UDC 621.771.074

Contact Pressure Relief on Universal Drive Bearings of Rolling Mills

A. G. Savchenko, Yu. A. Mushenkov, E. A. Kolesnik

National Metallurgical Academy of Ukraine
4 Gagarin Ave., Dnipropetrovsk, 49600, Ukraine

Effect of stiffness of universal drive yoke of rolling mill on the contact pressures acting on the bearings is investigated. It is possible to reduce contact pressures by 22 % by means of changing yoke stiffness.

Keywords: SPINDLE, YOKE, ROLLING-CONTACT BEARING, UNIVERSAL DRIVE, STIFFNESS, CONTACT PRESSURES, ROLLING MILL

Introduction

At JSC "Dnipropetrovsk Iron& Steel Works named after Petrovskiy" rotation from pinion stands stands is passed to work rolls on mill 550 through the universal spindles on bronze slip bearings. Operation of these spindles is unsatisfactory because of angular plays [1]. Spindle assemblies with links on the roller bearings do not have such disadvantage. However, it is not always possible to replace the universal spindles with such assemblies as the external diameter of spindle hinge is limited by half of center-to-center distance of pinion rolls from the drive, and cardan joints are large-sized as compared to joints on slip bearings.

The external diameter of cardan joint is mainly defined by sizes of roller bearings in which several rollers in certain position carry the major load. Contact pressures upon rolling elements which define durability of the bearing considerably depend on bearing unit design.

The task of present research is reduction of maximum contact pressure by means of selection of yoke stiffness so that loading is distributed more uniformly on the surface of external bearing ring. First, this circumstance will enable to increase durability of the bearing, and secondly, to reduce rotary joint dimensions.

Earlier, these problems were solved experimentally [2, 3], for example, by polarization-optical method. Now we can work out contact problems in the appendix CosmosWorks of program SolidWorks [4] by finite element method and obtain rather exact results.

Methodology

To define contact pressures acting on the bearing we constructed a computational model of the yoke and bearing of universal shaft drive line in SolidWorks (**Figure 1**). For comprehensible results of calculation it is necessary to compact a lattice on the contact surfaces of bearing and yoke where required contact pressures occur.

The direction of resulting force $\overline{Q}$ of distributed load acting on the internal bearing ring from crossarm pin is defined by angle $\Theta_{1,2}$, i.e. angle between the line of resulting force action and axis OO_1 (**Figure 1**). For the yokes of input (output) and intermediate shafts of universal shaft drive line the angles Θ_1 and Θ_2 are defined respectively from equations (1) and (2) [5, 6]:

$$\Theta_1 = arctg(tg\alpha_{12} \sin\varphi_1); \qquad \text{(Eq. 1)}$$

$$\Theta_2 = \arcsin(\sin\alpha_{12} \cos\varphi_1), \qquad \text{(Eq. 2)}$$

where α_{12} - angle between gimbal axes; φ_1 - drive shaft rotation angle.

Hence, angles Θ_1 and Θ_2 can take values $-\alpha_{12} \le \Theta_{1,2} \le \alpha_{12}$ per one rotation of input shaft.

To reduce the computing process of contact pressures in CosmosWorks we used *operating load* function. This type of loading enables to simulate the action of crossarm pin on the internal bearing ring under cosine law at specified resulting force

$\overline{Q}$. According to data in [7] for breakdown stand 670 of mill 550 $Q = 4 \cdot 10^5 N$. The calculation is carried out in CosmosWorks after determination of material and kinematic boundary conditions.

Results and Discussion

Figure 2 illustrates the results of contact pressure calculation. At $\Theta_{1,2}$ = -15° maximum value is p_{max}=19.3 MPa, at $\Theta_{1,2}$ = 0 p_{max}=17.8 MPa and at $\Theta_{1,2} = 15^{\circ}$ p_{max}=16 MPa. Hence, the maximum value of contact pressure changes within the limits of 16 MPa ≤ p_{max}≤19.3 MPa per drive shaft rotation at $\alpha_{12} = 15^{\circ}$, i.e. the average value corresponds to angle $\Theta_{1,2}$ = 0. The yoke is cut out as shown in **Figure 3** to change stiffness. Sizes of this cutout are defined by parameters: radius R, width Δ, depth h and angle φ. To reveal the effect of these parameters on contact pressure value the following research was conducted: contact pressures are computed at change of one parameter (other parameters are invariable). Calculation with cutout is carried out at angle $\Theta_{1,2}$ = 0. The results helped reveal the effect of each geometrical parameter of the cutout on the value of maximum contact pressure (**Figure 4**).

