FEM model of a precision spindle for simulating manufacturing and assembly errors

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Introduction

The Machine-Tool headstock (which generate the rotational movement of the workpiece/tool) largely determine the geometric accuracy of the machined part. In Machine-Tools, bearings are generally more common, mainly due to their low cost, but also because of their good quality in terms of runout error, vibrations, temperature, and lifespan.

However, it has been experimentally observed that assembly errors and the geometric errors of the other components of the headstock play a crucial role in rotational accuracy. Figure *1* shows the runout error of the same headstock that was assembled (assembly 1), disassembled, and then assembled again (assembly 2). This proves that not only is the precision of the headstock components critical, but so is the assembly process:

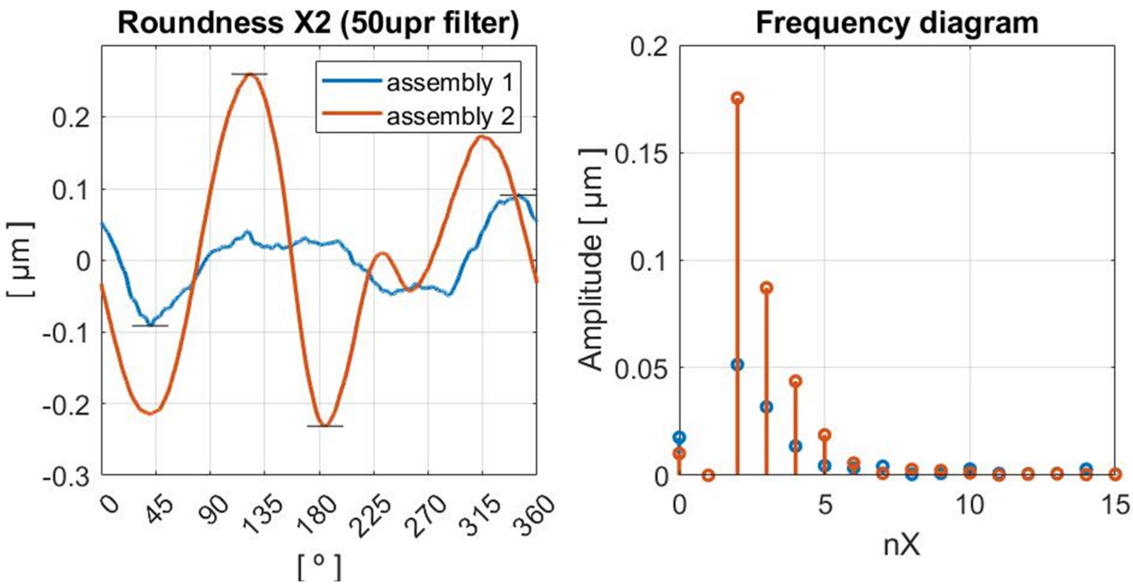


Figure 1. Two measurements of the same headstock, different assembly process

Given the lack of control in manufacturing and assembling precision headstocks guided by bearings, it is essential to have a model that allows manufacturing and assembly errors to be introduced as variables. For this purpose, it is necessary to develop a FEM load distribution model for the headstock and bearings.

The load distribution problem aims to achieve the equilibrium between the external forces or displacements applied to the bearing rings and the contact forces generated on the rolling elements. To do this, it is first necessary to solve the contact problem between the rolling elements and the raceways.

The first notable approach to solving the point contact problem was carried out by Hertz, who solved it by assuming small elastic deformations compared to the radius of curvature of the contacting bodies, which is also known as Hertzian theory [2]. Additionally, he assumed that the surfaces were non-conforming, without imperfections or friction forces, so that the contact problem only considered normal forces.

