Cylindrical roller bearings with profiled contacting surfaces

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Abstract. An initial loading of cylindrical roller bearings in the elastic-plastic domain was performed to induce elastic shakedown phenomena able to change the basic profiles of both, rollers and raceways, which endorses a different value for the basic reference rating life. Fatigue life tests carried out on four lots of NJ206 cylindrical roller bearings revealed much higher values of $L_{10}$ and $L_m$ criteria for the bearings lots which experienced a suitable initial loading operation in the elastic-plastic domain. The reference rating lives, evaluated by using the lamina technique, confirmed the superiority of bearings lots which undergone an appropriate primary loading in the elastic-plastic domain.

1. Introduction
Reviewing papers by Hoeprich [1], de Mull et al. [2], and Reusner [3] pointed out that the contact pressures have a tendency to be significantly higher around the profile’s discontinuities than those attained in the middle of the contact area. The edge loading effect is considerably amplified for one end and diminished at the other when the bearing operates with misalignment between the inner and outer raceways. Consequently, the primary target was to reduce the stresses in the roller-raceway contact by modifying the roller profile. The diversity of crowning profiles includes: the single straight line with chamfer ends, single circular arc or a combination of multiple circular arcs, cylindrical-crowned (ZB), and the logarithmic profile.

Kapoor and Johnson [4], and Williams [5] found that after few cycles of primary rolling contact loading in the elastic-plastic domain, the material elastically shakedowns to a slightly modified roller axial profile and a stable state of compressive residual stresses.

These plastic deformations, achieved in primary loading cycles or in transient overloads, alter the initial contact geometry and diminish the edge effect.

2. Cylindrical roller bearings primary loaded in elastic-plastic domain

2.1. Rolling bearing and equipment for the primary loading operation
The NJ206 cylindrical roller bearing was selected for this study. Bearing’s internal data were as follows: inner raceway diameter $D=38.5$ mm, roller diameter: $D_v=7.5$ mm, total roller length: $L_v=7.5$ mm, roller’s end chamfer: $R_{ch}=(0.3 - 0.5)$ mm, number of cylindrical rollers: $Z_w=12$, $C2$ internal clearance: $S=(0 - 25)$ µm, P0 precision class. The bearings rings and cylindrical rollers were manufactured from 100Cr6
through hardened rolling bearing steel with the traction ultimate strength $R_m = 2200$ MPa, and the traction yield limit $R_{p0.2} = 1700$ MPa.

The target was to establish new cylindrical roller profiles after bearing’s running a number of 200 inner ring rotations at high radial loads. The equipment used for the primary loading of NJ206 bearing assured small rotation speeds of both bearing’s rings during the gradual increasing of radial load to the required level. During the primary loading operation the NJ206 bearing was symmetrical positioned to avoid shaft misalignment and unequal deformations.

2.2. Experimental procedure and results
The radial loads were chosen to create, in the contact achieved between the most loaded roller and inner ring raceway, the following values for the Hertzian pressure $p_{H0} \in \{3.5$ GPa, $4$ GPa, $4.5$ GPa, $5$ GPa$\}$. For each load level, the primary loading procedure was accomplished for two bearings. The results of measurements, performed along cylindrical roller generatrix, are encapsulated in figure 1.

![Figure 1. Plastic deformations caused by primary loadings and profile functions.](image)

2.3. Profile functions for cylindrical rollers subjected to elastic-plastic primary loading
Particular profile functions $P_k(x)$ were obtained for each level of the primary loading force (figure 1). The interpolations of the experimental data were accomplished using the model of logarithmic function revealed in standard ISO 16281 [6]:

$$P_k(x) = A_k \cdot D_w \cdot \ln \left[ \frac{1}{1-(2x/L_{we})^{0.8}} \right] \quad (1)$$

3. Fatigue life tests
3.1. Testing conditions
3.1.1. Testing machines. The fatigue life tests were performed on a battery of 12 machines. In each machine (figure 2) two specimen bearings were positioned at the ends of the machine’s shaft, so that the specimen bearings supported the same operating values regarding the load, misalignment angle, rotation speed, temperature of the lubrication oil, temperatures of inner rings, outer rings and rollers, respectively.
3.1.2. Cylindrical roller bearings. Four lots of 24 bearings each lot were durability tested. Two lots of bearings had cylindrical rollers with straight line profile (SLP), terminated with chamfers, and the other two lots, as result of a primary loading operation, had the cylindrical rollers and inner raceways with slightly modified profiles (PMP). The primary loading was performed a number of 200 inner ring rotations with a radial force able to generate, in the contact between the most loaded roller and inner raceway, a hertzian pressure $P_{10} = 4.5$ GPa. Before the primary loading, the roughness of the active surfaces were as follows: for inner and outer raceways $R_{a,r} = 0.14$ µm, and for cylindrical surface of rollers $R_{a,w} = 0.1$ µm. After the primary loading the roughness diminishes to $R_{a,r} = 0.1$ µm for the inner and outer raceways, and to $R_{a,w} = 0.08$ µm for cylindrical surface of rollers.

