

INFLUENCE OF WEAR ON DEEP GROOVE BALL BEARING SERVICE LIFE

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Abstract. *Frictional sliding which follows rolling of balls along rings raceways causes rolling bearing wear. Internal radial clearance of rolling bearing becomes larger due to wear of rolling elements and raceways. With increasing internal radial clearance, load distribution between rolling elements becomes more unequal. It is cause of decreasing rolling bearing life. Functional dependence of deep groove ball bearing life on wear is presented in this paper. Development of this function is based on classical expressions of bearing life theories and an original mathematical model of load distribution between rolling elements.*

1. INTRODUCTION

Operational quality and reliability of machines appreciably depend on the appropriate operational characteristics of their rolling bearings. The selection of types and sizes of rolling bearings for bearing arrangements is carried out on the basis of operational conditions: character and magnitude of external load, number of revolutions, temperature, and also expected service life. Thus, the correctly chosen bearing should keep the operational ability during predicted service life of the machine or period between regular repairs, when it can be replaced by the new bearing. The bearing life is calculated on the basis of classical *Lundberg and Palmgren* expression (1947) complemented by the adjustment factors, which take into account various levels of reliability, as well as characteristics of material and lubrication. The given expression is recommended by appropriate standard norms - ISO 281 [2].

The experience shows that the rolling bearings frequently do not achieve predicted service life. Namely, the bearing life value calculated with given expression is considered as a constant. Meanwhile, during operation of the bearing operational conditions can be changed (unpredicted increase of external load, temperature or speed). Static and dynamic load rating of bearing, as well as characteristics of lubricant and lubrication also can be

changed due to various reasons. Consequently, the bearing life is reduced in relation to calculated value. Very often bearing life is reduced by wear of contact surfaces in the bearing due to changes of bearing statement and operational conditions.

2. WEAR OF ROLLING BEARING

Under regular conditions (good sealing and lubrication with the optimum amount of clean lubricant) should not take place wear of bearing parts. In a case of inappropriate operational conditions, wear is regular phenomenon.

The most often cause of wear is the inappropriate lubrication - insufficient quantity or insufficient quality of lubricant (decrease of lubricant viscosity). In such cases, the formation of a lubricant film between contact surfaces with sufficient carrying capacity is impossible, and there is a direct metal-metal contact. Thus, wear is caused by the increased friction due to sliding, accompanying rolling of rolling elements along raceways. The process is more intensive in the cases of heavy loads and smaller speeds.

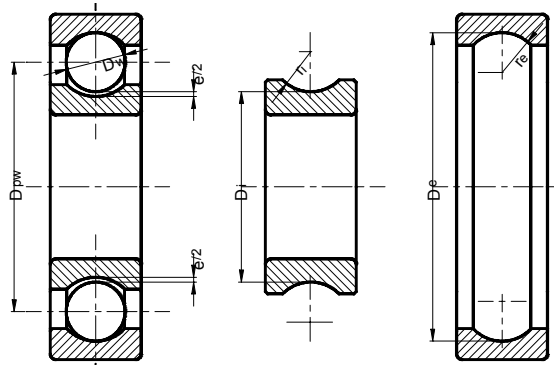


Fig. 1. Bearing internal geometry

Consequence of wear process during N revolutions of a rotating ring is linear wear rate h , which can be presented by expression [3]:

$$h = I\Delta l_N \quad (1)$$

where:

I – intensity of wear in rolling contact with sliding;

Δl_N – sliding distance during N revolutions of bearing.

Sliding distance Δl_N is a distance passed by a rolling element along rotating, as a rule inner ring. As for a revolution of inner ring of the radially loaded bearing the rolling element passes distance approximately equal to half of raceway perimeter, then can be written:

$$\Delta l_N = \frac{D_i}{2} N\pi \quad (2)$$

where: D_i – diameter of the inner ring raceway (Fig.1)

By substitution of Eq. (2) in Eq. (1), can be written:

$$h = I \frac{D_i}{2} N \pi. \quad (3)$$

On the basis of Eq. (3) can be made a conclusion, that the wear of the rotating ring raceway is linear function of its diameter, operating time (number of bearing revolutions) and intensity of wear.

