ANNALS of Faculty Engineering Hunedoara – International Journal of Engineering
Tome XVIII [2020] | Fascicule 4 [November]

ANNALS of Faculty Engineering Hunedoara – International Journal of Engineering
ISSN 1584 - 2665 (printed version); ISSN 2601 - 2332 (online); ISSN-L 1584 - 2665

1.Aleksandar ŽIVKOVIĆ, 2.Miloš KNEŽEV,
3.Milan ZELJKOVIĆ, 4.Milivoje MIJUŠKOVIĆ

STATIC ANALYSIS OF FOUR- POINT CONTACT BALL BEARINGS FOR AGRICULTURAL MECHANIZATION

1-3.University of Novi Sad, Faculty of technical sciences, Novi Sad, SERBIA
4.FKL d.o.o., Temerin, SERBIA

Abstract: A four-point contact ball bearing makes it easy to simplify machine designs that have a combination of radial, axial and moment loads because it can handle all three simultaneously. They are primarily used for slow to moderate-speed applications, or where oscillatory movement is predominant such as agricultural mechanization. Four-point contact ball bearing in this paper is single row angular contact ball bearing with raceways that is designed to support axial loads in both directions. The centre points of curvature raceways on the inner and outer ring are offset relative to each other in such a way that the balls are in contact with the bearing rings at four points under radial load. For a given axial load, a radial load is also considered. This paper presents numerical model (finite element model) for computing the behavior of statically loaded ball bearings with four-point contact, for agricultural mechanization. The numerical model has to predict an influence of axial and radial load on the stress level, and elastic deformation in the explanation condition. The results show that the contact loads of four-point contact ball bearing increase with the value of clearances, small changes of the clearance have a great influence on the contact pressure.
Keywords: four-point contact ball bearing, static analysis, finite element method

1. INTRODUCTION

In agriculture, like in all industries, there are growing demands on the machines and the comfort of their operation. The result is the growing demand on equipment parts and bearings as well. Increase of efficiency in agriculture changed the work habits. Decrease of the manual work, mechanization of various activities, increased performance of machines and equipment enabled processing of heavier objects, cultivation of larger areas. In today's agriculture industry is becoming more common developing the bearings of machinery based on to reduce the weight and dimensions as well as to improve overall performance. The typical bearing of seed drill disc application is four-point contact ball bearing with triple lip seals, a guarantee of precision sowing and long service life. Bearings are exposed to shock and high axial, radial and moment loads due to angle of pull.

Four-point contact ball bearings are radial single row angular contact ball bearings with raceways that are designed to support axial loads in both directions. This bearing outer ring consists of two symmetrical raceways. Each of them has a circular shape. The two raceways intersect each other in the middle of the ring in form of a singular point. The centre's of the two raceway circles have a small off-set in axial direction. The inner ring is split into two halves and has the same design principle as the outer ring. This provides an advantage compared to the deep groove ball bearing design. The design of the outer ring raceways allows defining the contact angle and the axial clearance independently. Afterwards, it is possible to design a specific axial clearance without changing the osculation. Generally, one ball theoretical has four different contact points to the rings, due to applied only radial load (Figure1a). Other hand, if load cases with pure axial load only two contact points in diagonal position transfer load (Figure 1b).

The two contact points move to the opposite position if the axial load changes its direction. This explains why this bearing type can accommodate axial loads in both directions. There are also applications with combined loads, but dominating axial load. In that case three contact points transfer load (Figure 1c) [4].

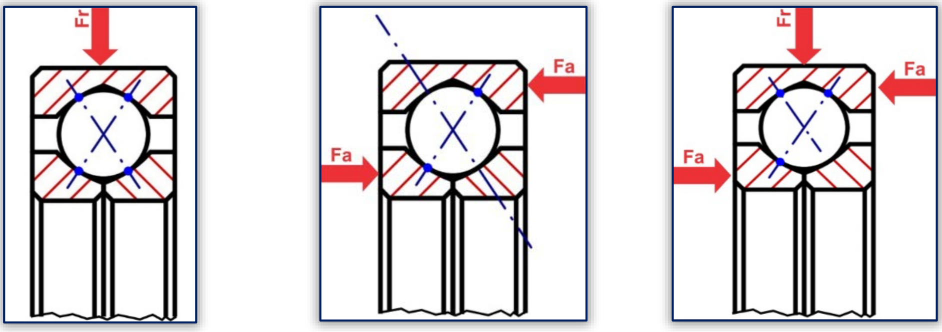


Figure 1. Load transmission in four-point contact ball bearing: a) radial load; b) axial load; c) combined loads