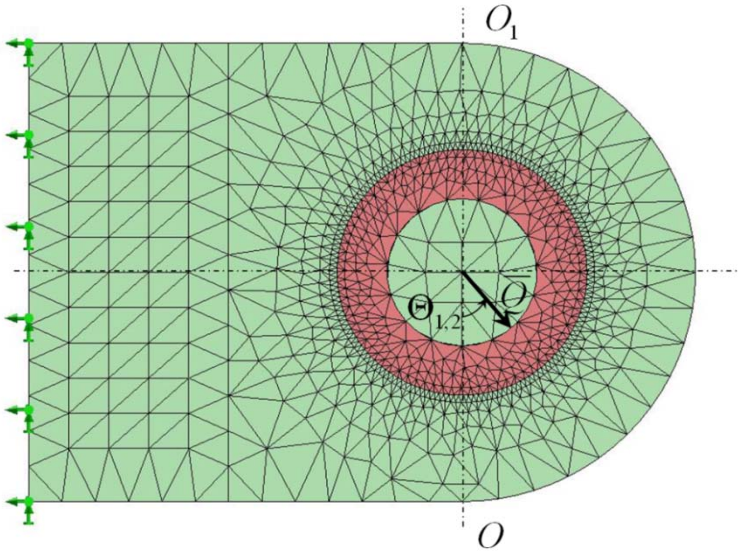

Figure 1. Yoke computational model

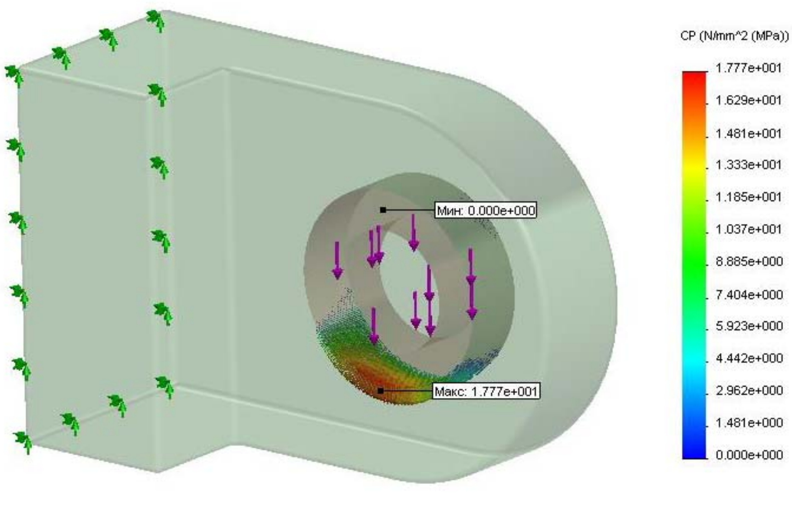

Figure 2. Distribution of contact pressures without cutout in the yoke

Pressures depend the most on radius R. If the cutout is too near the bearing, the maximum contact pressures are more than in the yoke without cutout (I zone). There is a certain zone II where the contact pressures are less.

At angle $150^\circ \le \varphi \le 180^\circ$ the loading on pivot pin bearing drops. Cutout width Δ in permissible limits has insignificant effect on the contact pressures. As the cutout becomes deeper, the contact pressures decrease.

These results (**Figure 4**) enabled to define geometrical parameters of the cutout at which the contact pressures are minimal.

$$R \approx R_{bearing} + 0.65 \cdot a;$$
$$\varphi \approx 180^\circ;$$
$$h \approx H_{bearing};$$
$$\Delta \approx 0.12 \cdot a. \qquad \text{(Eq. 3)}$$

where $R_{bearing}$ - bearing radius, a - width of yoke radial cross-section (**Figure 3**), $H_{bearing}$ - bearing width.

Further we calculated the contact pressures with geometrical parameters of the cutout defined by formulas (3). The maximum contact pressures between external bearing ring and bearing seat decreased to p_{max}= 13.8 MPa, i.e. loading on the most loaded bearing roller dropped by 22 %.

Distribution of contact pressures on the surface of external bearing ring without cutout (surface 1) and for yoke cut out (surface 2) is shown in **Figure 5**. ψ is a contact angle. According to these results of investigation the contact pressures are redistributed more uniformly on the surface of external bearing ring at yoke stiffness loss.

The cutout in the yoke is a stress concentrator, as a result, assurance factor by Mises criterion counted in CosmosWorks decreased from 4.2 to 3.8.

Nevertheless it is recommended to lower loading on the bearing and reduce assurance factor.