The cause of bearing parts wear can be abrasive action of particles, which can get in the bearing during assembly, as well as during operation, due to bad sealing or polluted lubricant.

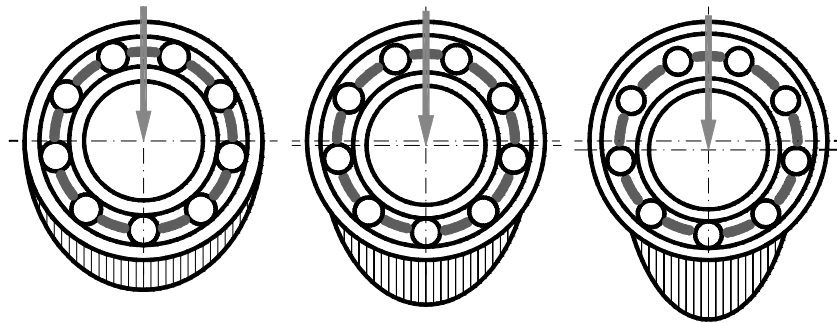


Fig. 2. Influence of wear on load distribution between rolling elements

Without regard to the cause of wear, during this process the bearing internal geometry and load distribution between rolling elements are changed (Fig. 2). Consequently, all operational characteristics of the bearing (static and dynamic load ratings, friction torque, vibration, noise, life) are also changed.

3. CHANGE OF BEARING INTERNAL GEOMETRY DUE TO WEAR

The main characteristics of the ball bearing internal geometry are the diameters of raceways and rolling elements (Fig. 1). Calculation of the internal radial clearance, important geometrical and functional characteristic of the bearing dependent on these diameters is based on expression:

$$e = D_i - D_e - 2D_w \quad (4)$$

In the case of pure radial load of the bearing with rotating inner ring and fixed outer ring, all bearing parts wear out. The wear trace on the inner ring has constant width, is located on the middle of a raceway and is stretched along whole perimeter. On the outer ring raceway the wear trace is present only along the loaded zone, has greatest width in a direction of external load, and is narrowed at edges (Fig. 3a).

For the further considerations the following is supposed:

- linear wear rate of inner ring is constant along whole raceway perimeter;
- linear wear rate of outer ring is constant along loaded zone length;
- linear wear of all balls is equal.

The maximal total linear wear rate of the ball bearing in a radial direction of external load is equal to the sum of partial wear rates of bearing parts (Fig. 3):

$$h = h_i + h_e + 2h_w \quad (5)$$

Wear rates of the bearing parts can be expressed through changes of the appropriate diameters:

$$h = \left| \Delta \left(\frac{D_i}{2} \right) \right| + \left| \Delta \left(\frac{D_e}{2} \right) \right| + |\Delta D_w| \quad (6)$$

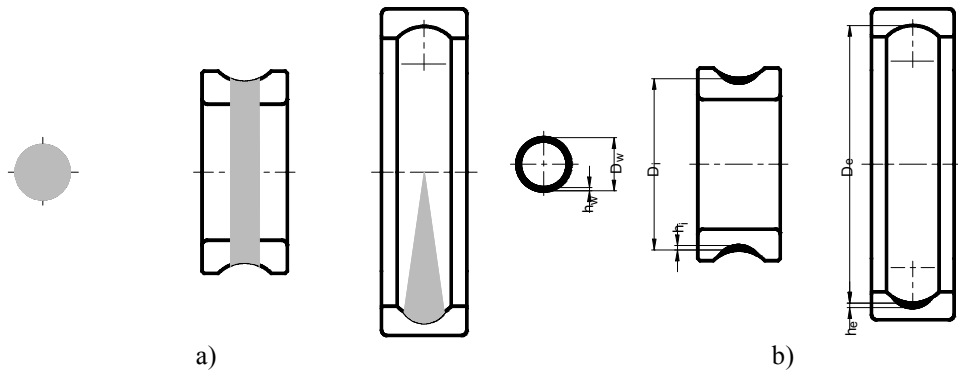


Fig. 3. Change of bearing internal geometry due to wear

On the basis of Eq. (4) and Eq. (6) the following equation can be written:

$$h = \Delta \left(\frac{e}{2} \right), \quad (7)$$

i.e. linear wear rate of ball bearing is a linear function of the internal radial clearance. Thus degree of bearing wear can be considered on the basis of change of internal radial clearance.

4. THE ANALYSIS OF INFLUENCE OF WEAR ON THE BEARING SERVICE LIFE

Ball bearing life, in million revolutions, can be determined on the basis of the following expression [1]:

$$L = \left(\frac{L_i^{\frac{10}{9}} + L_e^{\frac{10}{9}}}{L_o^{\frac{10}{9}}} \right)^{\frac{9}{10}} \quad (8)$$

where:

- L_i – life of rotating (usually inner) ring;
- L_o – life of fixed (usually outer) ring.

The ring life is function of the dynamic load rating and equivalent load of ring [1]:

$$L_{i(e)} = \left(\frac{F_{c,i(e)}}{F_{ekv,i(e)}} \right)^3 \tag{9}$$

The rings dynamic load rating is complex function of the bearing geometrical characteristics and it can be determined on the basis of the following expression:

$$F_{c,i(e)} = A \left(\frac{2f_{i(e)}}{2f_{i(e)} - 1} \right)^{0,41} \frac{(1 \mp \gamma)^{1,39}}{(1 \pm \gamma)^{1,3}} \gamma^{0,3} Z^{-\frac{1}{3}} \tag{10}$$

where:

A – constant [1];

$f_{i(e)} = \frac{r_{i(e)}}{D_w}$ – auxiliary value;

$\gamma = \frac{D_w}{D_{pw}}$ – auxiliary value;

$r_{i(e)}$ – groove radius of inner (outer) ring;

D_{pw} – pitch diameter of the ball set (Fig.1).

Geometrical characteristics of the bearing parts are changed during the wear. Consequently, the dynamic load rating of rings decreases. In this paper, this influence is not taken into account, i.e. rings dynamic load rating is considered constant during operation of the bearing. Because of that and because of small differences in geometry of raceways of both outer and inner ring the following relation can be assumed:

$$F_{c,i} \approx F_{c,e} = F_c = \text{const.} \tag{11}$$

On the basis of (8), (9) and (11) the following expression for bearing life can be written:

$$L = F_c^3 \left(F_{ekv,i}^{\frac{10}{3}} + F_{ekv,e}^{\frac{10}{3}} \right)^{-\frac{9}{10}} \tag{12}$$

External load of the rolling bearing is transferred from one ring to another one through rolling elements. In a case of radially loaded bearing not all rolling elements participate simultaneously in the transfer of external load. The rolling elements, simultaneously participating in transfer of external load are in the loaded zone. The loaded zone size is determined by number of simultaneously active rolling elements, dependent on internal radial clearance and total number of rolling elements. In a case of the geometrically ideal bearing, the loaded zone is located symmetrically in relation to direction of external load. Load distribution in the loaded zone is unequal [4]. *Lundberg* has established, that unequally distributed external load can be replaced by the average cubic load with sufficient accuracy [1]:

$$F_{ekv,i} = \left(\frac{1}{Z} \sum_{j=1}^Z F_j^3 \right)^{\frac{1}{3}}; \quad F_{ekv,e} = \left(\frac{1}{Z} \sum_{j=1}^Z F_j^{\frac{10}{3}} \right)^{\frac{3}{10}} \tag{13}$$

where:

F_j – load of a j -rolling element in the loaded zone ($j=1,2,\dots,Z$);
 Z – total number of rolling elements in the bearing.