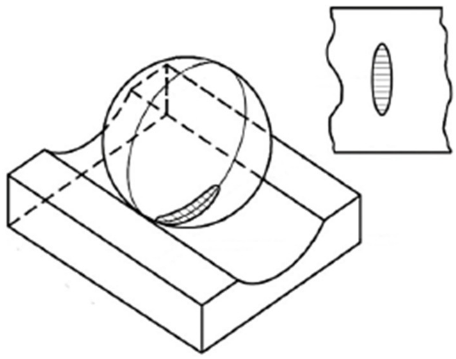


Figure 2. Ball-raceway contact ellipse detail [3]

Brewe y Hamrock [3] later offered a simplified approach to solving the contact problem, which avoids solving the elliptic integrals of Hertzian theory. Their research proposed explicit equations to calculate the major and minor semi-axes and the normal contact deformation of the contact ellipse. Later, Houpert also developed a different approach to avoid the same elliptic integrals for both Hertzian contacts [4] and non-Hertzian contacts [5]. His alternative provided some expressions based on tabulated constants, which were fewer and easier to understand and apply.

Starvin and Manisekar [6] and Aithal et al. [7] used finite element analysis (FEA) to calculate the effect of manufacturing errors on the load capacity and load distribution in large-size angular contact ball bearings, demonstrating that these effects can be significant.

In Golbach's research [8], the balls were replaced by a four-node element (two on each raceway) that fulfilled the objective of simulating the elastic behavior of the contact as well as the variation of the contact angle. To achieve this, the centers of the raceways were rigidly connected to two points on the raceway itself. Then, to represent the contact stiffness, both raceway centers were connected with a nonlinear element. This element modeled the behavior of the ball as two non-conforming elastic bodies based on the work previously conducted by Brewe and Hamrock [3].

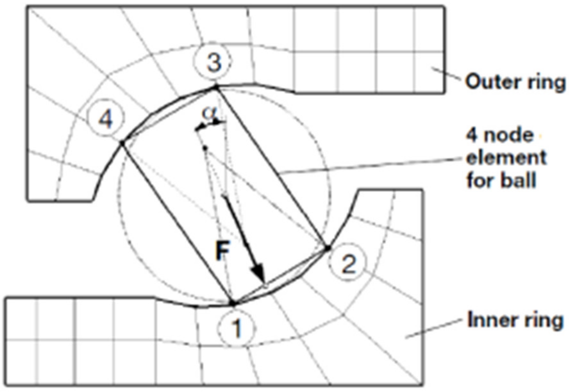


Figure 3. Proposed ball-raceway contact model by Golbach [8]

Daidié et al. [9] adapted Golbach's work to four-point contact slewing bearings, replicating the model for each contact diagonal. In their work, the raceway centers were connected with tension-only spring elements. To represent the ball-raceway load-deformation behavior with these elements, they applied Houpert's work [4].

Finally, Escanciano et al. [10] proposed an analytical methodology to calculate the friction torque in ball slewing bearings, considering the ball preload scatter caused by manufacturing errors and the assembly process. This methodology successfully correlated with experimental tests under compression loads.

FEM modelling

A FEM model of a workhead of a horizontal cylindrical grinding machine was developed, guided by 6 angular contact bearings: 4 in the nose section (near where the workpiece will be placed) and 2 in the rear section, in an "O" configuration:

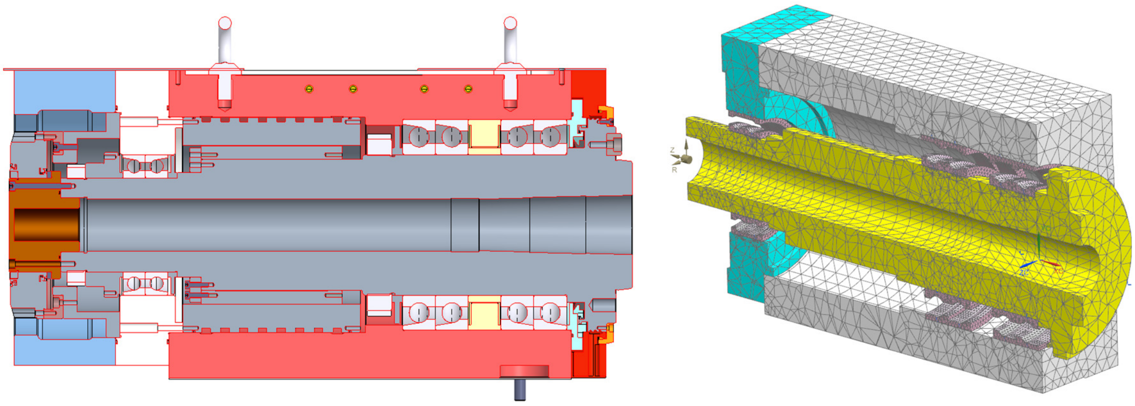


Figure 4. Headstock FEM model

The bearing rings were also modeled as 3D solids, and the rolling elements (in this case, balls) were modeled as proposed by Daidié [9], with 1D rigid elements extending from each raceway to its center of curvature. These two points were connected by a spring with variable stiffness, calculated using Hertzian theory [1]. This spring only works in tension, and the known preload with which the bearings are assembled was considered.

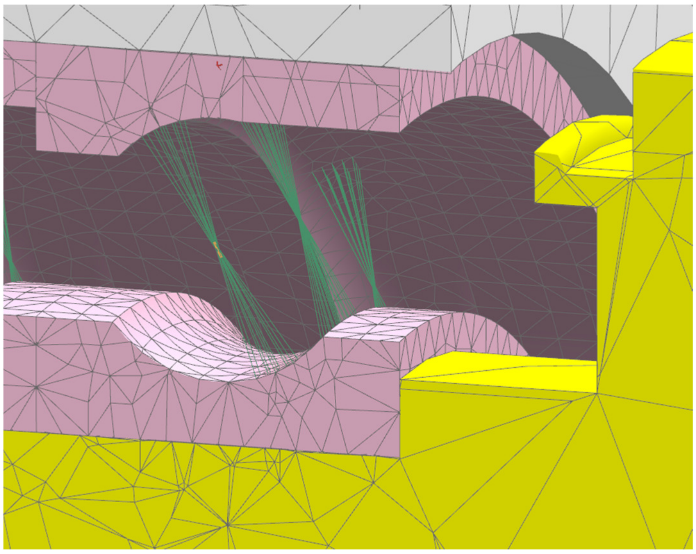


Figure 5. Modelling of the rolling elements

It was necessary to ensure that the local deformation in the raceways was not excessive, so care was taken when defining the area connected to the rigid elements. The footprint (a, b) calculated using the method proposed by Houpert [4] was used as a reference. By taking an area of 4a*8b, the local deformation of the raceways is 10% of the deformation experienced by the spring, which is considered acceptable:

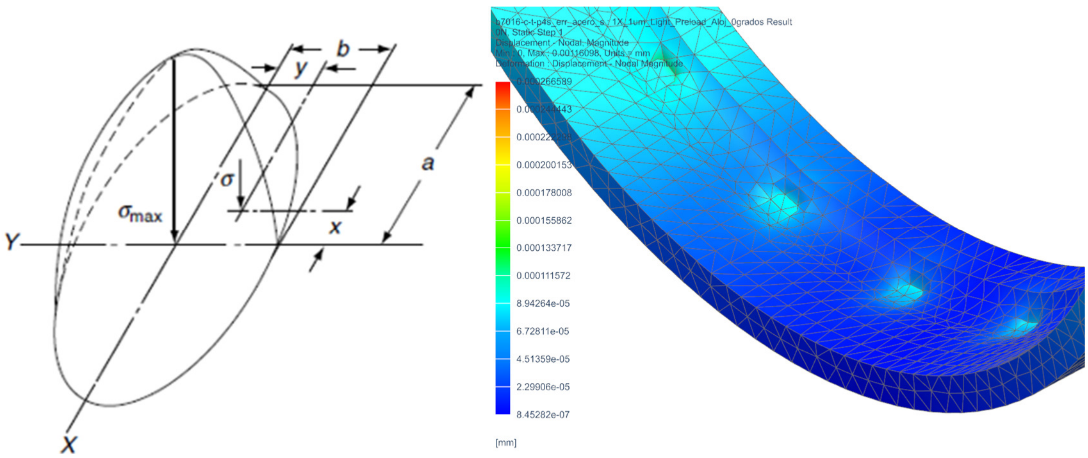


Figure 6. Ball-raceway contact ellipse [3] and raceway local deformation

The bearing model was validated by comparing the results with Schaeffler's commercial software BEARINX® and with an analytical model developed at IDEKO in a Matlab environment.

