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Impact of geometric errors in critical components of a precision spindle on runout error through FEM modeling

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Abstract

This study analyses the impact of geometric errors—bearing preload, diametral interference, and ring thickness variations—on spindle runout and workpiece roundness in cylindrical grinding machines. A validated FEM-based model of a grinding machine headstock was developed to introduce manufacturing and assembly errors as variables.
Results show that spindle runout consistently exhibits a 2X frequency, with preload significantly affecting error magnitude. Lower preload reduces runout, while heavier preload increases it. Diametral interference slightly influences runout error within certain tolerances. Furthermore, if a 2X thickness variation is detected in the inner rings, a better result can be achieved by angularly misaligning them with respect to each other.
These findings underscore the need for precise control of geometric errors to optimize spindle performance and machining precision in high-accuracy grinding applications.

Finite Element Method (FEM), Modelling, Ball Bearing, Precision, Spindle

1. Introduction

The Machine-Tool spindles (which generate the rotational movement of the workpiece/tool) largely determine the geometric accuracy of the machined part. In the case of cylindrical grinding machines, as shown in Figure 1 below, the roundness error of the machined workpiece is primarily determined by the radial-horizontal runout error of the headstock:

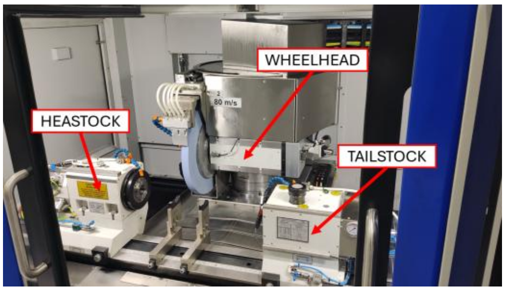

Figure 1. Cylindrical horizontal grinding machine from DANOBAT

The rotational movement of these spindles are commonly guided by rolling technology, by using common bearings, mainly due to their low cost, but also because of their good quality in terms of runout error, vibrations, temperature, and lifespan.

Analysis of the radial runout error of the headstock reveals a consistent pattern, with a predominant frequency occurring twice per revolution of the shaft, referred to as 2X.

On the other hand, the roundness error of the machined workpiece is determined by the radial spindle runout error in the machining plane. In Figure 2, in addition to the predominant 2X frequency of the spindle runout, a higher frequency (28X) is observed in the workpiece roundness. This is due to machining process parameters, specifically the speed ratio between the grinding head and the headstock, which is 27.8:

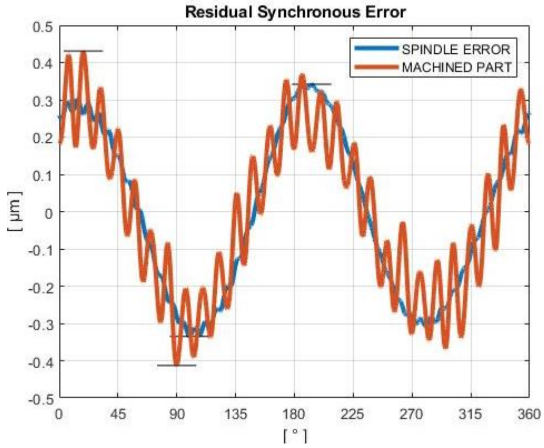

Figure 2. Headstock run-out error in radial-horizontal direction VS workpiece roundness

Since the origin of this 2X frequency and the factors (related to manufacturing and assembly) that influence its amplitude are not known, 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-based 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 00.

Brewe and Hamrock 0 later offered a simplified approach to solving the contact problem, which avoids solving the elliptic integrals of Hertzian theory. Later, Houpert also developed a different approach to avoid the same elliptic integrals for both Hertzian contacts 0 and non-Hertzian contacts 0. His alternative provided some expressions based on tabulated constants, which were fewer and easier to understand and apply.

Starvin and Manisekar 0 and Aithal et al. 0 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 0, the balls were replaced by a four-node element (two on each raceway) that fulfilled the objective of simulating the elastic behaviour of the contact as well as the variation of the contact angle. To achieve this, the centres of the raceways were rigidly connected to two points on the raceway itself. Then, to represent the contact stiffness, both raceway centres were connected with a nonlinear element, as shown in Figure 3a. This element modeled the behaviour of the ball as two non-conforming elastic bodies based on the work previously conducted by Brewe and Hamrock 0.