In the radially loaded bearing only part of total number of rolling elements simultaneously participates in transfer of load (Fig. 4). Hence, Eq.(13) can be written in the form:

$$F_{ekv,i} = \left(\frac{1}{Z_s} \left(F_0^3 + 2 \sum_{i=1}^n F_i^3 \right) \right)^{\frac{1}{3}}; \quad F_{ekv,e} = \left(\frac{1}{Z_s} \left(F_0^{\frac{10}{3}} + 2 \sum_{i=1}^n F_i^{\frac{10}{3}} \right) \right)^{\frac{3}{10}} \quad (14)$$

where:

Z_s – number of rolling elements, which simultaneously participate in load transfer;
 $n = \frac{Z_s - 1}{2}$ – auxiliary value.

Load of the most loaded rolling element F_0 and other rolling elements in the loaded zone F_i can be expressed as a function of external radial load F_r [4]:

$$F_0 = K_{r0} F_r$$

$$F_i = \frac{K_{ri}}{\cos(i\psi)} F_r, \quad i = 1, \dots, n \quad (15)$$

where:

K_{r0}, K_{ri} – the load distribution factor;
 $\psi = \frac{2\pi}{Z}$ – angular distance between rolling elements (fig.4).

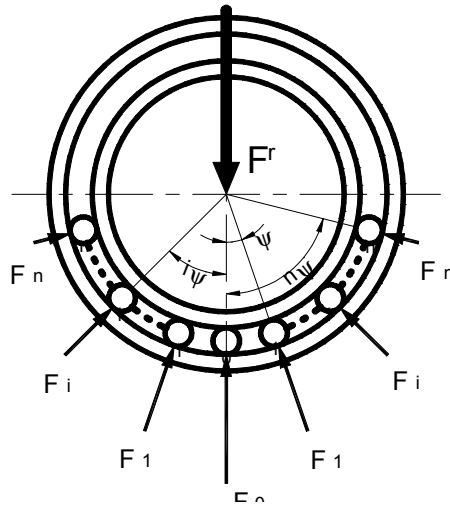


Fig. 4. Load distribution between rolling elements

On the basis of (12), (14) and (15) the expression for rolling bearing life as a function of the load distribution factor is obtained in the form:

$$L = \left(\frac{F_c}{F_r}\right)^3 \left(\left(\frac{1}{Z_s} \left(K_{r0}^3 + 2 \sum_{i=1}^n \left(\frac{K_{ri}}{\cos(i\psi)} \right)^3 \right) \right)^{\frac{10}{9}} + \frac{1}{Z_s} \left(K_{r0}^{\frac{10}{3}} + 2 \sum_{i=1}^n \left(\frac{K_{ri}}{\cos(i\psi)} \right)^{\frac{10}{3}} \right) \right)^{\frac{9}{10}} \quad (16)$$

The load distribution factor is the quantitative characteristic of load distribution, i.e. shows a degree of engagement of each rolling element in transfer of external load. The load distribution factor is function of total number of rolling elements and internal radial clearance, as well as function of bearing wear respecting Eq. (7).

By introduction of expression for the load distribution factor [4,5] in Eq. (16), taking into account the Eq. (7), the final expression for life of bearing with an initial zero clearance is obtained:

$$L = f_h \cdot \left(\frac{F_c}{F_r}\right)^3 \quad (17)$$

where:

$$f_h = \left(\frac{\left(\frac{1}{Z_s} \left(1 + 2 \sum_{i=1}^n \left(\cos(i\psi) + \frac{h}{\delta_0} (\cos(i\psi) - 1) \right)^2 \right) \right)^{\frac{10}{9}} + \frac{1}{Z_s} \left(1 + 2 \sum_{i=1}^n \left(\cos(i\psi) + \frac{h}{\delta_0} (\cos(i\psi) - 1) \right)^5 \right) \right)^{\frac{9}{10}}}{\left(1 + 2 \sum_{i=1}^n \left(\cos(i\psi) + \frac{h}{\delta_0} (\cos(i\psi) - 1) \right)^2 \cos(i\psi) \right)^{\frac{10}{3}}} \right)$$

The function f_h in expression (17) represents mathematical model of influence of bearing wear on the bearing life. In [5] this influence is considered in function of ratio h/w , where w is total relative radial displacement of rings under external load. As the displacement of rings depends on wear rate h (Fig. 5), the ratio h/w is characterized by some non-linearity, which is rather small and can be neglected. In this paper this non-linearity is avoided by introduction of relative wear in relation to total contact deformation of the most loaded rolling element – h/δ_0 .

