Analysis of the influence of geometric and assembly errors in angular contact ball bearing spindles

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Introduction

Machine-tool spindles (responsible of the rotational movement of the workpiece and the tool), determine the geometric precision of the machined parts. In precision applications, such as precision grinding and hard turning, hydrostatic journal bearings are more widely used, while in machine-tool rolling bearings (from now on referred as "bearings") are more common mainly due to their low cost, while maintaining good dynamic and thermal behaviour, and life expectancy. The geometric and assembling precision of the surrounding elements, as well as the precision of the bearing itself, directly affects the motion error of the spindle,.

There is a particular predominant motion error that appears twice per revolution, especially when angular contact ball bearings are used, and it is due to the misalignment between front and rear bearings, as [1] and [2] mention. Figure 1 shows the error motion of a couple of examples of spindles with ball bearings and it can be seen how the 2X is the predominant frequency:

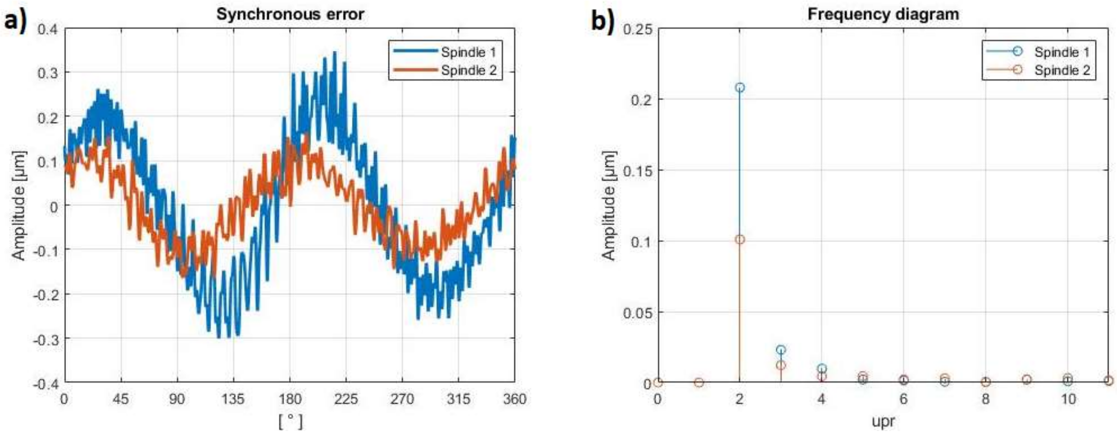

Figure 1. Synchronous radial error (as defined by ISO 230-7) of two angular contact ball bearing based spindles from experimental measurements (a), and their frequency response (b) (the signals have been filtered at 150upr)

The measurements corresponding to Figure 1 have been carried out at the Ideko facilities, and as recommended by [3], capacitive displacement transducers with nanometric resolution have been used, measuring on a master with spheres of below 50nm.

The motivation of this work is to understand this particular behaviour, find the sources of the error and improve it.

Modelling

An analytical model of spindles with angular contact ball bearings has been developed in MATLAB environment, in order to predict the run-out error caused by geometric errors of the shaft and the body (internal and external housings of the bearing).

To a great extent, the work carried out by Lim and Rijnberg is followed. Lim [4] obtained the stiffness matrix for a single bearing and Rijnberg [5] was capable to obtain it for a set of two bearings separated by a distance, so the terms defined by the first author have been used to generate the stiffness matrix and the terms of each bearing have been grouped according to the expressions of the second author. To solve it, it was considered that the system has five degrees of freedom (displacement in X, Y, Z axis and rotation around Y, Z axis). Both authors were based on Hertz's contact law, well explained by [6] and [7], and then they added their own hypotheses to solve the problem.