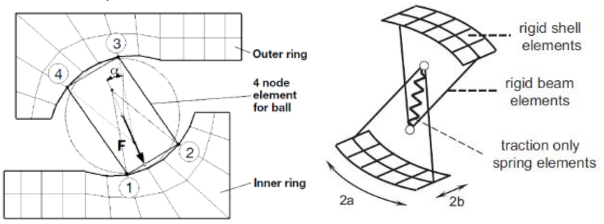

Figure 3. Proposed ball-raceway contact model by Golbach [8], and schematic representation of the model for the ball and the ball–raceway contact [10]

Daidié et al. 0 adapted Golbach's work to four-point contact slewing bearings, replicating the model for each contact diagonal. In their work, the raceway centres were connected with tension-only spring elements. To represent the ball-raceway load-deformation behaviour with these elements, they applied Houpert's work 0.

Finally, Escanciano et al. 0 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, successfully correlated with experimental tests under compression loads.

2. FEM modeling

A FEM model of a headstock 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:

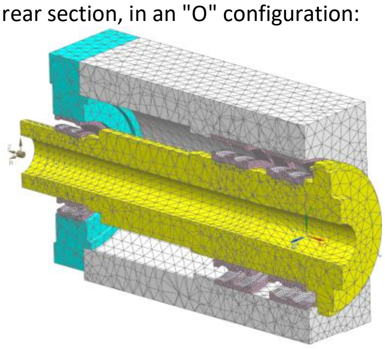

Figure 4. Headstock FEM model

The bearing rings were also modeled as 3D solids, and the rolling elements (in this case, balls) were modeled as proposed by Daidié 0, with 1D rigid elements extending from each raceway to its centre of curvature. These two points were connected by a spring with variable stiffness, calculated using Hertzian theory 0, as shown in Figure 3b. This spring only works in tension, and the known preload with which the bearings are assembled was considered.

To introduce a preload into the rolling elements of the bearings, the inner ring is displaced axially with respect to the outer ring (the displacement value "dr" is defined by the manufacturer). To simulate this effect in the model, the compression or decompression of the spring resulting from the inner ring displacement is calculated, and the stiffness curve of each spring is shifted accordingly, either to the right or left:

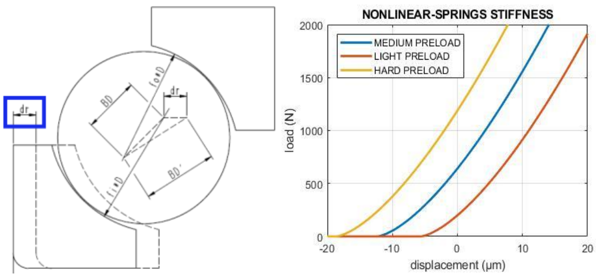

Figure 5. Schematic representation of the section of the bearing and stiffness curves of each spring for different preloads

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.

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:

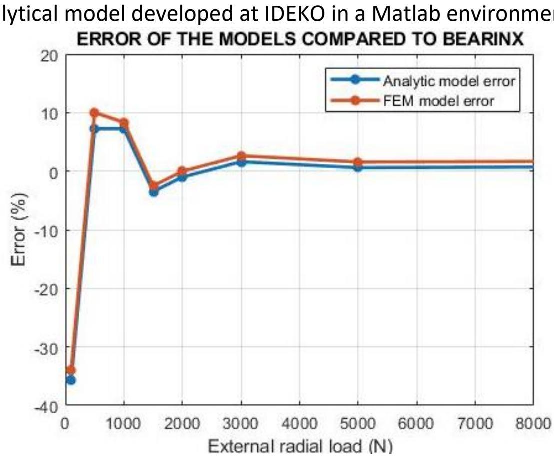

Figure 6. Percentage error of the bearing radial stiffness between the developed models and BEARINX commercial software

The error is higher when the force-displacement values are low, due to the limited resolution of the results provided by the BEARINX software. However, when examining the error at higher force-displacement values, the error of the analytical model is 0.7%, and that of the FEM is 1.7%. 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.