3.1.3. Operating conditions. Radial load on a bearing $F_r = 8500$ N, misalignment due to shaft’s elastic bending $\psi = 1.6$ minutes, rotation speed of the inner ring $n_i = 2880$ rev./min., oil bath lubrication ($e_c = 0.6$), lubrication oil- ISO VG 46 ($v_{40} = 46 \text{ mm}^2/s$, $v_{80} = 12 \text{ mm}^2/s$), operating temperatures: inner ring $t_i = 65$ °C, outer ring $t_e = 45$ °C, ambient $t_a = 20$ °C, shaft tolerance – k6, housing tolerance - H7.

3.1.4. Testing method. In order to shorten the testing time, the accelerated durability test method [7] was used. To maintain a high confidence level, the fatigue life test of each lot were performed until minimum 8 bearings of the lot failed by rolling contact fatigue.

3.2. Results provided by the fatigue life testing program
3.2.1. Types of failures. All bearings failed by rolling contact fatigue phenomena manifested practically complete as pitting on inner raceways (figure 3). The statistic of failures is revealed in table 1.
Table 1. Statistics of failures by rolling contact fatigue.

| Bearing lots | Number of tested bearings | Number of failed bearings | Bearing’s element failed | Inner raceway | Middle | Outer raceway | Roller |
|--------------|----------------------------|---------------------------|--------------------------|---------------|--------|---------------|--------|
| SLP 1        | 22                         | 11                        | 11                       | 0             | 0      | 0             | 0      |
| SLP 2        | 26                         | 14                        | 13                       | 1             | 0      | 0             | 0      |
| SLP 1+2      | 48                         | 25                        | 24                       | 1             | 0      | 0             | 0      |
| PDP 1        | 24                         | 9                         | 2                        | 7             | 0      | 0             | 0      |
| PDP 2        | 20                         | 8                         | 1                        | 0             | 1      | 0             | 0      |
| PDP 1+2      | 44                         | 17                        | 3                        | 13            | 0      | 0             | 1      |

3.2.2. The distribution function and parameters estimation. The Weibull distribution function has been hypothesized as describing the distribution of the fatigue lives. The estimation of parameters for each lot was accomplished by: (a) the graphical method in which the values P(i) of the distribution function are plotted on Weibull probability papers (WPP), [7, 8]; (b) the maximum likelihood method (ML), [8, 9]. The summary of fatigue lives is presented in table 2.

Table 2. Results of fatigue life tests.

| Bearing Groupb | Hardness (roller/ ring) (HRC) | Primary loading pressure (MPa) | Method of estimationb | Shape factor (L10 rating) | Lm median lives (hours) | Lives 90% confidence limits |
|----------------|-------------------------------|-------------------------------|-----------------------|---------------------------|-------------------------|----------------------------|
| SLP 1          | 61 / 61                        | WPP                           | ML                    | 1.96                      | 50.3                    | 50.3                      |
|                |                               |                               | WPP                   | 1.76                      | 58.8                    | 58.8                      |
| SLP 2          | 63 / 61                        | 4500                          | WPP                   | 1.73                      | 66.5                    | 66.5                      |
|                |                               |                               | ML                    | 2.47                      | 51.9                    | 51.9                      |
| PDP 1          | 61 / 61                        | 4500                          | WPP                   | 3.24                      | 202                     | 202                       |
|                |                               |                               | ML                    | 3.84                      | 193                     | 193                       |
| PDP 2          | 63 / 61                        | 4500                          | WPP                   | 1.49                      | 416                     | 416                       |
|                |                               |                               | ML                    | 0.92                      | 762                     | 762                       |

a SLP, straight line profiles; PDP, primary deformed profiles.
b WPP, Weibull probability paper; ML, maximum likelihood.

4. Rating Lives

4.1. Basic rating life
The basic rating life is provided by equation:

\[ L_{10} = \left( \frac{C_r}{P} \right)^{p} \]  \hspace{1cm} (2)

where \( P \) is the dynamic equivalent radial load and the exponent \( p \) has the value \( p=10/3 \) for line contact. For the case of bearings with unmodified line profiles the value of the basic dynamic load rating \( C_r \) was diminished by the correction factor \( \lambda =0.83 \), ISO 281 [10].
4.2. Modified rating life
To evaluate the modified rating life of bearings, the Standard ISO 281 [10] uses the equation:

\[ L_{10m} = a_1a_{ISO} \cdot L_{10} = a_1 \cdot a_{ISO} \cdot \left( \frac{C}{F_r} \right)^p \]  

(3)

where \( a_1 = 1 \) when a 90% reliability is admitted.

