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Effect of ring misalignment on the fatigue life of the radial cylindrical roller bearing

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ABSTRACT

An effect of ring misalignment in the radial cylindrical roller bearing on its fatigue life is analysed. In order to predict the fatigue life, a methodology, which allows bearing geometrical parameters, including profiles of rolling generators, bearing radial clearance, angular tilting of rings and complex load to be accounted for, was applied. Stress distributions indispensable to calculate the predicted fatigue life were determined via solving the Boussinesq problem numerically for the elastic half-space and with the FEM. In the calculation of the predicted bearing life fatigue, Lundberg and Palmgren's model was employed. The results of calculations were compared to the recommendations of manufacturers of roller bearings referring to admissible tilt angles of rings.

1. Introduction

Catalogue methods for calculations of the fatigue life of roller bearings do not allow for consideration of numerous factors which affect considerably the bearing survival probability. One of them is misalignment of bearing ring axes resulting from manufacturing imperfections or too low housing rigidity or inclination of the shaft suspended in the bearing. In catalogues of roller bearings, only some indications on admissible ring tilting, which does not result in deterioration of the bearing performance and its fatigue life, are to be found. At the same time, catalogues do not provide any information about a degree the fatigue life will be reduced because of ring misalignment.

In radial cylindrical roller bearings, due to misalignment of ring axes, rollers are inclined with respect to the raceway, which results in nonuniform pressure distributions in contacts characterised by a pressure increase in one end of the contact line (Fig. 1). It is followed by a decrease in the bearing fatigue life.

In order to predict the fatigue life of rolling bearings, it is necessary to know the stresses occurring under the contact surface of mating elements. This stress distribution is determined mainly by pressures on the contact surface. Pressure distributions in cylindrical roller bearings differ significantly from the Hertzian distribution. One of the first researchers who dealt with an effect of misalignment on the fatigue life in cylindrical roller bearings was Harris. In [1] he presented an analytical method for investigations of an influence of misalignment of rolling elements with correction of generators. The attained results allowed for an evaluation of the percentage decrease of the bearing fatigue life versus the tilt angle of bearing ring axes.

So far, the most accurate information on the pressure distribution nature and the corresponding subsurface stress distributions can be obtained with the finite element method. Unfortunately, computations with the FEM require very high computing capacities, particularly in cases where there is a need to repeat calculations for a number of contacts present in a complex rolling pair. Seeking for load distributions on bearing rolling elements is even more time-consuming, in particular when contact phenomena on contact surfaces are to be taken into account. For these reasons, the authors of publications on the determination of load and stress distributions in roller bearings often used flat (2D) FEM models [2–4].

When the problem under analysis required a 3D model of the roller bearing to be built, then numerous researchers replaced rolling elements with elements defined by the user, i.e., the so-called superelements, for the sake of simplification. Such superelements were truss elements of nonlinear characteristics, often including an additional beam element, used by Golbach [5]. Claesson [6] applied similar elements. In his study, he emphasised that an application of complete computational models of bearings

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Nomenclature

A	material constant
B	bearing width
C	basic dynamic load rating
D	bearing outside diameter
D_r	roller diameter
E	Young's modulus
F_r	radial load of the bearing
F_a	axial load of the bearing
L	number of revolutions, fatigue life
L_r	roller length
N	number of load cycles
Q	resultant normal force in the roller-main race contact
Q_f	resultant normal force in the roller-flange contact
Z	depth of occurrence of maximal von Mises stresses along the x axis
Z_r	number of rollers in the bearing
c	exponent in the equation determining the survival probability
d	bearing bore diameter
d_{bi}	diameter of the inner ring raceway
e	Weibull slope
g	radial clearance in the bearing
h	exponent in the equation determining the survival probability
l	length of the roller-main race contact area
p	surface pressure, exponent in the survival probability equation
p_p	shakedown limit for the elastic-perfectly plastic material
q	force per unit length
r_b	radius of the main race
r_c	roller chamfer
u	number of load cycles per one revolution
ΔV	volume of the material subject to stresses
η	roller tilt angle
ξ	tilt angle of the bearing ring axes
$\Sigma\sigma$	contact curvature
σ	maximal von Mises stress occurring along the x axis
σ_o	tensile yield strength
φ	survival probability of the bearing ring element
ψ	angle measured along the bearing circumference