If the same functionality should be reached with single row angular contact ball bearings, two identical bearings in a back-to-back or face-to-face arrangement need to be matched. Two single-row angular contact ball bearings can be substituted by one four-point contact ball bearing. This means with a four-point contact ball bearing usually makes a second bearing unnecessary, which provides some very important benefits such as: space savings, weight savings, stiffness and accuracy of rotation, faster installation and etc.

This paper presents finite element and analytical model for computing the static behavior ball bearings with four-point contact under external axial and radial load. Using the finite element method, deformations between the ball and raceways as well as the stress were determined, while the dynamic capacity and bearing life were determined by the analytical model. The application of these models also analyzed the influence of the external load and clearance on the static characteristics of the four-contact point bearing for drill disc of seed.

2. MATHEMATICAL MODEL OF THE FOUR-POINT CONTACT BALL BEARING

The geometric parameters of the bearing, raceways deformation, contact loads, and stress are considered for each angular position of the ball. Based on the external axial and radial load of the bearing, deformation, stress and contact pressure of balls and raceways are established. Afterwards the contact loads for each ball and raceways as a result of external load and deformation are determined. In the end, based on the geometry of raceway and external load dynamic capacity is determined, while on the basis of contact loads, the bearing life is obtained. The process of analysis is shown in Figure 2.

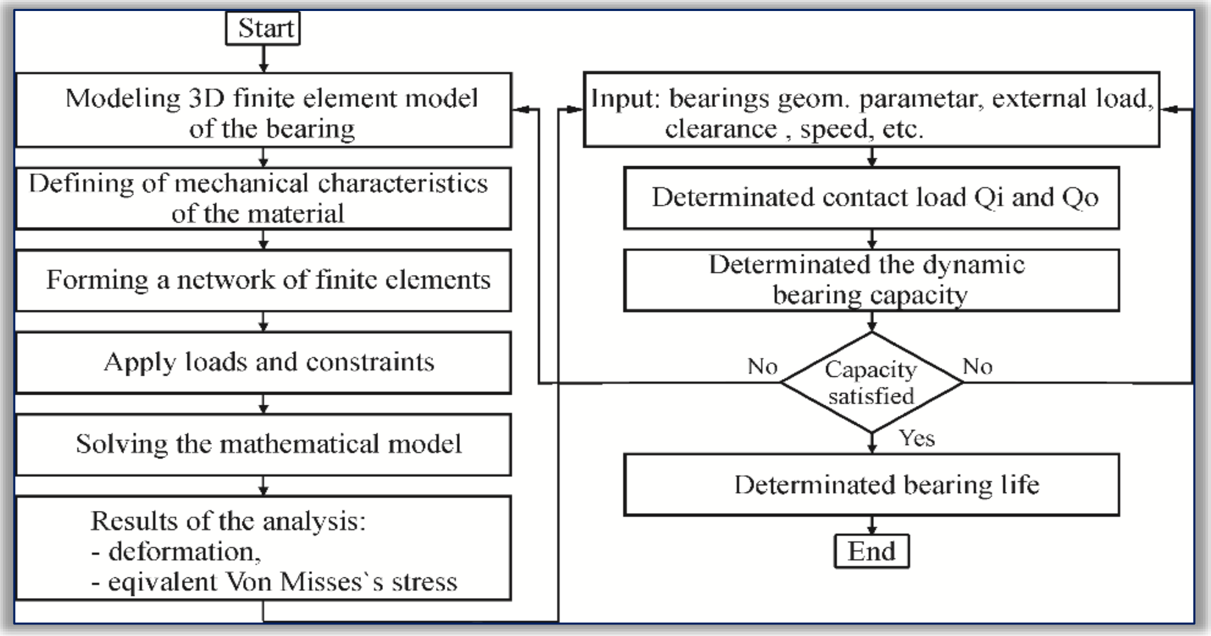


Figure 2. Analysis process of the bearing

— Determination of the dynamic and static bearing capacity

Dynamic bearing capacity significantly depends on the geometry of raceway and bearing contact angle, i.e. the preload and external load. Angle contact bearings are changed during operation due to the rotation of the ball and the load. For these reasons, for accurate determination of the bearing life, dynamic capacity is necessary to determine over relationship [1]:

$$C = f_c\left(i\cos\alpha\right)^{0,7} Z^{2/3} d_b^{1,8} \tag{1}$$

where are: i – number of rows, α - contact angle, Z – number of ball, d_b - diameter of the ball. The inner ring raceway rotates relative to the load and for ball bearing f_c is delaminated as [1]:

$$f_c = 39,9\left(1+\left[1,04\left(\frac{1-\gamma}{1+\gamma}\right)^{1,72}\left(\frac{f_i}{f_o}\cdot\frac{(2f_o-1)}{(2f_i-1)}\right)^{0,41}\right]^{3,33}\right)^{-0,3}\frac{\gamma^{0,3}(1-\gamma)^{1,39}}{(1+\gamma)^{1/3}}\left(\frac{2f_i}{2f-1}\right)^{0,41} \tag{2}$$

where: $\gamma = d_b\cos\alpha/d_m$; d_m is pitch diameter of the bearing; $f_{i,o} = r_{i,o}/d_b$; $r_{i,o}$ is radius of the inner and outer raceways.

The maximum static load capacity of the bearing has obtained on finite element model, when the most loaded ball reaches 4200 MPa according to EN ISO 76:2006.

— Determination of the bearing life

According to the classical theory of Lundberg and Palmgren, the nominal lifetime of roller bearing race L (in revolutions) can be determined at a 90% probability as follows [2]:

$$L_v = \left(\frac{C}{Q_e}\right)^3 \cdot 10^6 \tag{3}$$

where Q_e is the bearing equivalent contact load. The equivalent contact load Q_e are sum the contact forces and can be determined as [3]::

$$Q_e = \frac{1}{Z}\sum_{j=1}^{Z} K_n \delta^{3/2} \tag{4}$$

where K_n is the stiffness bearing along the lines of contact, obtained on the basis of Hertz's contact theory, and, δ is contact deformation between balls and raceways to be estimated using the finite element model.

— Finite element model of the bearing

In defining the finite element model of the four-contact point bearing for drill disc of seed, i.e., the rings, balls, and the housing, the hexahedral element was used to mesh solid structure. Model consists of 145709 solid elements and 84330 nodes in total. The finite contact elements were used to simulate contact joints. The three-dimensional model was considered, as shown in Figure 3. The defined loads and constraints are also shown in Figure 3

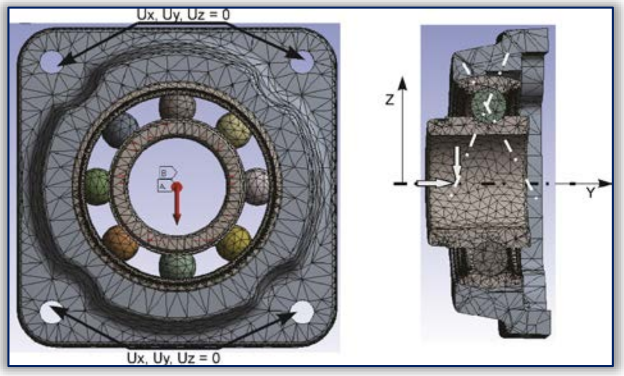
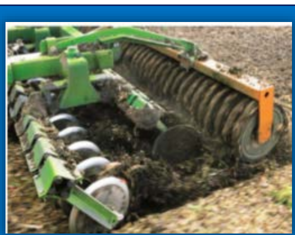
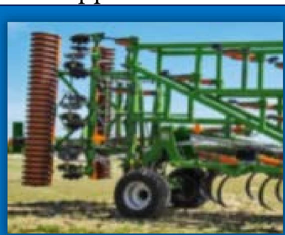
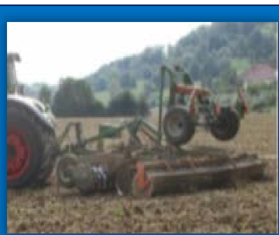