The life modification factor \( a_{ISO} \) can be derived from equation:

\[ a_{ISO} = f \left( \frac{e_C e_u}{p}, \kappa \right) \]  

(4)

where \( C_u \) is the fatigue load limit whereas the parameters \( e_C \) and \( \kappa \) take into account the contamination and lubrication conditions. Operating circumstances as: lubrication (the lubricant type, viscosity, bearing speed, bearing size, additives), environment (the contamination level, seals), contaminant particles (the hardness and particle size, lubrication method, filtration) and mounting, are considered by standard ISO 281[24] in selecting values for parameters \( e_C \) and \( \kappa \). For mineral oil lubrication and bearing raceway surfaces machined with good manufacturing quality, the condition of lubricant separation is described by the complex parameter \( \kappa \), defined as the ratio of the actual kinematic viscosity \( \nu \) to the reference kinematic viscosity \( \nu_1 \):

\[ \kappa = \frac{\nu}{\nu_1} \]  

(5)

For a more detailed estimation of the \( \kappa \) value, e.g. for especially machined surfaces, \( \lambda \)-ratio is first evaluated as the ratio between the minimum thickness of the EHD oil film, \( h_{EHD,\text{min}} \) (formed between the most loaded roller and inner raceway) and the composed roughness of the involved surfaces:

\[ \lambda = h_{EHD,\text{min}} \left/ \sqrt{1.25 \cdot \left( R_a^2_{\text{raceway}} + R_a^2_{\text{roller}} \right)^{\text{1/2}}} \right. \]  

(6)

Further, the complex parameter \( \kappa \) value is estimated with the equation:

\[ \kappa = \lambda^{1.3} \]  

(7)

4.3. Modified reference rating life
Additional to the influencing parameters described by ISO 281 [10] the standard ISO 16281 [6] takes into account three more operating circumstances: the misalignment, operating internal clearance and internal load distribution on rolling elements.

For bearings which sustained a primary loading, the elastic-shakedown provided permanent deformations of inner raceways roughly one third of values experienced by cylindrical rollers, while the permanent deformations of the outer raceways attained irrelevant values and were neglected.

The effect of residual stresses is not considered in the mentioned ISO standards.

4.3.1. The lamina technique. The lamina technique, as presented in standard ISO-16281 [6] was used to evaluate the modified reference rating life. In this procedure the elastic pressures distributions developed in contact areas of each loaded roller with both, the inner raceway (i) and outer raceway (e), have to be considered. For computations presented in the paper, a number of 256 uniformly spaced rectangular laminae were built.
4.3.2. Internal load distribution. The method used to determine the internal load distribution took into account operating radial clearance and rollers’ tilt, figure 4. The operating clearance considers the different operating temperatures of inner and outer raceways as well as the interference fits.

4.3.3. Pressures distributions. A semi-analytical method (SAM) was used to solve non-hertzian contacts achieved between the cylindrical roller and corresponding raceways, [11, 12]. Using an elastic-perfect plastic constitutive law, the SAM provides 3D pressures distributions achieved between each loaded roller with inner and outer raceways. For each lamina, the maximum value of the pressure was needed only. The 2D pressures distributions, developed by the internal loads $Q_2$, $Q_1$ and $Q_{12}$ on contact areas situated on inner raceway are exemplified in figure 5, figure 6 and figure 7, respectively, for bearings with straight line profiles and bearings with profiles modified by the primary loading procedure.

**Figure 4.** Internal load distribution of the external load.

**Figure 5.** Pressures distributions along rollers 2 and 10.
Figure 6. Pressures distributions along rollers 1 and 11.

Figure 7. Pressures distributions along the most loaded roller.

4.3.4 Rating lives comparison. The same operating conditions as used in fatigue life tests were considered as the input data in the computations of rating lives. The results are summarized in table 3.

For bearings with straight line profiles, as well as for bearings with modified profiles by primary loading procedure, the corresponding value of the modified reference rating life is well situated inside 90% confidence band of the $L_{10}$ life provided by the fatigue life tests.
Table 3. Results of rating lives computations.

| Bearing Group | Basic rating life \( L_{10} \) | Modified rating life \( L_{10m} \) | Modified reference rating life \( L_{10mr} \) | 90% life confidence limits |
|---------------|----------------------------------|----------------------------------|----------------------------------|--------------------------|
| SLP           | 57.2                             | 40.1                             | 19                               | 18 11\(\ldots\)30       |
|               |                                  |                                  |                                  | 20.3 11\(\ldots\)38       |
| PDP           | 107.1                            | 90.3                             | 135                              | 112 85\(\ldots\)168      |
|               |                                  |                                  |                                  | 102 50\(\ldots\)209       |

\( ^a \)SLP, straight line profiles; PDP, primary deformed profiles.

6. Conclusions
A few number of loading cycles in elastic-plastic domain causes an elastic shakedown phenomenon able to create favorable changes for profiles of rolling elements.

Fatigue life tests performed on four lots of NJ206 cylindrical roller bearings revealed the superiority of bearings lots that primary supported a few cycles of rolling loading in the elastic-plastic domain.

The modified reference rating lives, evaluated using the lamina technique, exposed a good agreement with the \( L_{10} \) lives accomplished by durability tests.

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