In order to validate this initial model, a specific bearing configuration was chosen, typically used in workheads of cylindrical grinding machines, with two B71910 bearings at the back and three B7016 at the front, as showed in Figure *2*:

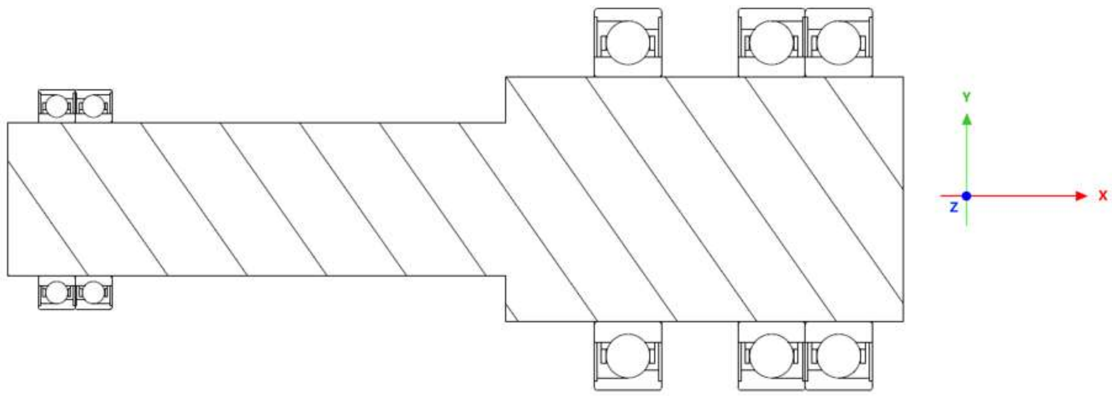

Figure 2. Cylindrical grinding machine workhead section representation

By introducing different external forces as input parameters in the model, the radial stiffness curve was calculated, it was compared with the results of Schaeffler's BEARINX commercial software and they showed a similar results, with a maximum 0.5% deviation.

Modelling of the 2X behaviour

As previously mentioned, experiments show that the spindles have a radial run-out error while they rotate without any external load, caused by geometric and assembling errors of the surrounding components. As this error has a dominant 2X shape the following hypothesis of the cause of this effect was made: The perimetral stiffness of the bearing packages is 2X shaped and rotates, and there is a static radial force. As mentioned by [8], the perimetral radial stiffness of a pair of angular contact ball bearings under a moment takes a 2X shape. This effect could take place with the combination of some of the assembling errors shown in Figure 3:

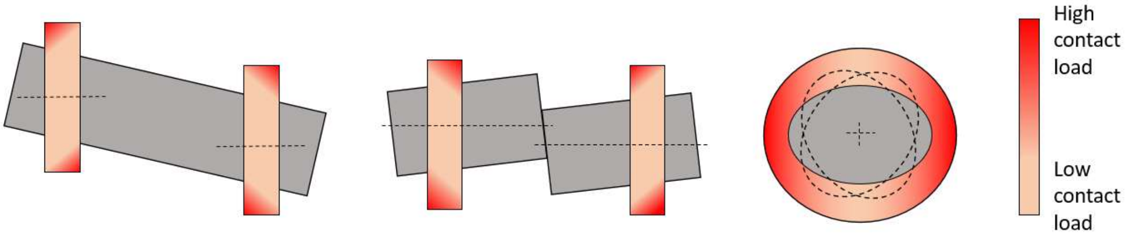

Figure 3. Assembling errors

- Bearing outer housing parallel misalignment.
- Shaft housing parallel misalignment.
- Shaft cylindricity error with 2X shape.

The parallel misalignment between the bearings outer housings induces a static radial force on each bearing package, and the shaft with parallel misalignment provokes a rotary 2X perimetral stiffness. Therefore, the combination of these two effects would result in a 2X run-out error.

Furthermore, as a shaft's 2X shape error provokes a rotary 2X perimetral stiffness on the bearings, the same parallel misalignment between bearings would induce a 2X run-out error too. Therefore, these geometric error options were considered in the model.

Result and discussion

In these calculations the next combinations were evaluated:

- The parallel misalignment of the bearing packages and a non-concentric parallel shaft
- The parallel misalignment of the bearing packages and a shaft with a 2X shape error

For this calculations the flexibility of the shaft wasn't taken into account. The first result was obtained with a 4µm misalignment error between bearings and a 2µm non-concentric error of the shaft, and the second one with a 4µm misalignment error between bearings and a 1.5µm 2X shape error of the shaft.