Figure 3. Finite element model of the bearing

In the analysis of the bearing behavior was considered under constant external axial (Fy) and radial loads (Fz) and when inner ring of the bearing was rotating at 280 [rpm], which corresponds to a speed of 20 km. The applied maximum loads on the inner ring are defined based on the exploitation conditions in the soil processing for three cases shown in Table 1.

Table 1. The cases of applied load

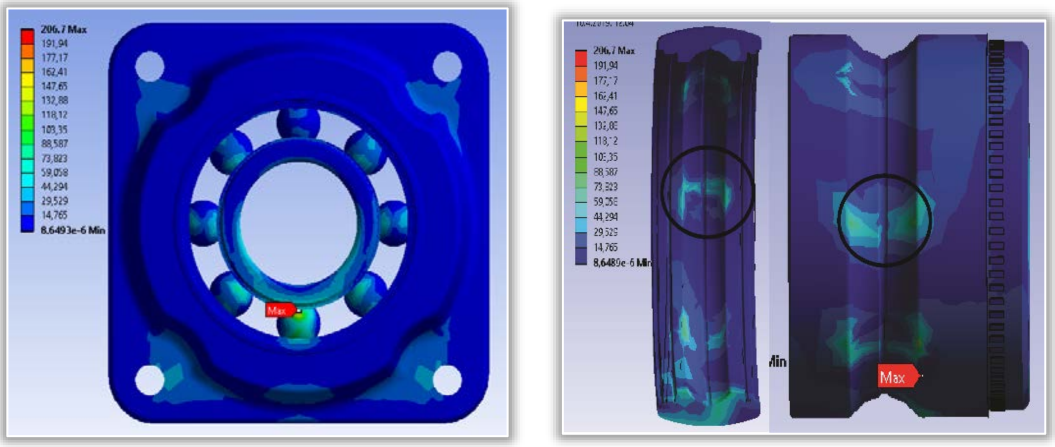
Cases of load / Load	 Working position	 Transport position	 Turning on the headland
F_{xmax}	5400 N	2000 N	3700 N
F_{ymax}	6200 N	9100 N	17600 N

3. RESULTS AND DISCUSSION

Analysis of the four contact point bearing for drill disc of seed static behavior, static and dynamic capacity and bearing life includes of the bearings FKL LSQFR 308 TBT. Also, in the analysis, the influence of the radial clearance (Gr = 15 to 35 μm) on the stress, deformation of the raceways and bearing life.

— Analysis of the equivalent Von Misses's stress and contact pressure

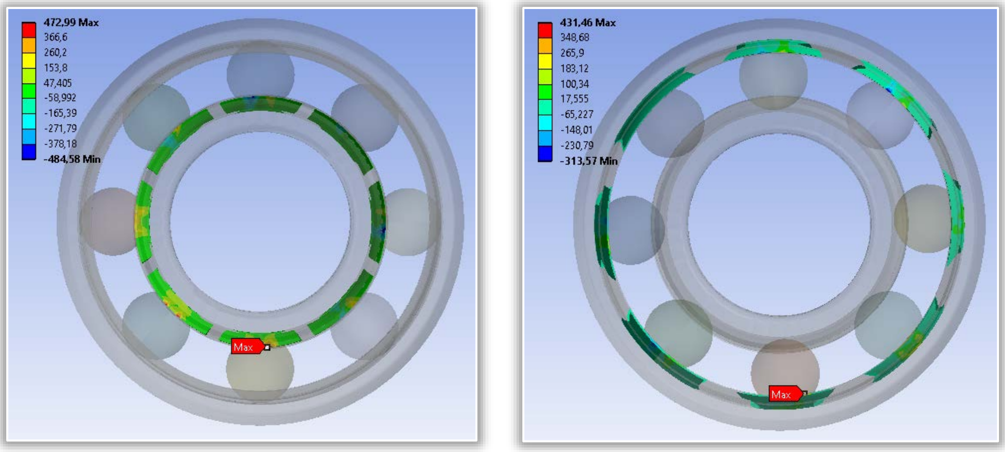
For the considered cases of load, the maximum stresses as well as the maximum contact pressure occur in the case of turning on the headland. Distributions of the maximum equivalent stresses for the turning on the headland are shown in Figures 4. The maximum equivalent stresses occur on the inner ring, of the bearing as shown in the figure 4 and amounts to 206 MPa. As shown in Figures 4b, the equivalent stresses at this bearing are approximately evenly distributed along the radius of the inner and outer raceways.



