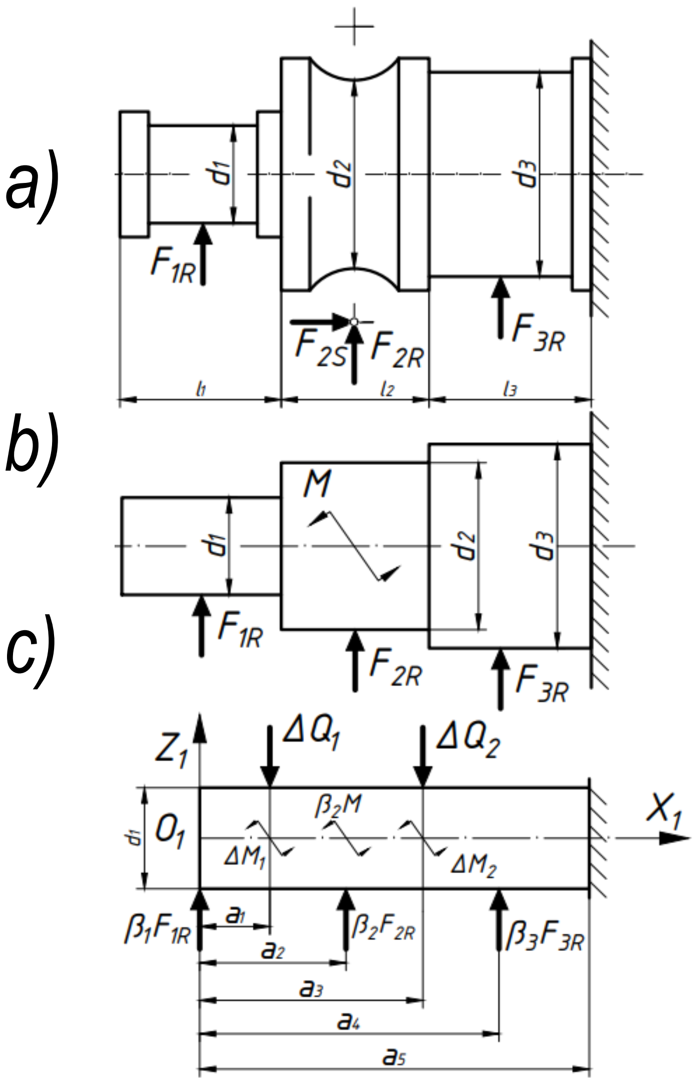

Fig 3: Scheme for calculating the deformation of the bearing pin

Obviously, without introducing a significant error in the result, the bearing pin of the actual shape (Fig. 3a) can be replaced by the bearing pin of a simplified shape (Fig.3b), having three cylindrical sections, which diameters are equal to the diameters of the races on the bearing pins of a small roller, ball and large roller bearing. The force F_{2S} is transferred to the axis of the beam, adding a couple of forces with the moment:

$$M = \frac{F_{2S} * d_2}{2} \qquad (7)$$

Since the force F_{2S}, transferred to the beam axis, does not affect the beam bending, then this force will not be taken further into account. Let us define the beam deflections at the points of forces application F_{1R}, F_{2R} and F_{3R}. The determination of displacements by the method of direct integration of the differential equation of the elastic beam line for the case under consideration is connected with considerable difficulties. They consist in the necessity of compiling and solving a system of linear algebraic equations for determining the arbitrary integration constants. The number of equations constituting the system is defined as 2xn, where n is the number of sections which the beam is divided to. In our case, it would be necessary to solve a system of ten equations. Therefore, to determine the displacements at the points of interest to us, let us use a modified method of initial parameters [11]. Let us transform the stepped beam (Fig. 3b) into an equivalent beam with constant cross section with a moment of inertia I_0 equal to the moment of inertia of the first section of the beam.To ensure that the beam of constant cross section is equivalent to the original one, it is necessary to multiply all the acting external loads by the corresponding reduction coefficients β, and add additional loads ΔQ and ΔM at the interfaces of different sections (Fig. 3c).