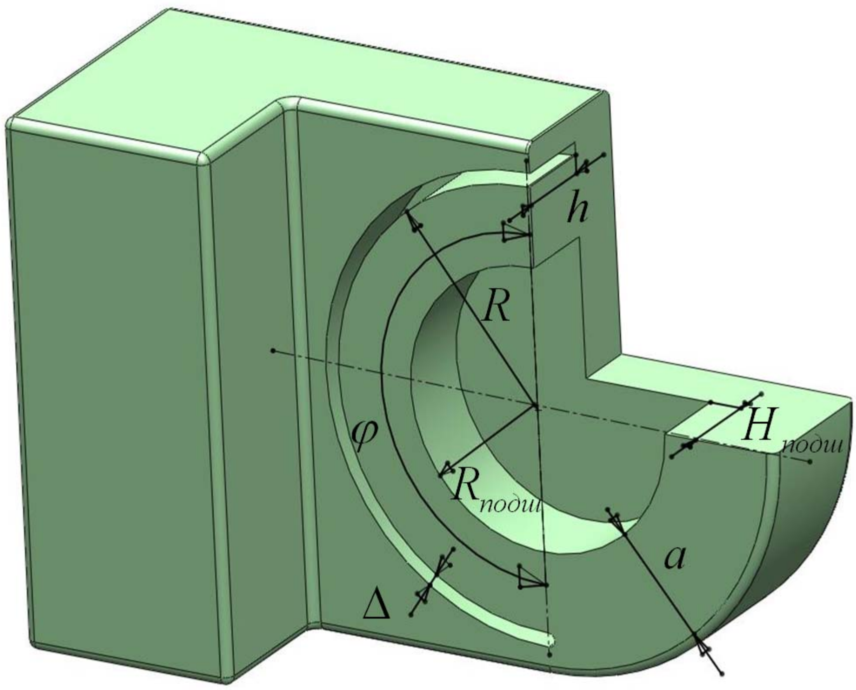

Figure 3. Yoke with cutout

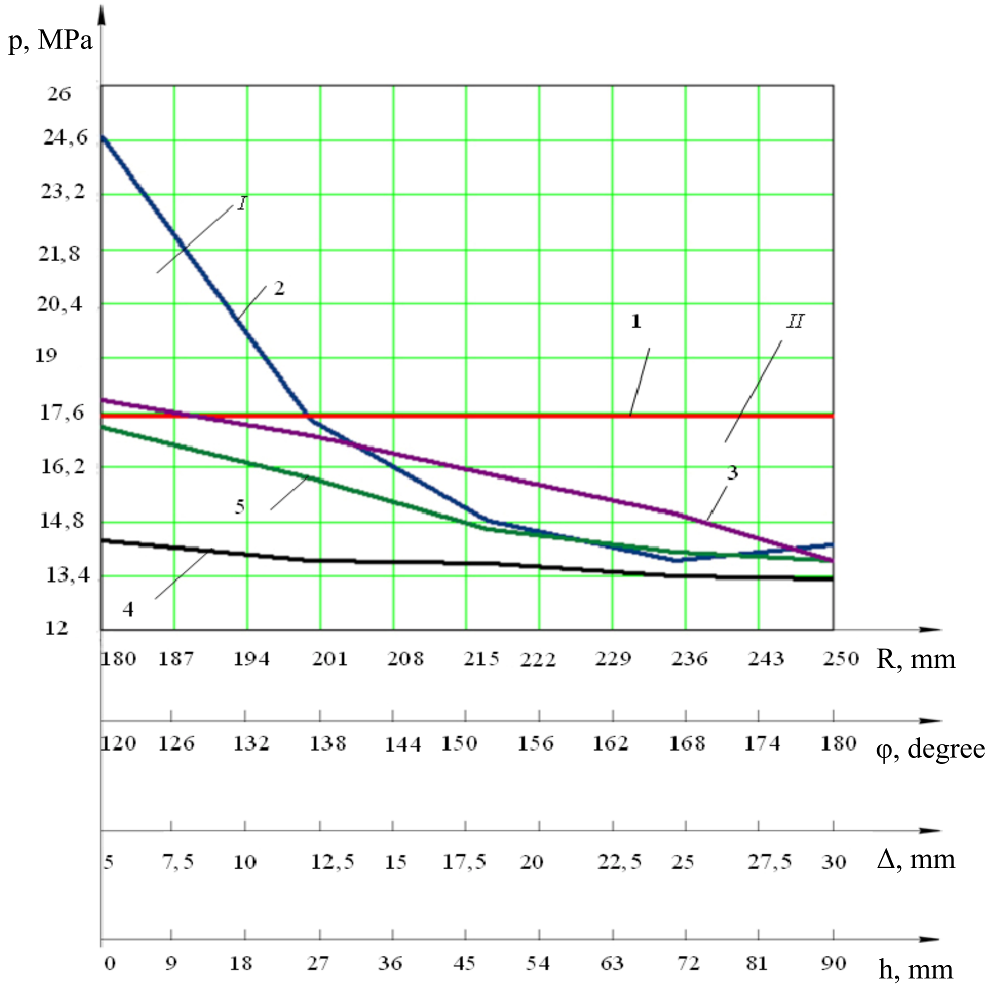

Figure 4. Effect of cutout geometrical parameters on the value of maximum contact pressures**:** *1* – no cutout; *2* – radius *R*; *3* – angle φ; *4* – width Δ; *5* – depth *h*