a) b)

Figure 4. Distribution of the maximum equivalent stresses on the: a) bearing assembly; b) outer/inner ring, for turning on the headland

Distributions of the maximum contact pressure of the raceways for the turning on the headland are shown in Figures 5. As shown in Figure 5, the maximum contact pressure between ball and outer raceway and ball and inner raceway appears at the position of the most loaded ball and for the inner raceway is 484 MPa, while at the outer raceway is approx. 431 MPa. Also, in Figure 5, it can be seen that the contact pressure between balls and raceways are in the range from 47 to 484 MPa for the inner raceway and from 17 to 431 MPa on the outer raceway, depending on the position of the balls.



a) b)

Figure 5. Distribution of the maximum contact pressure between balls and: a) inner raceway; b) outer raceway for turning on the headland

Other hand, contact pressure between ball/outer raceway and ball/inner raceway on the position of the most loaded ball is increased with increasing internal radial clearance as shown in Figure 6. By increasing the radial clearance, the maximum contact pressure passes from the inner to the outer raceway on the position of the most loaded ball.

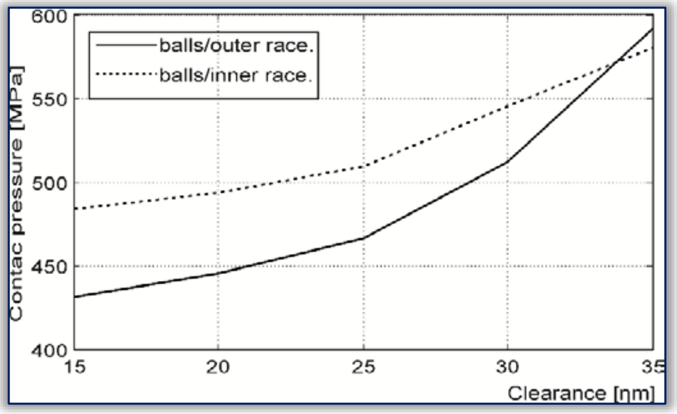
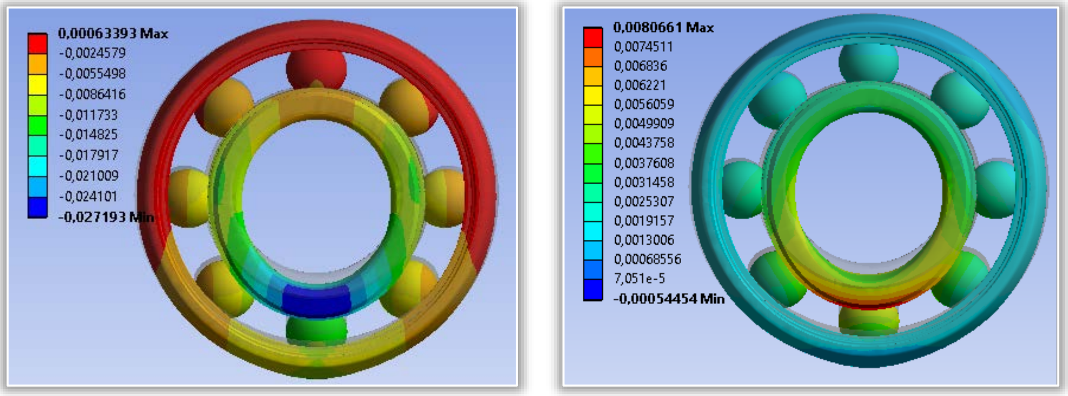