The rotational error motion of the spindle showed a shape similar to the results of the experimental tests with a predominant 2X, as can be seen in the Figure 4:

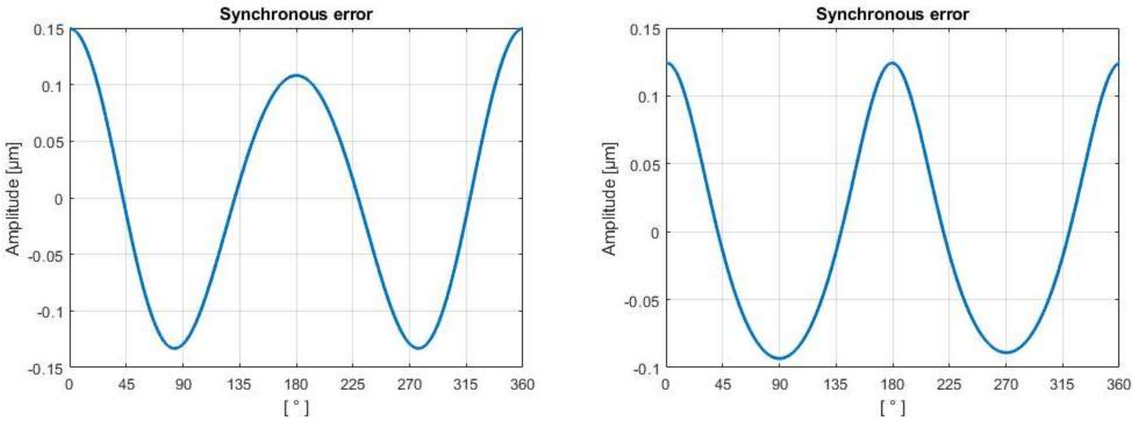

Figure 4. Modelling first results: (a) with outer housings and shaft parallel misalignment, (b) with outer housings misalignment and 2X shaft shape error

Conclusions and future work

First of all, an analytical model of spindles with angular contact bearings was developed, based on Hertz's law of contact. Secondly, it was validated by checking that the radial stiffness curve is the same as the one given by Shaeffler's BEARINX commercial software.

Then, the option of introducing assembling errors was included, in order to be able to predict the run-out error of the spindle and the cause of the 2X run-out error.

Finally, it was verified that with the combination of some geometric errors of the surrounding elements the rotational error motion of the spindle could have a shape similar to the results of the experimental tests, with a predominant 2X.

The geometric errors of the components of the spindle used for the experimental testing are unknown. This is the reason why in future works a test bench will be manufactured with a spindle whose surrounding components' geometric errors will be well known, in order to validate the developed mathematical model. In this bench it is very important that the rear bearing package can be aligned/misaligned with respect to the front package, with a sensitive mechanism.

References

[1] Patel T H and Darpe K 2009 *Experimental investigations on vibration response of misaligned rotors*, Mech. Syst. Signal Process., vol. 23, no. 7, pp. 2236–2252, Oct.

[2] Hariharan V and Srinivasan P S S 2009 *Vibration analysis of misaligned shaft-ball bearing system*, Indian J. Sci. Technol. 2, 45–50.

[3] Marsh E R 2010 *Precision spindle metrology 2nded*, DEStechPublications,Inc.

[4] Lim T C and Singh R 1990 *Vibration Transmission Through Rolling Element Bearings, Part I: Bearing Stiffness Equationtion*, Journal of Sound and Vibration, 139, pp. 179-199.

[5] Rijnberg J L 2007 *Theoretical modeling of the stiffness of angular contact ball bearings using a two DOF and a five DOF approach*. (DCT rapporten; Vol. 2007.129). Technische Universiteit Eindhoven.

[6] Hamrock B J and Anderson W J 1983 *Rolling-element bearings,* NASA Reference Publication 1105.

[7] Harris T A, Kotzalas M N 2007 *Essential concepts of bearing technology,* Rolling bearing analysis. Boca Raton, FL, USA: CRC Press.

[8] Zhang Y, Fang B, Kong L and Li Y 2020 *Effect of the ring misalignment on the service characteristics of ball bearing and rotor system,* Mechanism and Machine Theory.