in the FEM is impossible from the viewpoint of costs involved in conducting such calculation procedures. He pointed out to a necessity to employ simplified models of bearings. He developed three simplified FEM models for cylindrical and taper roller bearings. In all these models, the bearing rigidity was modelled, partially or globally, with nonlinear spring elements. The models enabled the misalignment of bearing rings to be considered in computations. The models of cylindrical roller bearings were verified by a comparison with the complete model. The author did not conduct, however, calculations of the fatigue life of the bearings under investigation.

If the main aim of the analysis was to investigate the phenomena occurring in a single rolling element-race contact, in general researchers were satisfied with building a 3D model of a part of the bearing, which included a rolling element and a section

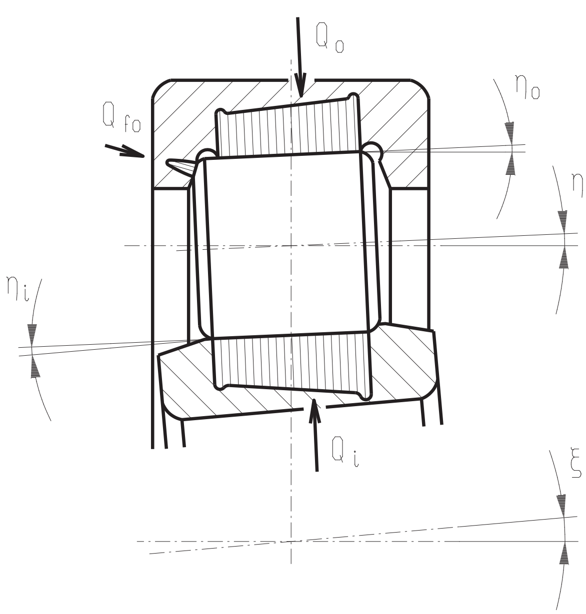

Fig. 1. Roller tilt angle η_i, η_o with respect to the bearing raceway at the tilt angle of the bearing ring axes ξ.

of one bearing ring. This approach was assumed by Xia et al. [7]. The paper was devoted to an analysis of the effects of misalignment of rings of the radial cylindrical roller bearing and the tilt angle of rollers with respect to the race, associated with this misalignment, on the pressure distribution in contact, and, consequently, on the bearing fatigue life. The authors used a 3D model of the bearing section, consisting of a half of the roller and the corresponding parts of both rings due to the symmetry. Benchea et al. [8] followed a similar approach. They presented the investigations on effects of misalignment of cylindrical roller bearings, which results in accumulation of pressures and an increase in von Mises stresses, and thus in a decrease in the bearing load rating. Rollers with chord–arch (ZB) correction were subject to analysis. Ramu and Murthy underlined an important role played by correction of roller generators in radial cylindrical roller bearings in their study [9]. They carried out an analysis of stresses on the basis of 2D models of the roller and the raceway. They analysed rollers with rectilinear, rounded and logarithmic profiles to mate races of rectilinear generators. The authors paid attention to the importance of correction as a way to avoid a negative effect of misalignment of rings on the bearing fatigue life, however they did not disseminate the results of stress analysis of the rollers tilted with respect to the race. A 3D model of a part of the bearing was also used by Göncz et al. [10]. They dealt with assessment of the static bearing load rating with a hardened raceway. The 3D numerical model assumed by the authors accounted for the roller geometry (various kinds of generator correction, etc.), mechanical properties of the material and the roller tilt angle with respect to the race. Calculations of contact stresses were carried out with the commercial ABAQUS/CAE software package. A contact of a single roller with a segment of the slewing bearing raceway was subject to analysis.

To determine the load distribution on bearing rolling elements, researchers often applied numerical methods other that the FEM, using the FEM to determine stresses in single contacts only, see, e.g., Zhenhuan et al. [11], who dealt with analysis of loads and contact stresses in roller bearings designed to operate under high speeds, including misalignment effects of bearing rings. The quasi-dynamic method was used by them to determine a load distribution on rolling parts, whereas pressure distributions in rollers