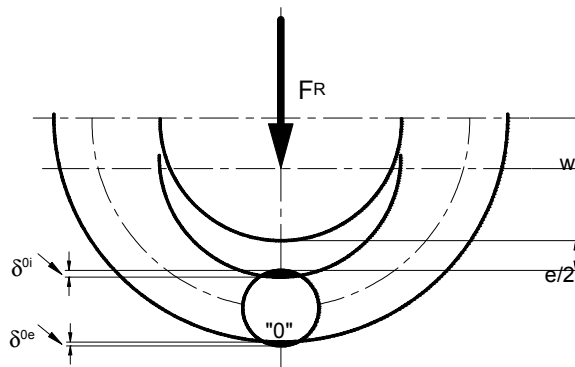


Fig. 5. Relative radial displacement of rings under external load

The total contact deformation δ_0 on a place of the most loaded "0"-rolling element can be expressed as follows (Fig.5):

$$\delta_0 = \delta_{0,i} + \delta_{0,e} \quad (18)$$

where:

$\delta_{0,i}$ – total deformation in contact of a "0"-rolling element and inner ring;

$\delta_{0,e}$ – total deformation in contact of a "0"-rolling element and outer ring.

Dependence of value f_h on number of rolling elements and bearing wear is shown in Fig. 6. With increasing total number of rolling elements value of function f_h increases. It is caused by increase of number of rolling elements simultaneously participating in transfer of external load and reduction of a degree of load distribution inequality. Irrespective of number of rolling elements, f_h decreases with increasing wear, and it causes reduction of bearing life.

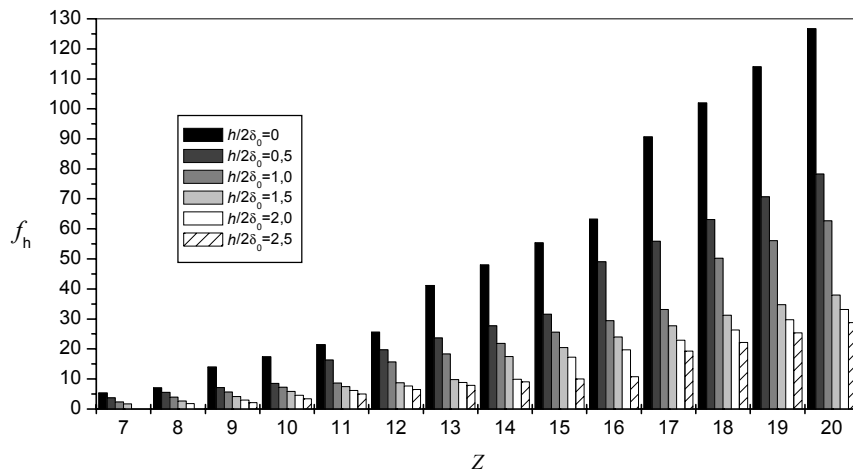


Fig. 6. Function f_h versus linear wear rate and total number of rolling elements

On the basis Eq. (17) can be made a conclusion, that the bearing life depends on wear, total number of rolling elements and number of simultaneously active in load transfer rolling elements (function f_h), as well as on dynamic load rating and external load.

The influence of wear on rolling bearing life for various number of rolling elements Z and number of simultaneously active rolling elements Z_s is shown in Fig. 7.

