Tribology International 115 (2017) 59–69

Contents lists available at ScienceDirect

Tribology International

journal homepage: www.elsevier.com/locate/triboint

Friction torque in four-point contact slewing bearings: Applicability and limitations of current analytical formulations

Iker Heras [a,*], Josu Aguirrebeitia [a], Mikel Abasolo [a], Jon Plaza [b]

[a] Department of Mechanical Engineering, ETSI-BILBAO, University of the Basque Country, Alameda de Urquijo, s/n, 48013 Bilbao, Spain
[b] Simulation Tools Development Department, Cikatek S.Coop., 48710, Berriatua, Spain

ARTICLE INFO

Keywords:
Slewing bearing
Four-point contact
Friction torque
Ball-raceway contact

ABSTRACT

The friction torque is a key parameter in the design of slewing bearings. The most challenging issue in its calculation resides in the modelling of rolling element contact mechanics. In this manuscript two models for the friction torque calculation are compared: a Finite Element model and the analytical model proposed by Leblanc and Nelias. The last one assumes that full sliding occurs in the ball-raceway contact, which is demonstrated to be an appropriate simplification for many cases, but not for applications where large tilting moments are involved. The objective of this work is to determine the field of application of the analytical model, highlighting its limitations and therefore justifying further research to implement more complex formulations for the contact simulation.

1. Introduction

Slewing bearings are used in a wide variety of applications. Their capacity to face large tilting moments and loads in any axial or radial directions is essential for orientation purposes in tower cranes, vertical lathe tables or radio telescopes, among others. Besides, slewing bearings are used for yaw and pitch rotations in wind turbines and the orientation of the solar trackers. For that reason, the renewable energy industry is demanding a deeper knowledge about these elements. As it is known, the tendency in renewable energy technologies is to increase the dimensions in order to obtain the maximum possible energy, which increases the involved loads as well. The more demanding work conditions, combined with the need to optimize the machines to make them competitive in the energy market, constitute a demand for a better understanding of the involved components.

In this context, the torque required to rotate the orientation system is a very relevant parameter. An actuation system is needed to apply this torque, so estimating its value is mandatory. In order to correctly estimate the friction torque, the load distribution among the balls (due to acting loads) must be determined as a first step. This was studied by Zupan and Prebil for single row four-point contact slewing bearings, which proposed [1] and experimentally validated [2] a method to study the effect of geometrical parameters and the stiffness of the supporting structures in the load carrying capacity. Later, Amasorrain et al. discussed an analytical model for determining the ball loads where only the stiffness of the contact between the rolling elements and raceways was taken into account [3], dismissing the effect of ring deformation. With the same assumption, Aguirrebeitia et al. developed a procedure to determine the acceptance surface in the load space [4–6], considering ball preload and contact angle variation. Olave et al. used superelement techniques to implement the elastic behaviour of the rings and the adjacent structures in the load distribution problem [7] to Amasorrain's model, proving that this fact can affect significantly the results. The superelement techniques were also used by Plaza et al. for wind turbine applications [8], significantly reducing the computational cost for a particular pitch bearing assembly, without accuracy loss. Another key aspect is the raceway manufacturing errors, whose influence on the ball load distribution was studied by Aithal et al. [9]. In this research line, the authors proposed an analytical procedure to calculate the ball-raceway interferences due to manufacturing errors [10], further studying its effect on the friction torque.

On the other hand, for the friction torque calculation, the behaviour of the ball-raceway contact must be characterized. In this sense, Jones solved ball kinematics for angular contact ball bearings by imposing the equilibrium conditions in the bearing, assuming full sliding in the ball-raceway contact [11]. This work was generalized for four-point contact bearings by Leblanc and Nelias [12,13]. Later, several methods where proposed by Lacroix et al. to account for the flexibility of the rings [14]. All of these works included inertial effects. Nevertheless, in slewing bearings for orientation purposes, the operational velocities are usually

* Corresponding author.
E-mail addresses: iker.heras@ehu.eus (I. Heras), josu.aguirrebeitia@ehu.eus (J. Aguirrebeitia), mikel.abasolo@ehu.eus (M. Abasolo), j.plaza.gonzalez@gmail.com (J. Plaza).

http://dx.doi.org/10.1016/j.triboint.2017.05.011
Received 14 February 2017; Received in revised form 3 May 2017; Accepted 6 May 2017
Available online 18 May 2017

low and thus inertial effects are negligible. Consequently, the load distribution problem and the kinematics can be decoupled and therefore solved separately, which simplifies the solution of the equation system. This way, Joshi et al. particularized the problem for slow speed applications and formulated the friction torque calculation [15].

Like in Jones' work, state of the art models for four-point contact bearings assume full sliding in the contact. However, in slewing bearings for wind turbines or solar trackers, there are only two points in contact in regular working conditions because of the large tilting moments (Fig. 1a), so balls roll like in a typical angular contact bearing, and consequently stick regions will exist in the contact area according to the Heathcote slip [16]. Due to the low velocities, the area of the stick regions can be significant in the contact ellipse (Fig. 1b), contravening the assumption of full sliding made by Jones and Leblanc and Nelias. Thus, it is expected that the described models may not be accurate enough for the friction torque calculation in wind or solar slewing bearings. In this sense, the aim of this work is to define the field of application and therefore the limitations of Leblanc's model [12,13] by Finite Element (FE) analyses using the model previously developed by the authors [17]. Besides, a sensitivity analysis of different parameters in the stick region has been conducted. The results demonstrate that not considering stick regions may lead to inaccurate results when estimating the friction torque in slewing bearings with large tilting moments. Therefore, the need for more complex contact formulations, such as the ones proposed by Kalker [18], is evidenced.

