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Fatigue life prediction of the radial roller bearing with the correction of roller generators

Bogdan Warda [a,*], Agnieszka Chudzik [b,1]

[a] Lodz University of Technology, Department of Vehicles and Fundamentals of Machine Design, Zeromskiego 116, 90-924 Lodz, Poland
[b] Lodz University of Technology, Department of Dynamics, Stefanowskiego 1/15, 90-924 Lodz, Poland

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ABSTRACT

The paper presents the methodology of fatigue life prediction of radial cylindrical roller bearings, which allows to take into account in the fatigue life calculation geometric parameters of the bearing, including radial clearance and the profiles of rollers. In addition, the methodology takes into account the effect of combined load and misalignment of the bearing rings on the fatigue life. The stress distributions which are necessary to calculate the predicted fatigue life were determined by solving numerically the Boussinesq problem for elastic half-space. The Lundberg and Palmgren model was used for the calculation of the predicted fatigue life of the bearing. The paper focuses on determining the effect of roller profiles on the bearing fatigue life. Pressure distributions obtained by the described methodology were compared to the distributions determined according to the finite element method. The calculated fatigue life of cylindrical roller bearing was compared with the experimental results.

1. Introduction

To predict the fatigue life of rolling bearings, it is necessary to know the stresses occurring under the contact surface of mating elements. The stress distribution is determined mainly by the pressure on the contact surface. Pressure distributions in roller bearings, both cylindrical, conical and spherical, differ significantly from the Hertzian contact pressure distribution. So far, the most accurate information about the pressure distribution can be obtained using a finite element method. This method allows not only to take into account in the calculation any shape of generators of solids in contact, but also their finite length. Unfortunately, the calculation of pressure distributions and corresponding subsurface stress distribution using FEM require very high computing power. This is particularly important in cases where there is a need to examine not only the phenomena occurring in a single contact, but if it is necessary to repeat calculations for a number of contacts present in a complex rolling couple.

Usually full, solid (3D) FEM models were applied to ball bearings. An example might be the work by Kang et al. [1], dedicated to the analysis of deformations in the deep-groove ball bearing. To determine the strain authors used A modification of the Jones–Harris method the modified Jones–Harris method, for which developed a more economical, simplified partial model of the bearing. Full 3D model, which requires a longer computation time, the authors used only to verify the obtained results. In a similar way acted Łazarz et al. [2], using for the stiffness determination of ball bearing 3D FEM model of only a part of the bearing.

A large number of rolling elements cause that the building of a full bearing model and the modelling of the contact problem of each rolling element-bearing race contact is practically impossible. For these reasons, the authors of publications often used the flat (2D) FEM models in order to determine load and stress distributions in roller bearings. Among others, the paper by Zhao [3] is such a publication. In it the author presented load distributions on the rollers of the radial cylindrical roller bearing with solid and hollow rollers, which were calculated using FEM. The author also examined the effect of radial clearance on load distribution. Similarly, Demirhan and Kanber [4] used a 2D model in studies on distribution of loads and stresses in the roller bearing. Another example of the use of flat roller bearing model are the works by Laniado-Jácome et al. [5,6], dedicated to the development of the numerical model for mechanical events simulations occurring in rolling bearings, including a study of sliding between rollers and races.

Effective application of the finite element method for the analysis of loads and stresses in the rolling bearing requires an appropriate modeling of the rolling elements. In order to simplify the three-dimensional model of a rolling bearing, many researchers replaced

* Corresponding author. Tel.: +48 426312251.
E-mail addresses: bogdan.warda@p.lodz.pl (B. Warda), agnieszka.chudzik@p.lodz.pl (A. Chudzik).
[1] Tel.: +48 426312420.

