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General static load-carrying capacity for the design and selection of four contact point slewing bearings: Finite element calculations and theoretical model validation

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ABSTRACT

In previous publication, the authors developed a theoretical model to evaluate the static load capacity of four contact point slewing bearings. The method culminated in the formulation of an acceptance surface in the load space, in such a way that the validity of a load combination is assured by an inside-surface checking. In the present work a multiparametric Finite Element model has been developed. Three commercial bearings are analyzed using this model and the FE acceptance surfaces are obtained. From the comparison of the theoretical and the FE surfaces, the strengths and weaknesses of the theoretical model are discussed.

1. Introduction

Slewing bearings are large sized rotary elements used in applications in which large rotational functional elements are involved, such as boring machines, tower cranes, wind turbine generators, etc. There are many different types of slewing bearings depending on the number of rows and on the type of rolling elements. Thus, there are bearings with one, two or three rows, whereas the rolling elements can be balls or rollers. Fig. 1 outlines the topology a four contact point slewing bearing. Fig. 2 shows the usual load system acting on it: axial and radial forces, as well as a tilting moment.

In previous publications, several concepts relevant to the assessment of the static load-carrying capacity of four contact point slewing bearings have been examined. Amasorrain et al. [1] developed a procedure to work out the load distribution in this type of bearings when subjected to combinations of axial and radial forces and tilting moments. Liao and Lin [2] developed a similar procedure where only axial and radial forces were taken into account. Both procedures are analogous to the procedure used by Zupan and Prebil [3] to estimate the influence of geometrical and stiffness parameters on the calculation of the load-carrying capacity. The above works propose a generalization of the equations obtained by Jones [4], in which the load distribution is calculated from the known external loads. Nevertheless, none of these approaches is useful to work out the load combinations that cause the most loaded ball arrives to its critical contact stress value (4200 MPa according to International Organization for Standardization [5]), i.e. to the static failure. In fact, these load combinations would have to be obtained iteratively, with the consequent high computational cost.

In [6] the authors developed a theoretical model under a different focus, which consists on directly calculating the load combinations for which the most loaded element (ball) presents a static failure. In this sense, the works of Sjoväll [7] and Rumbarger [8] were generalized in order to obtain this new theoretical approach. Based on the classical geometrical interference model, it enables to obtain a three-dimensional acceptance condition in the form of a surface inequation in the load space. This acceptance surface, shown in Fig. 3, can be used by the designer as a straightforward way to select a bearing. Figs. 4–6 show respectively the F_A–M_T, F_A–F_R and F_R–M_T curves, i.e. the coordinate planes, of the acceptance surface. Note that the axial load-carrying capacity C_{0a}, whose value can be obtained from standards [5] or even experimentally, was used to normalize the axes of the surface. The contact angle of the balls α (see Fig. 2) is assumed to remain constant, $\alpha = 45^\circ$. Diameter d is the ball center diameter, $(da+di)/2$ in Fig. 7.

The present work presents a multiparametric finite element model of the bearing. It has been used to analyze several commercial bearings and their corresponding acceptance surfaces have been obtained; these surfaces have been compared with the theoretical one so as to validate the theoretical model.

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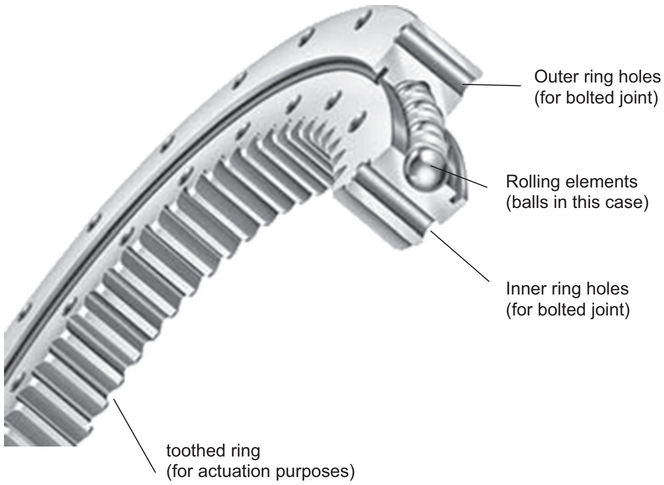

Fig. 1. Topology of a four contact point slewing bearing.

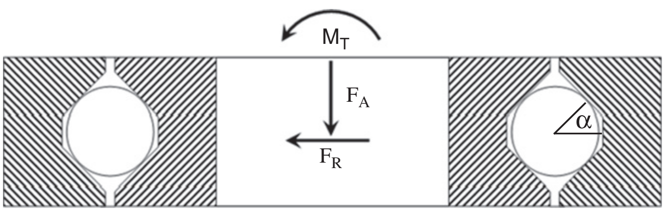

Fig. 2. Load system on a four contact point slewing bearing.