2.. Friction torque calculation models

As explained in the previous section, for slewing bearings the load distribution problem and the kinematics can be decoupled. In this case, Fig. 2 shows the flowchart for the calculation of the friction torque, where the different simulation techniques for each problem are presented. In this work, friction torque is computed from the FE model and the analytical model proposed by Leblanc and Nelias. For both models, the ball loads (Q) and contact angles (α) used for torque calculation are the same (as it will be explained later), so a direct comparison is made between analytical and FE results.

2.1. Finite Element model

The FE model was previously developed by the authors [17] in ANSYS® Workbench. It is a fully parametric model, where only one ball is considered, with the corresponding portions of inner and outer rings. In this study, ball-raceway interferences (δ) for given contact angle (α) are directly applied instead of external loads, as in Ref. [10]; in order to apply these interferences, an offset is introduced to the ball-raceway contact in the first load step, as done by Aithal et al. [9]. Then, in a second step, the inner ring is rotated while the outer one remains fixed. The rotation is gradually introduced, so the kinematics and the contact phenomena can be tracked over time. This study focuses on the local behaviour of the contact and not on the global response of the bearing. In this sense, no deformation of the outer surfaces of the rings is allowed (Fig. 3a); this condition simulates rigid rings, while the contact elasticity is suitably simulated.

The main complexity of the model lies on the correct formulation of the ball-raceway contact. In this sense, the Augmented Lagrange formulation is used, while the stiffness of the contact is updated in each iteration, allowing a maximum mesh penetration of 0.1 μm. A regular high order hexahedron based mesh is used for the contact region (Fig. 3b), which has been adequately refined (Fig. 3c) in comparison with the model from Ref. [10] in order to obtain detailed contact results. Thus, a FE model with $3.6{\cdot}10^6$ degrees of freedom has been used, with a computational cost of approximately 10 h per each analysis in a high performance work station. Taking into account the number of calculations needed for the study presented in this manuscript and the available calculation capacity, a more refined model was unapproachable in terms of computational cost. The model shows no convergence problems with the described contact formulation and mesh.

Several outputs are obtained from the analysis. From the first load step, normal loads at different contact points (Q) are computed. From the second load step, the kinematics of the bearing, the contact status (which shows the stick and slip regions of the contact ellipse) and shear stress field and finally the friction torque are reported. Regarding shear stresses, ANSYS® Workbench only plots the modulus but not the direction, so an APDL macro was developed to export these results and visualize them in Matlab®.

2.2. Analytical model

The particularization of the analytical model of Leblanc and Nelias for the case of slow speeds proposed by Joshi et al. [15] was programmed in Matlab®. Using the ball loads (Q) obtained from the FE model as explained in the previous section, the analytical model solves the kinematics and calculates the contact shear stresses and the friction torque of the bearing. As mentioned, the purpose of this work is to compare analytical and FE results in order to outline the applicability and limitations of the analytical model.

Four parameters must be calculated in the kinematic problem: the angular velocities of the outer ring (ω_o), inner ring (ω_i) and ball (ω_B) for a viewer located in the centre of the ball, and finally the angle β that defines the direction of ω_B [12]. Contact shear stress field and friction torque depend on the relation between ω_o, ω_i, ω_B and β; however, since inertial effects are negligible in slewing bearings, these results do not depend on the rotation speed. Thus, one of the four unknown parameters must be an input for the model, while the other ones will be output parameters; in this work, ω_i was taken as the input parameter (obviously, contact shear stress field and friction torque results were found to be the same regardless of the magnitude of ω_i).

The equilibrium conditions to solve the kinematics are formulated in Ref. [15]. An accurate initial guess of parameters ω_o, ω_i, ω_B and β is necessary to ensure the convergence. In this work, the following expressions are proposed from the assumption that ball-raceway slip is equally divided among the contacting points:

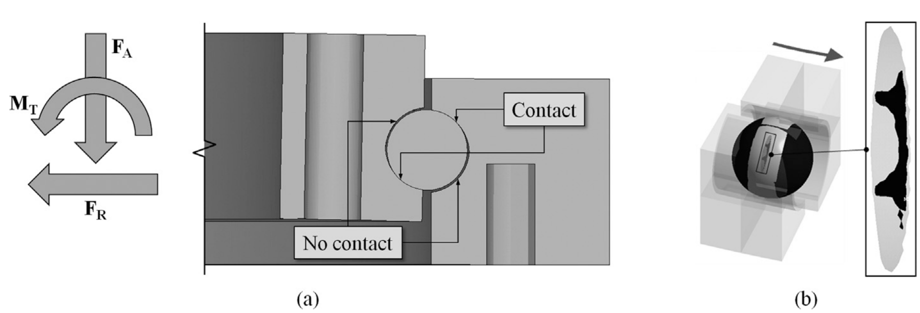

Fig. 1. Two contact point case: (a) deformed shape of a bearing under applied loads; (b) stick region in a two contact point case.