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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
E'	equivalent 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
M	plastic modulus
N	number of load cycles
Q	resultant normal force in the roller-main race contact
Q_f	resultant normal force in the roller end-flange contact
Z, z_o	depth at which the maximum failure-causing stress occurs
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 contact area of the roller and the main race
p	surface pressure, exponent in the fatigue life equation
p_k	shakedown limit for the elastic linear kinematic-hardening plastic (ELKP) material
p_p	shakedown limit for the elastic-perfectly plastic material
q	force per unit length
r_b	radius of main race
r_c	roller chamfer
u	number of load cycles
ΔV	volume of the material subjected to the stress
δ_f	angle of inclination of the flange
η	roller tilt angle
θ	roller skew angle
θ_f	depth of penetration of the roller and flange
ν	Poisson's ratio
ξ	tilt angle of the bearing rings axes
σ	maximum von Mises stress
σ_k	kinematic yield strength
σ_o	tensile yield strength
τ_o	maximum orthogonal shear stress
φ	survival probability
χ	angle between the roller end and the flange
ψ_{lim}	angle specifies the size of the load-bearing rollers zone for $F_a > 0$
ψ_ε	angle specifies the size of the load-bearing rollers zone for $F_a = 0$
ω	deformation of an elastic half-space exposed to the pressure

the rolling elements, rollers or balls by the elements defined by the user, the so-called superelements. Superelements are the truss elements of nonlinear load-deformation characteristics, often with an additional beam element as in the paper by Golbach [7], or in the paper by Kania [8], on the slewing bearings. Rolling element models described in both articles allowed to determine the effect of correction of the rolling element generators constituting on the load and stress distributions in the bearings [7,8] as well as the bearing fatigue life [7]. Kabus and Pedersen [9] applied a similar model of the roller to simulate the load distribution in roller bearing, based on the use of elastic elements of non-linear characteristics.

If the main aim of the analysis was to investigate the phenomena occurring in a single rolling element-race contact, researchers generally were satisfied with building of a 3D model of a part of the bearing, which included the rolling element and a section of one of the bearing rings (often replaced with a solid with a flat surface) or, in the case of symmetrical components, only a half or a quarter of the bearing element. This approach was assumed by Poplawski et al. [10]. The authors studied the effect of the profile of the roller on the fatigue life of cylindrical roller bearing. They analyzed several types of roller correction, including the case of roller with rectilinear generators, and determined using FEM the distributions of maximum orthogonal shear stress and the maximum von Mises stress. Fatigue life calculations were made not for the complete bearing but only for the bearing raceway, which had a flat surface in the applied 3D model. Similar problems have been discussed in papers written by Xia et al. The first of these concerned the selection of the optimal range of interval of arc modification length of cylindrical roller [11]. The second work was dedicated to the analysis of the effects of misalignment of rings of the radial cylindrical roller bearing and tilt of the rollers associated with the misalignment on the pressure distribution in contact, and consequently on the bearing fatigue life [12]. In both works the authors used a partial 3D model of the bearing, due to the symmetry consisting of half of the roller and the corresponding parts of both rings.

The results of the analyses presented in the papers [10,11,12], however, did not in a direct way allow to determine the effect of rollers profile, misalignment of rings or, as the work [3], bearing clearance on the fatigue life of the bearing. The aim of the present study was to develop a methodology that allows fast prediction of fatigue life of radial cylindrical roller bearings, much faster than using the finite element method, allowing for the simultaneous including in the calculation of all the factors described above, as well as the effect on the fatigue life of a combined load, which can carry cylindrical roller bearings of NJ type. This goal has been achieved by solving the equilibrium equations of rings and rollers numerically, and found the pressure and subsurface stress distributions by solving the Boussinesq problem for elastic half-space using the algorithm described in the papers [13,14]. The modified Lundberg and Palmgren model [15,16] was used for the calculation of the predicted fatigue life of the bearing. The paper focuses on determining the effect of roller profiles on the bearing fatigue life. Pressure distributions obtained by the described methodology were compared to the distributions determined according to the finite element method, the method that ensures so far the most accurate determination of the stresses in complex mechanical systems. The calculated fatigue life of cylindrical roller bearing was compared with the experimental results described in the dissertation submitted by Waligóra [17].

2. Fatigue life of cylindrical roller bearing

To determine the fatigue life of a cylindrical roller bearing the methodology based on the basic assumptions of the fatigue life