Determination of the coefficients of reduction and detailed calculations are given in the paper[12]. After performing the transformations and introducing a number of additional notation, the final expression for determining the deflections of the bearing pin were obtained:

$$\delta_{1y} = B_1F_{1R} + B_2F_{2R} + B_3F_{3R} - B_4F_{1S}; \qquad (8)$$
$$\delta_{2y} = B_5F_{1R} + B_6F_{2R} + B_7F_{3R} - B_8F_{1S}; \qquad (9)$$
$$\delta_{3y} = B_9F_{1R} + B_{10}F_{2R} + B_{11}F_{3R} - B_{12}F_{2S}; \qquad (10)$$

Coefficients $B_1 - B_{12}$ are constant values, depending on the geometric dimensions and material of the bearing pin.

Let us turn to the determination of the displacement of the points 1,2,3 (Fig. 1), which occur as a result of the total contact deformations in the bearings, which consist of deformations due to contact of the rolling bodies with the rolling cone and the bearing pin. So in the small roller bearing it is:

$$\delta_{1k} = \delta_{1kc} + \delta_{1ks}; \qquad (11)$$

There are:
δ_{1kc} –displacement of the point 1 as a result of contact deformation of rollers and rolling cone;
δ_{1ks} –displacement of the point 1 as a result of contact deformation of rollers and the bearing pin.

The deformation will be determined at the contact point of the most loaded roller the force to which at a normal radial gap, is determined as follows [13]:

$$Q_{1max} = \frac{5F_{1R}}{z_1} \qquad (12)$$

Where:
Z_1 –number of rollers in the bearing.

When the rollers come into contact with the bearing pin, which corresponds to the case of external contact of two cylinders with parallel axes of the same material with a Poisson's ratio M = 0.3, the axial approach can be determined from the expression obtained by B.S. Kovalski [14], which takes into account not only the deformation at the contact, but also the total deformation of the cylinders:

$$\delta_{1ks} = 0{,}579\frac{q_1}{E}\left[\ln\frac{d_{y1}d_{P1}}{l_{P1}^2} + 0{,}814\right]; \qquad (13)$$

Where:
l_{P1}, d_{P1} –length and diameter of the roller;
$q_1 = \frac{Q_{1max}}{l_{P1}}$ –load per unit length of roller.
E = 210 – hPa - modulus of elasticity of steel.

After substitution and transformation, we obtain:
$\delta_{1ks} = B_{13} * F_{1R}$ (14)
$B_{13} = \frac{2.895}{l_{P1}Z_{P1}}\left[\ln\frac{d_{y1}d_{P1}}{l_{P1}^2} + 0.814\right];$ (15)

When the rollers come into contact with the rolling cone, which corresponds to the case of a cylinder contact with a cylindrical cavity, the convergence can be determined [15] from expression:

$\delta_{1kc} = \frac{2(1-\mu^2)}{E}q_1(1 - \ln c);$ (16)

Where:
μ - Poisson's ratio;
$c = 1.08\sqrt{\frac{q_1}{E} * \frac{d_{c1}d_{P1}}{d_{c1}-d_{P1}}};$ (17)

Taking into account that in our case the diameter of the cylindrical cavity d_{c1} is several times larger than the diameter of the cylinder d_{P1}, to calculate the displacement δ_{1kc} the empirical formula can be used [16], which is valid for the contact of a steel cylinder with a plane:

$\delta_{1kc} = \frac{0.462}{10^3\sqrt[3]{d_{P1}}} * \frac{Q_{1max}}{l_{P1}};$ (18)

Calculations showed that using the expression (18) instead of the expression (16) with the existing proportions of linear dimensions in support bearings of the bit support introduces an error, which value does not exceed 2.5%. Therefore, we will use expression (18), presenting it in the form:

$\delta_{1kc} = B_{14} * F_{1R};$ (19)

Where:
$B_{14} = \frac{2.31*10^{-3}}{\sqrt[3]{d_{P1}}Z_1 l_{P1}};$ (20)

Thus, the total contact deformation in a small roller bearing is determined:

$\delta_{1k} = F_{1R}(B_{13+}B_{14});$ (21)

Reasoning similarly, let us define the total contact deformation in a large roller bearing:

$\delta_{3k} = F_{3R}(B_{15+}B_{16}),$ (22)

Where:

$B_{15} = \frac{2.895}{l_{P3}ZE}\left[\ln\frac{d_3 d_{p3}}{l_{P3}^2} + 0.814\right]$ (23)
$B_{16} = \frac{2.31*10^{-3}}{\sqrt[3]{d_{P3}}Z_{P3}l_{P3}}$ (24)

d_{P3}, l_{P3}, Z_{P3} –the diameter, length and number of rollers in a large bearing,respectively.

Let us further consider the ball bearing. The total contact deformation is determined for it as follows:

$\delta_{2k} = \delta_{2sk} + \delta_{2ck};$ (25)

Approximation of two steel bodies with point contact can generally be determined by the method described in the paper [16]. Detailed calculations are given in the paper [12]. Substituting the obtained expressions for displacements in (3) and carrying out the transformations, we obtain the required condition establishing the relationship between the unknown reactions in the support bearings:

$$B_{18}F_{1R} + B_{19}F_{2R} + B_{20}F_{2R}^{\frac{2}{3}} + B_{21}F_{3R} - B_{22}F_{25} = 0; \quad (26)$$

Where:

$B_{18} - B_{22}$ −constant coefficients, depending on the dimensions and material of the support elements.

Taking into account the equations system (1) and introducing a notation number the expression (26)is transformed. An equation of the third degree with one unknown F_{2R} is obtained

$$F_{2R}^{3} + C_1F_{2R}^{2} + C_2F_{2R} + C_3 = 0; \quad (27)$$

Where:

C_1, C_2, C_3 −coefficients which depend on the dimensions and material of the support elements, as well as on the size and radius of the force application P.

DISCUSSION

Equation (27) can have one or three roots [17]. When solving it, only one largest algebraic value was taken into account, the value F_{2R}. The values of the remaining reactions in the bearings are determined from equations:

$$F_{25} = B_{24}P; \quad (28)$$

$$F_{3R} = B_{26}P - B_{27}F_{2R};$$

$$F_{1R} = B_{28}P - B_{29}F_{2R};$$

It is not possible to obtain explicitly the dependence of the reaction quantities in the support bearings on the force P and the radius of its application R in explicit form. Therefore, these dependencies were obtained in the form of tables of reaction values for different P and R for the 215.9K-PV bit support. Based on the calculation results, graphical dependencies were constructed. When the radius R changes the load between the bearings is redistributed (Fig. 4).