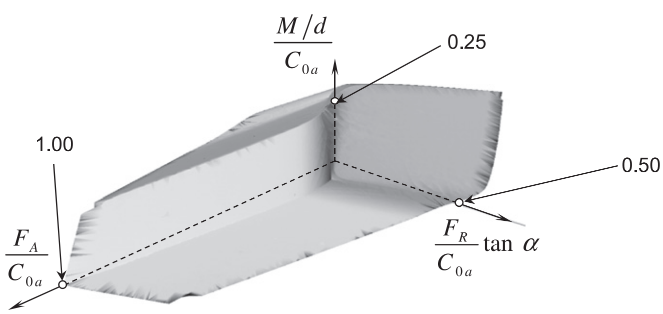

Fig. 3. Acceptance surface of a four contact point slewing bearing [6].

2. The multiparametric FE model

Given a solid model in CAD format (STEP, PARASOLID, CATIA, NX, etc.), any commercial FE package preprocessor can automatically generate a FE mesh for a subsequent analysis. However, when a parametric geometry is to be meshed, a perfect control of the location of nodes and elements is advisable in order to easily and automatically compare results and to avoid differences in the discretization error. Consequently, the design of an appropriate geometry-related mesh tool is considered to be essential.

The geometry of a four contact point slewing bearing is piecewise cyclic-symmetric. It depends on the 15 basic parameters shown in Fig. 7 and listed in Table 1. The mesh tool developed for the finite element model is based upon these 15 parameters. Avoiding non-relevant details, all of the manufacturers observe those parameters and adopt the same value ranges for most of them. Table 1 shows the value ranges used in this work, which correspond to the 220 series in the catalog of the manufacturer IRAUNDI.

Meshing one sector is usually enough in models with cyclic symmetry; afterwards the meshed sector is repeated as many times as the number of sectors in which the model is divided.

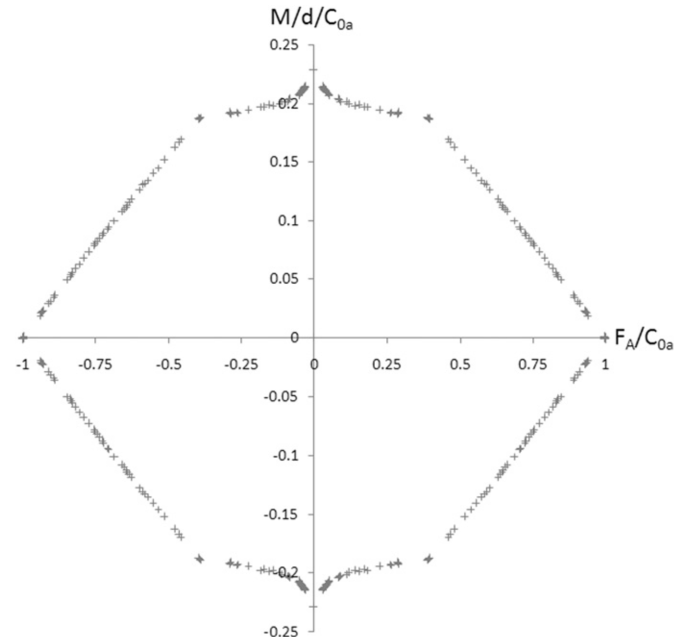

Fig. 4. F_A–M_T curves of the acceptance surface of a four contact point slewing. Theoretical results.

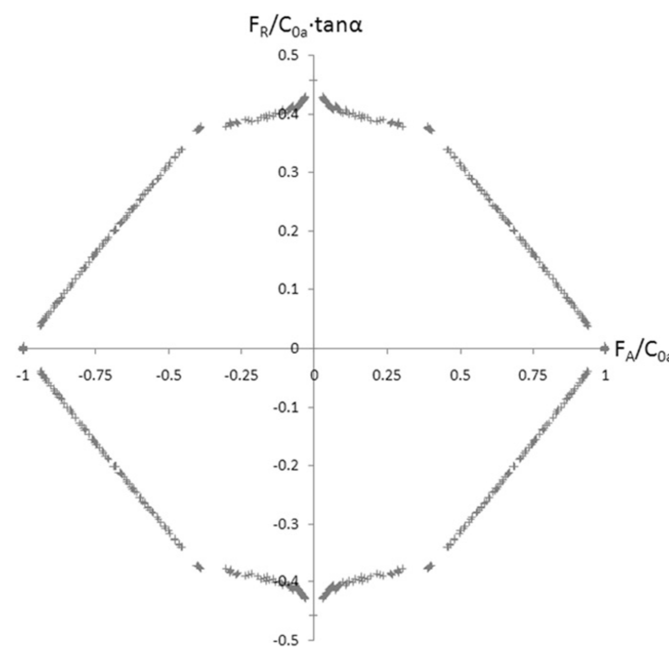

Fig. 5. F_A–F_R curves of the acceptance surface of a four contact point slewing. Theoretical results.

Unfortunately, bearings do not have the same number of balls, holes in the outer ring and holes in the inner ring. Therefore, it is not possible to define a unique sector for the whole bearing. Instead, three different sectors have to be defined as shown in Fig. 8: one for the balls and raceways, one for the outer ring and one for the inner ring. Each one of these sectors has been meshed using a different pattern, and afterwards a bonded contact has been defined among them. The resulting stress discontinuity in the interface between the sectors is located far enough from the ball–raceway contact area; thus, it is considered that it has no influence in the contact pressure of the balls. Being this contact pressure the key magnitude to assess the load capacity of the