During bearing wear, the wear rate increases. Due to this bearing internal geometry is changed and consequently the internal radial clearance increases. With increasing internal clearance the loaded zone is narrowed and inequality of load distribution increases. It causes reduction of the remainder service life. In a case of intensive wear (large h), the reduction of bearing life in relation to calculated value could be even 80 %.

Sharp changes of gradient of life function for some wear rate values are noticed on diagrams in the Fig. 7. This phenomenon is caused by narrowing the loaded zone due to reduction of number of rolling elements, participating in load transfer in relation to theoretically expected number of active rolling elements ($Z_s' < Z_s$).

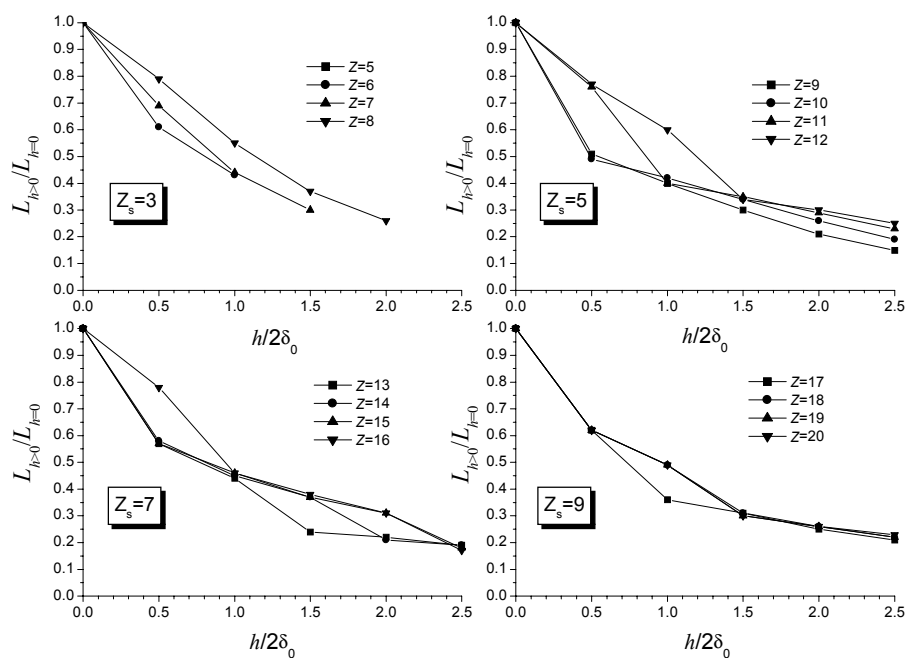


Fig. 7. Bearing service life as a function of linear wear rate

5. CONCLUSION

The bearing internal geometry is changed due to wear. Consequently, diameters of balls and inner raceway decrease, and diameter of outer ring raceway increases. With change of these dimensions the internal radial clearance of the bearing increases, owing to what the load distribution between rolling elements becomes more unequal. The mathematical relation between rolling bearing life and dynamic load rating, external load, linear wear rate, total number of rolling elements is established in this paper. The analysis of wear influence on bearing service life is enabled by introduction of new value – load distribution factor.

The results of the made analysis can be used for exacter prediction of the bearing life from aspect of internal geometry change, which is caused by wear.

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UTICAJ HABANJA NA RADNI VEK KUGLIČNOG KOTRLJAJNOG LEŽAJA SA RADIJALNIM DODIROM

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Habanje kotrljajnog ležaja je uslovljeno proklizavanjem koje prati kotrljanje kotrljajnih tela duž staza kotrljanja. Usled habanja spregnutih delova ležaja povećava se unutrašnji radijalni zazor. Sa povećanjem unutrašnjeg radijalnog zazora neravnomernost raspodele opterećenja na kotrljajna tela ležaja se povećava, što je uzrok smanjenja radnog veka ležaja. U ovom radu je izvršena analiza uticaja habanja na radni vek ležaja na osnovu klasične teorije radnog veka ležaja i originalnog matematičkog modela raspodele opterećenja na kotrljajna tela ležaja.