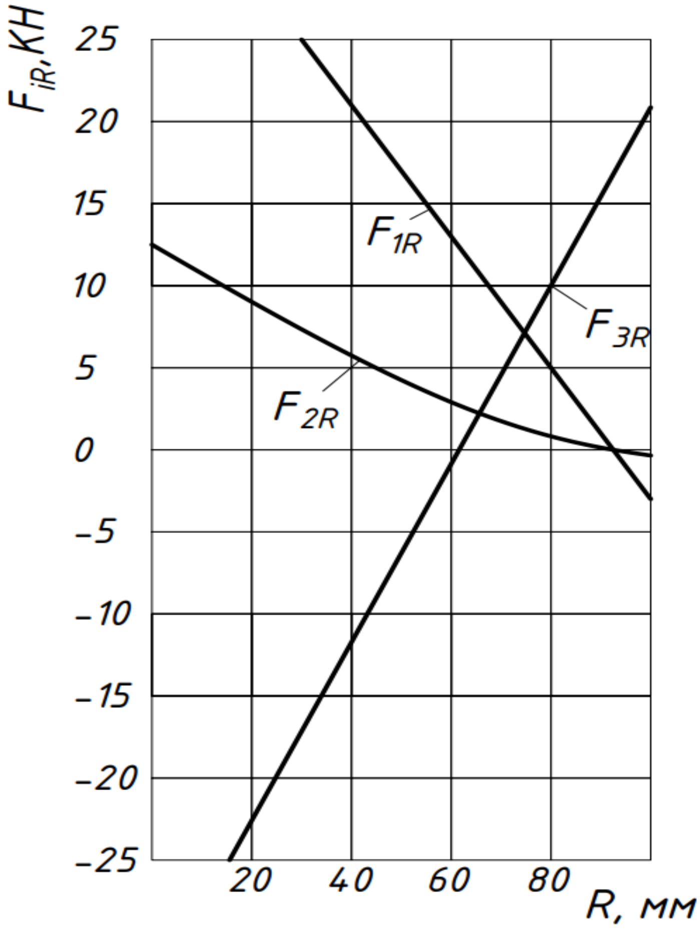

Fig 4: Calculated dependencies of radial forces in bearings on the radius of application of external force (P=10 kN)

For a small radius (R <60 mm), the calculated radial load on the peripheral roller bearing is negative, i.e. the load is perceived by the upper rollers and not by the lower ones. When displacement of the point of application of the force acting on the roller cone from the bit axis to the periphery, the load acting on the end roller bearing and the retaining roller bearing decreases, and the load on the peripheral roller bearing increases. In the case when the action line of the external force passes through the center of the most loaded lower ball, the load is distributed between all three support bearings, which significantly increase the contact rigidity of the support unit. When changing the position of the load point, the load is redistributed between the bearings.

However, the value of the total radial load cannot sufficiently characterize the stressed state of the bit support elements.

Using the obtained dependences to determine the radial components of the reactions in the roller bearings F_{1R}andF_{3R}, radial F_{2R} and axial F_{2S} making up reaction components in the ball bearing, let us determine the contact stresses value at the contact point of the most loaded ball and the rollers with the bearing pin. For the ball bearing it is [14]:

$$\sigma_{H2} = 0{,}408 \cdot 10^4 \frac{\Sigma \rho_2^{\frac{2}{3}}}{\mu \cdot \nu} \cdot F_{2\max}^{1/3} \qquad (29)$$

Where:

$\Sigma\rho_2$ - the sum of the curvatures of the contacting surfaces;

$F_{2\max}$ - force acting on one ball.

For roller bearings, the greatest contact stresses will be determined [14]:

$$\sigma_{H1} = 86{,}1\sqrt{\frac{\Sigma\rho_1}{2}\cdot\frac{F_{1\max}}{l_1}}\,; \qquad (30)$$

$$\sigma_{H3} = 86{,}1\sqrt{\frac{\Sigma\rho_3}{2}\cdot\frac{F_{3\max}}{l_3}}\,; \qquad (31)$$

Where:

$F_{1\max}$, $F_{3\max}$ - force acting on one bearing roller;

l_1, l_3 - roller length.

The results of calculations for the axial load on the rolling cone **P**=10 kN are presented in Fig. 5.