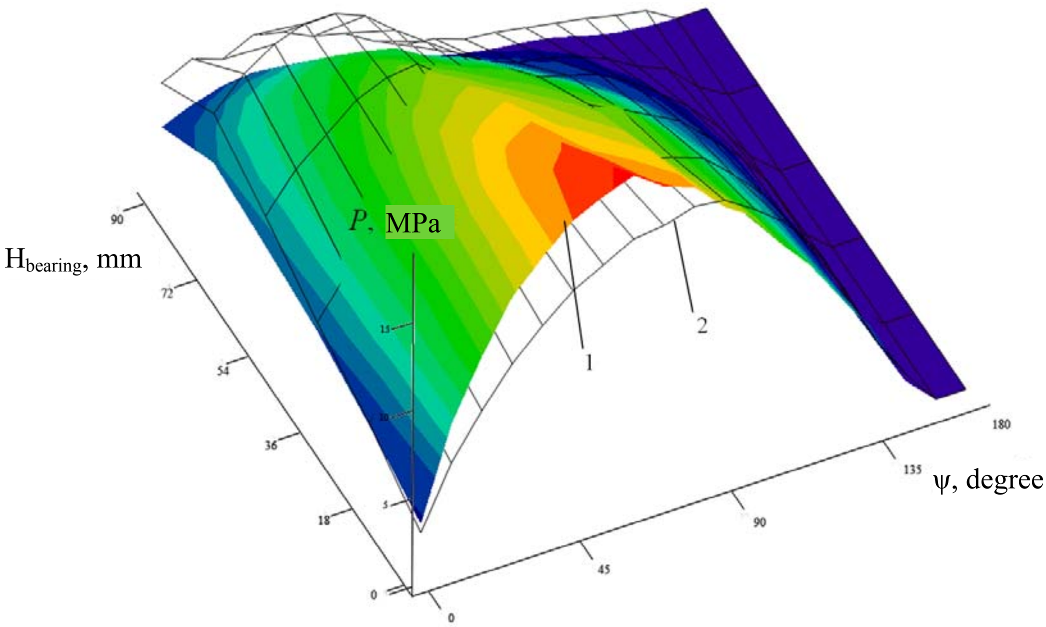

Figure 5. Distribution of contact pressures on the bearing surface: *1* – not cut out; *2* – cut out

Conclusions

Stiffness of universal shaft drive line yoke has a considerable effect on the contact pressures which act on the rolling-contact bearings. The possibility to reduce contact pressures on the bearing by 22 % is investigated in present work by means of yoke stiffness change. As a result, durability of rolling-contact bearings can be increased. The formulas for selection of rational geometrical parameters of the yoke are obtained.

References

1. Savchenko A. G., Mushenkov Yu. A. *Metallurgicheskaya i Gornorudnaya Promyshlennost*, 2008, No. 4, pp. 104-106.*
2. Sukharev I. P. *Prochnost` Sharnirnykh Uzlov Mashyn. Reference Book*, Moscow, Mashinostroenie, 1977, 258 p. *
3. Ivanchenko F. K., Polukhin P. I., Tylkin M. A., Polukhin V. P. *Dinamika i Prochnost` Prokatnogo Oborudovaniya*, Moscow, Metallurgiya, 1970, 486 p. *
4. Alyamovskiy A. A. *SolidWorks / COSMOSWorks 2006-2007. Inzhenernyy Analiz Metodom Konechnyh Elementov*, Moscow, DMK, 2007, 784 p. *
5. Kozhevnikov S. N., Perfilyev P. D. *Kardannye Peredachi*, Kiev, Tekhnika, 1978, 264 p. *
6. Savchenko A. G., Saliy E. N., Kuyan A. V. *Pidiyomno-Transportna Tekhnika*, 2004, No. 4, pp. 85-93.**
7. Savchenko A. G., Mushenkov Yu. A., Khanin M. I. *Metallurgicheskaya i Gornorudnaya Promyshlennost*, 2007, No. 6, pp. 103-105. *

* Published in Russian
** Published in Ukrainian

Received January 4, 2010

Уменьшение контактных давлений на подшипники карданной передачи прокатных станов

Савченко А.Г., Мушенков Ю.А., Колесник Э.А.

В работе исследуется влияние жесткости вилки карданной передачи прокатного стана на контактные давления, действующие на ее подшипники. Изменяя жесткость вилки, удалось уменьшить контактные давления на 22 %.