Figure 6. Change contact pressure between ball and raceways depending on the radial clearance

— Displacement analysis

In the displacement analysis, the displacements are considered in the direction of the force in Y direction (axis of rotation of the bearing) and with the Z direction (radial load). The maximum displacement in the Z direction of the considered bearing occurs on the inner ring is 9 μm for cases transport position, and 14 μm and 27 μm for cases working position and turning on the headland respectively. The maximum displacement in the Y direction occurs also on the inner ring is 2 to 8 μm, depending on the considered cases of load. Figure 7 shows the distribution of displacement in the Z and Y direction for turning on the headland.



a) b)

Figure 7 Distribution of the displacement in the: a) Z axes; b) Y axes

— Analysis of the dynamic and static bearing capacity and bearing life

Dynamic bearing capacity is obtained on the basis of relations (1) and (2) based on the internal bearing geometry, while the static bearing capacity is determined by finite element model, when the most loaded ball reaches 4200 MPa. Based on previous data of dynamic load capacity and contact deformation bearing life are determinate on the basis of relations (3) and (4). In Table 2 is shown the change of the bearing capacity and bearing life, depending on the radial clearance for considered bearing.

Table 2. Load capacity and bearing life depending on radial clearance

Radial clearance [μm]	Dynamic capacity [kN]	Static capacity [kN]	Bearing life [h]
15	59,2	48,7	14586
20		46,2	13959
25		41,5	13151
30		37,6	11875
35		35,1	10418

4. CONCLUSION

The paper presents a mathematical model for determinate main characteristic of the four-contact bearing for agricultural machinery. Due to their special technical features, four point contact bearings are highly suitable for bearing arrangements in agricultural vehicles and equipment. Radial clearance is geometrical characteristic with significant influence on important bearing characteristics, including bearing life. With increasing radial clearance the contact pressure between balls and raceways are increased and no uniform of load distribution is increased. Contact pressure on position of the most loaded ball is increased which causes an increase contact load between balls and raceways and it causes decreasing of bearing life.

Acknowledgements

In this paper some results of the project: "Contemporary approaches to the development of special solutions related to bearing supports in mechanical engineering and medical prosthetics" – TR 35025, carried out by the Faculty of Technical Sciences, University of Novi Sad, Serbia, are presented. The project is supported by Ministry of the education and science of the Republic of Serbia.

Note:

This paper is based on the paper presented at DEMI 2019 – The 14th International Conference on Accomplishments in Mechanical and Industrial Engineering, organized by Faculty of Mechanical Engineering, University of Banja Luka, BOSNIA & HERZEGOVINA, co–organized by Faculty of Mechanical Engineering, University of Niš, SERBIA, Faculty of Mechanical Engeneering Podgorica, University of Montenegro, MONTENEGRO and Faculty of Engineering Hunedoara, Unversity Politehnica Timisoara, ROMANIA, in Banja Luka, BOSNIA & HERZEGOVINA, 24–25 May 2019.

References

[1] Tedric, A. H., Michael, N. K. (2007). Rolling bearing analysis: Essential Concepts of Bearing Technology, Fifth edition, Taylor & Francis Group, ISBN: 0-8493-7183-X, 2007.
[2] Zverv, I., Pyoun, Y.S., Lee, B. K., Kim, J. D., Jo, I., Combs, A. (2005). An elastic deformation model of high-speed spindles built into ball bearings, Journal of Materials Processing Technology, vol. 170, p. 570-578.
[3] Zivkovic, A. (2013). Computer and experimental analysis of behavior ball bearings for special applications, Ph. D. thesis Faculty of technical science, University of Novi Sad, Novi Sad
[4] Evolution business and technology from SKF, from http://evolution.skf.com/four-point-contact-ball-bearings-two-in-one/, accessed on 202019-02-01.

ANNALS of Faculty Engineering Hunedoara – International Journal of Engineering
ISSN 1584 - 2665 (printed version); ISSN 2601 - 2332 (online); ISSN-L 1584 - 2665