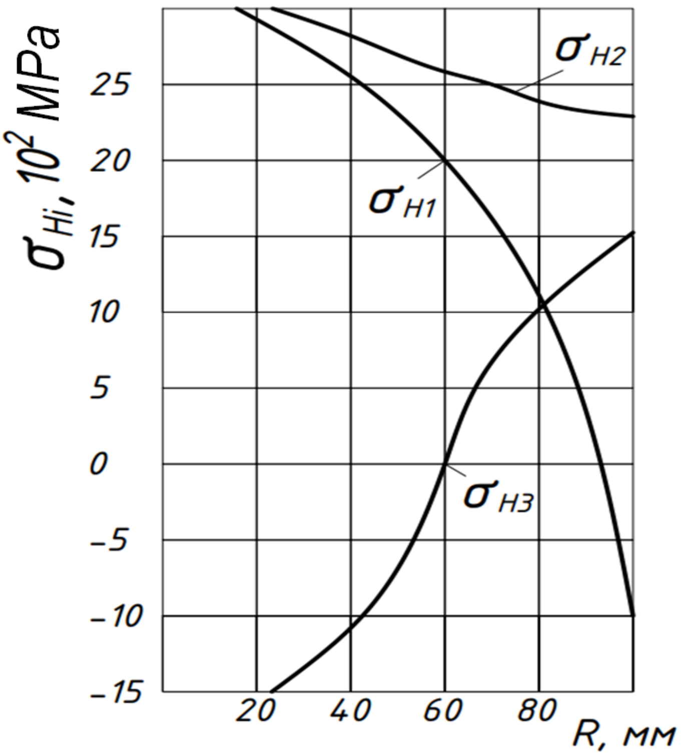

Fig 5: Influence of the application radius of the external load on the calculated value of the maximum contact stresses in the support bearings

(**P** = 10 kN).

CONCLUSION

The largest contact stresses at any application radius of the external load to the rolling cone arise in the retaining ball bearing. Thus, the calculations results qualitatively correspond to the conclusions of a number of researchers who maintain that the retaining ball bearing of the rolling cone support operates under the most severe conditions and limits the durability of the rolling cone support unit.

For experimental verification of the obtained analytical dependences, the polarization-optical method as one of the most modern and obvious methods of studying the actual stress state of complex structures was used. From an optically sensitive material, a flat model of a roller unit was manufactured, which dimensions correspond to the actual size of the supporting elements of the bitSH215,9K-PV. The load of each bearing can be estimated on the band order at the most stressed points of the rolling element models (rollers and ball).Let us considerFig.6. In this case, all three models of rolling bodies are loaded. The greatest difference in principal stresses arises at the center of the model of the ball, where the band order ism=5.5. The smaller bands order is observed in roller models (m=1.5 for a small roller and m=1.3 for the big roller).

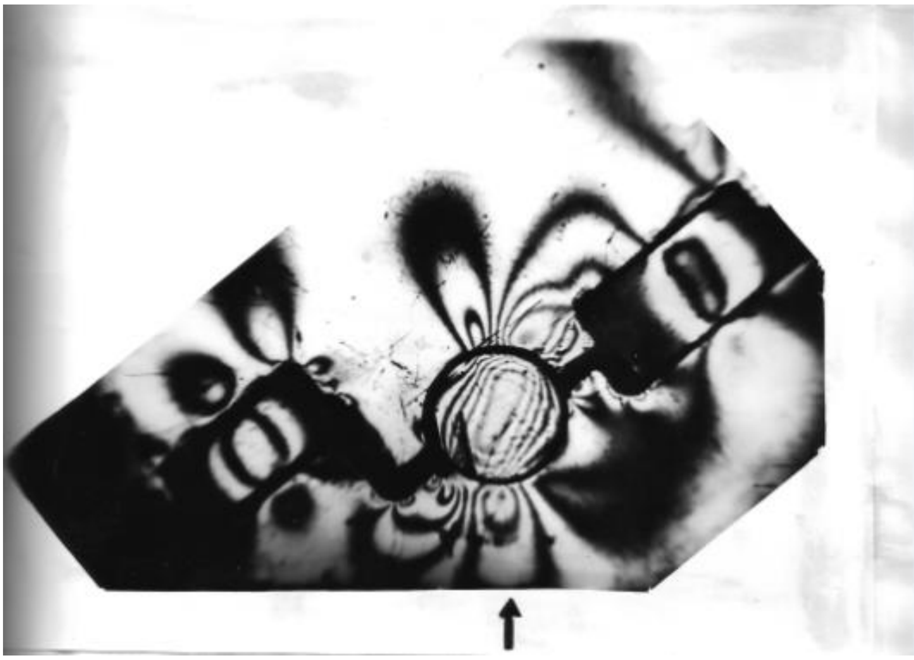

Fig 6: Pattern of strips of the model of rolling cutter unit

The results of a qualitative analysis of the load distribution between support bearings of the rolling cone are in complete agreement with the results of the analytical study and confirm the assumption of the highest loading of the ball bearing. The application of this technique can allow to assess the impact and to optimize the design of the support and cutting structure on the congestion and durability of the bearings at the design stage.

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