Preliminary simulations helped to understand that the rotation of a shaft with oval shape error can lead to a runout error with a 2X frequency, due to the deformation of the spindle housing, as shown in Figure 7, which displays a section of the spindle at different angular positions of the shaft:

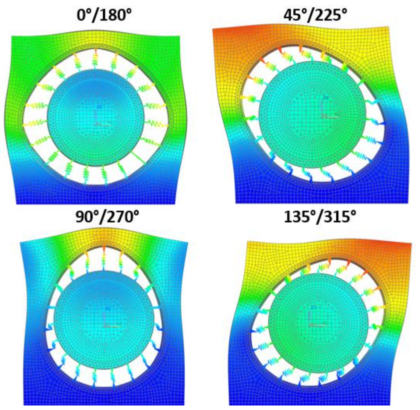

Figure 7. Images of the FEM results of the housing deformation

It can be observed that when the shaft with oval shape error rotates clockwise, the centre of the shaft first shifts to the left (45°), then to the right (135°), then back to the left (225°), and finally again to the right (315°), resulting in a runout error with a predominant 2X frequency.

3. 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 50 nm (https://www.lionprecision.com/probe-mount-and-master-ball-target-details/). 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 50 upr (units per revolution):

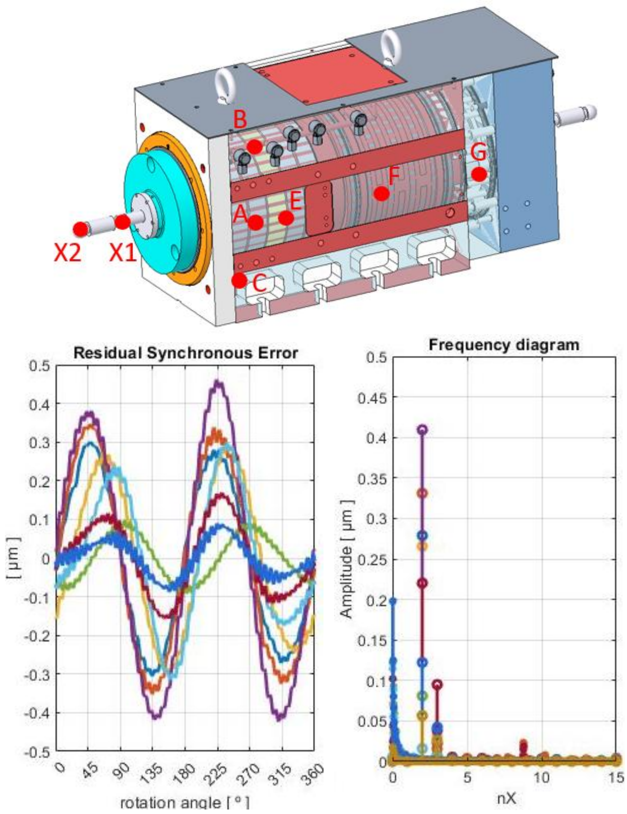

Figure 8. Residual synchronous error and FFT plots of experimental results

As can be observed in Figure 8, 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. This indicates that the housing itself undergoes deformation, as also shown in the FEM results.

4. Model validation

For the TCP (Tool Centre 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 5. Then, the stiffness curve of the nonlinear spring needs to be shifted to the left or right, as explained before.

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

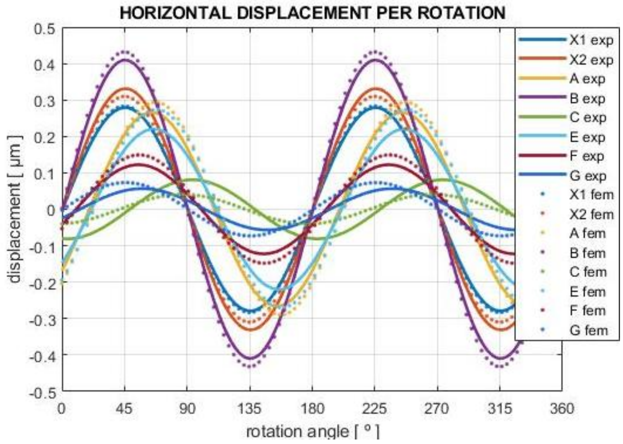

Figure 9. Experimental VS model results

As can be seen in Figure 9, the model values are very similar to those obtained through experimental testing, both on the rotating shaft and on the stator of the spindle.

From this point onward, the focus will be on the horizontal radial displacement, as it determines the roundness error of the machined part in the cylindrical grinder. The error in the vertical radial direction will generate a second-order shape error in the machined part.