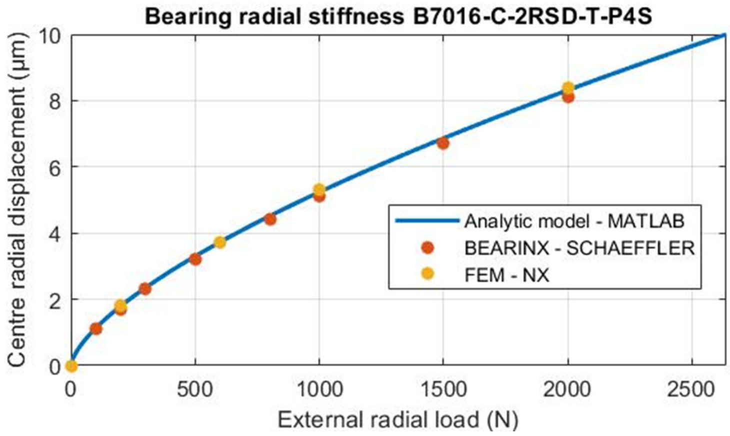
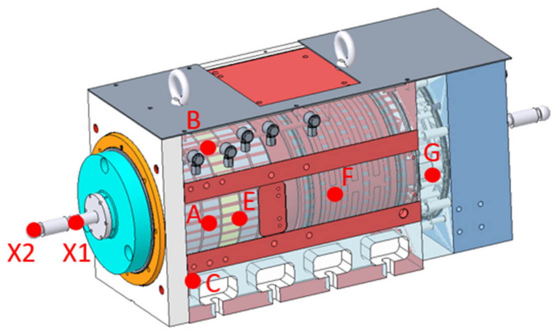


Figure 7. Comparison of the bearing radial stiffness with BEARINX software

In this way, a FEM model validated with software provided by a bearing manufacturer was obtained. In this model, manufacturing errors of components and assembly errors can be introduced by modifying the natural length of each spring.

Experimental measurements

The runout error of a headstock was measured experimentally using capacitive equipment with nanometric resolution on a master verification spindle with two spheres of sphericity below 50nm. The horizontal displacement of various points on the outer body of the headstock was also measured to provide more data for model validation. The results are shown below, filtered with 50upr (units per revolution):



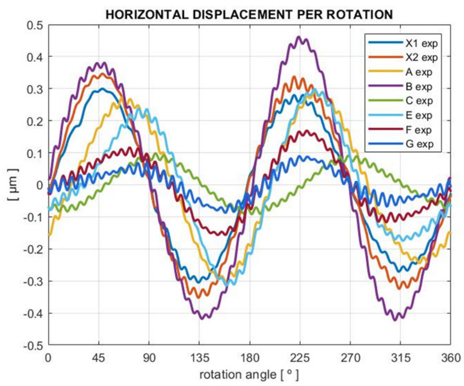


Figure 8. Experimental results

As can be observed, both the runout error of the shaft and the displacement of the headstock have a predominant frequency of 2X, which is very common when using angular contact bearings.

Result and discussion

For the TCP (Tool Center Point) of the headstock to exhibit a runout error with the previously mentioned particular shape and a predominant 2X component, one possible cause seems to be that the shaft has an oval shape roundness error (which would have two maxima and two minima). To replicate this in the FEM model, it is necessary to first calculate how much each spring, representing the stiffness of each rolling element, compresses or decompresses (BD'-BD) due to the shaft's roundness error at each point (dr), as shown in Figure 9. Then, the stiffness curve of the nonlinear spring needs to be shifted to the left or right, as appropriate:

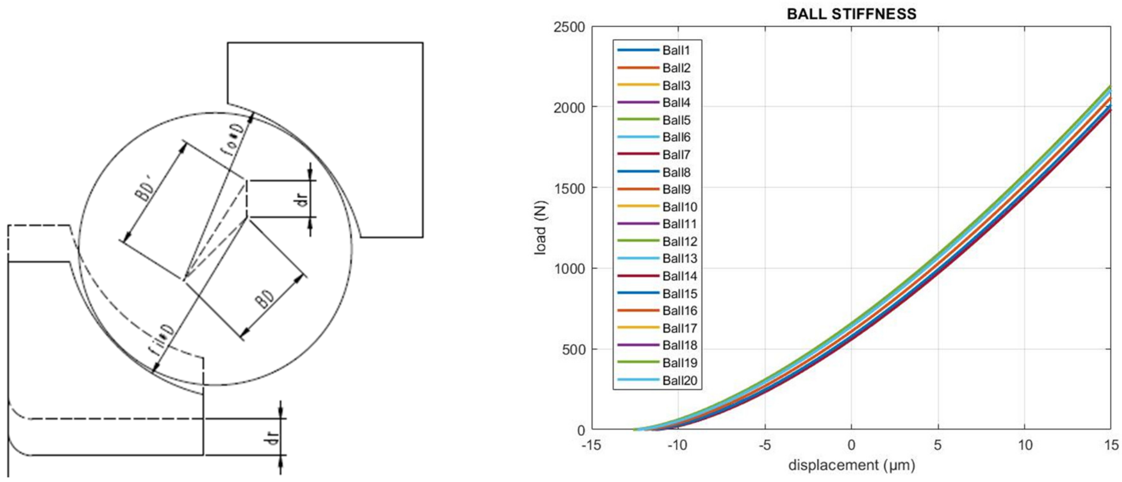
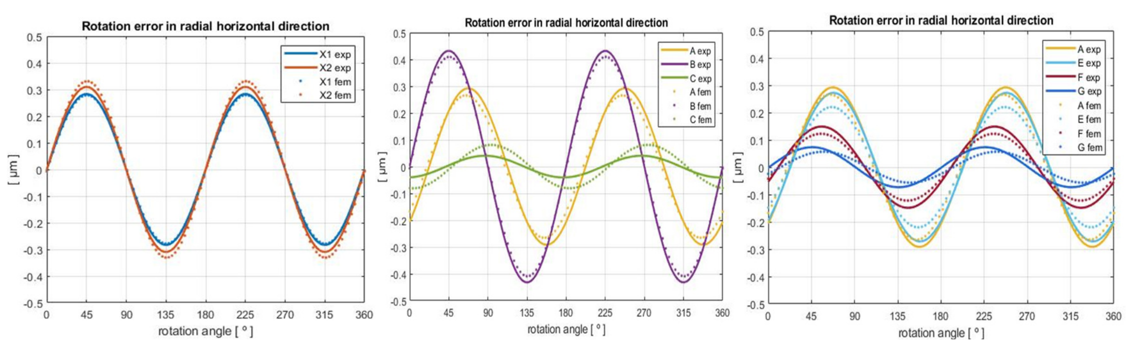


Figure 9. Stiffness curves of the springs representing the front bearings' balls, for a shaft roundness error of 0.75sin(2α)

If the shaft had a roundness error with a 2X shape and an amplitude of 0.75, the headstock displacements would be as follows, compared with the experimental results (considering only the second-order polynomial of the Fourier transform of the experimental results) :



Conclusions and future work

A FEM model of a headstock with angular contact bearings has been developed rigorously and accurately, allowing for the introduction of both manufacturing errors in the components and assembly errors of the headstock. The goal is to understand the influence of various factors on the headstock's runout error and to optimize the assembly process to minimize this error.

The results from the FEM model align very well with the experimentally obtained data when a 2X form error with an amplitude of 0.75μm is introduced to the cylindrical section of the shaft where the four front bearings are mounted.

The shape error of the shaft is the most apparent way to achieve a 2X runout error in a shaft. In future work, the aim is to identify additional combinations of component errors and assembly errors to reproduce the same errors measured experimentally. It is also planned to disassemble the headstock and meticulously measure all components in order to have more information.

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