5. Sensitivity analysis of bearing geometric errors

Once a validated FEM model of the headstock was obtained, a sensitivity analysis was performed on some geometric errors of the bearing rings. Specifically, the following variations were analysed:

5.1. Preload of Rolling elements

The bearing manufacturer defines three types of preload: light, medium, and heavy preload. Depending on the application, one type should be selected: light preload bearings for applications operating at higher speeds (since friction forces are lower and less heat is generated), heavy preload bearings for applications requiring higher load capacity and stiffness, and medium preload bearings for applications seeking a balance between these characteristics.

Through simulations with different preload settings, it has been observed that lower preload results in a smaller spindle runout error due to a shaft ovality error of 0.75 µm. Specifically, with light preload bearings, the runout error at point X2 decreases by 22.3 % compared to the medium preload case, while with heavy preload bearings, it increases by 14.7 %. These results suggest that adjusting the preload of the bearings can significantly influence the spindle runout error.

5.2. Shaft-bearing and housing-bearing radial interference

The diametral interference of the bearing in its housing has been measured in different headstocks, with the interference ranging from 5 to 20 µm. Varying the interference results in an effect similar to what was observed earlier, namely, a change in the preload of the bearing rolling elements.

For example, if there is a diametral interference of 20 µm (radial interference of 10 µm) in medium preload bearings, the spindle runout error increases by 8 %. This may seem insignificant, but it may need to be considered in certain cases. The interference should be kept within controlled values.

5.3 Thickness of outer ring bearings with 2X shape

The outer and inner diameters of the four frontal bearings in a headstock were measured using a Jenoptik form measurement machine. It was observed that the outer diameter of the outer rings of the bearings exhibits an oval shape, or 2X, with an amplitude of approximately 4 µm. Assuming that the bearing raceways have no shape error, i.e., they are perfect circles, the thickness of the outer ring would exhibit a 2X variation, causing a different preload on each rolling element of the bearings. The runout error of a shaft with a shape error of ovality with an amplitude of 0.75 µm (as before) was simulated, along with the rings and an error in the thickness of the outer bearing rings with a 2X shape of 4 µm amplitude. In the first simulation, the outer rings were mounted with the maximum thickness at 0° and 180°, and in the second simulation, the maximum thickness was at 45° and 225°.

If the maximum thickness is at 0° and 180°, the lateral rolling elements are more preloaded, while the upper and lower rolling elements are less preloaded. This results in no variations in the radial horizontal spindle runout. In the vertical direction, the runout error also does not change, but its mean value, i.e., the 0X, will change, which does not affect the precision of the workpiece.

If the maximum thickness is at 45° and 225°, the rolling elements in this area will be more preloaded, while those at 135° and 315° will be less preloaded. This leads to no variation in the runout error at X2, but the mean error, or 0X, will change by 1.65 µm. This does not affect the precision of the workpiece.

It can be concluded that an error in the thickness of the outer rings with a 2X shape does not affect the spindle runout error.

5.4 Thickness of inner ring bearings with 2X shape

The 0.75 µm amplitude ovality error of the shaft, with which the model was validated, will have the same effect as a shaft with no geometric error and inner bearing rings with 0.75 µm amplitude thickness variations, aligned with each other. In this case, an angular misalignment of the inner rings of the four front bearings, 90° between them, was tested. It was observed that by misaligning the inner rings, the amplitude of the 2X error of the headstock can be reduced. Specifically, by offsetting the rings at 0°90°0°90°, the error is reduced to 6 % of the original value, and by offsetting the rings at 0°90°0°90° the error is reduced to 24 %.

This means that if a 2X shape error is detected in the inner rings of the bearings, a better result can be achieved by angularly misaligning them with respect to each other.

6. Conclusions and future work

This study presents a FEM-based model to analyse the impact of geometric errors on spindle runout and workpiece roundness in cylindrical grinding machines. The spindle runout consistently exhibits a 2X frequency, independent of spindle speed.

Bearing preload and interference significantly affect runout error. A lighter preload reduces it, while a heavier preload increases it by up to 22.3 %. A 20 µm interference raises the error by 8 %, underscoring the need for precise control. The 2X shape error in outer ring thickness has minimal impact, except for a mean error shift at specific alignments. Misaligning inner rings by 90° reduces the 2X error amplitude by 94 %, enhancing performance.

Controlling preload, interference, and ring thickness variations is crucial for optimizing machining precision, with angular misalignment offering a way to mitigate 2X shape errors. Future work will explore non-destructive measurement of ring thickness and analyse additional geometric and assembly errors.

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