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Influence of bearing kinematics hypotheses on ball bearing heat generation

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Abstract

Spindle dynamics is a key issue in machining. Thermo-mechanical models of spindle need to be developed to understand and predict the complex behavior of spindles at high speed. Accurate bearing stiffness model is required, since it is a boundary condition of the shaft. Besides, bearings also play an important role in heat generation in the spindle. Bearing models rely on kinematics hypotheses at the ball-race contacts. The aim of the paper is to study the influence of these kinematics hypotheses on bearing heat generation. The different kinematics hypotheses that can be found in literature are presented and implemented into the bearing friction model. Simulations of bearing dynamics are performed and contact force and torque at ball-race contacts are compared. Behavior of the bearing models with different kinematic models is studied using a coupled thermo-mechanical model of a specialized test bed. Loss torque and temperatures simulated are compared and validated by experiments.

Keywords: spindle, thermics, bearing.

1. Introduction

Many issues in machining require simulations to be figured out. The spindle dynamic behavior is very complex, especially for High Speed Machining (HSM) spindles [1]. Thermo-mechanical models have been developed to predict the evolutions of the behavior of spindles at high speed due to thermal and dynamic effects. An accurate prediction of the bearing stiffness is therefore required and needs a proper modeling, since it is the boundary conditions of the spindle model [2].

Bearing models give a relation between the global load applied on the bearing ring with its global displacement and computes bearing stiffness. Analytical approach have been used with 2 and 5DoF models [3,4]. Jones proposed a numerical approach based on the consideration of a rigid displacement of the inner ring [5]. It has been presented in details by Harris [6] and enriched in [7,8]. From the bearing model simulations, the torques and heat generation can be computed and introduced in a thermo-mechanical (TM) model [9,10]. Determination of the ball bearing (BB) torque is subject of research of several authors and companies. The first investigator was Palmgren [3] who set the description of the BB torque. His work was enhanced by Harris [6]. Houpert proposed a detailed model of the passive BB torque in [4, 13].

This paper presents the influence of the kinematics hypotheses of bearing model on the bearing heat generation predictions. A bearing model based on Jones model is presented. Two different kinematic hypotheses are considered to define the pitch angle; from which results the distribution of gyroscopic moment between the inner and outer rings. The classical Outer-Race Control theory is firstly presented and

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then a hybrid hypothesis based on ball equilibrium is considered. A bearing torque model used to express the final torque from the bearing kinematic models outputs and from the temperature is presented. At last, the closed loop spindle thermal simulation model is detailed. Experimental setup to perform validations is finally presented with trial results.

2. Bearing kinematical models

2.1. Geometrical relations

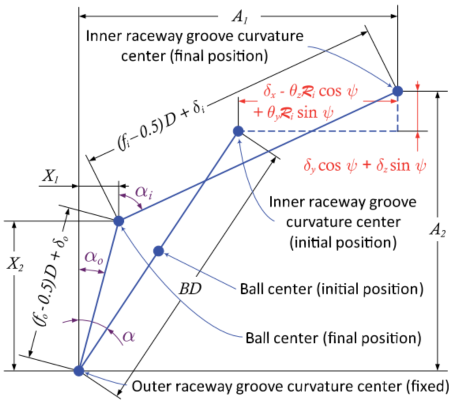

Fig. 1 Position of the raceway groove curvature centers

The bearing model is based on the expression of the inner and outer ring curvature centers as shown in Fig. 1 and detailed in [8]. These positions vary from unloaded bearing position (initial position) to the loaded bearing position (final position), due to modified contact angles α and loads *Q*. The distance between inner raceway groove curvature center from the initial to the final position are projected on the axial and radial direction.

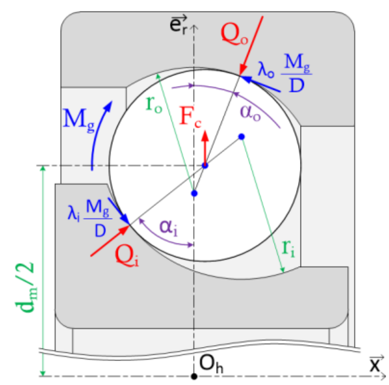

Fig. 2 Local equilibrium of the ball with dynamic effects.

The Newton second law of motion is then applied to each ball to perform local equilibrium (see Fig. 2). Dynamic effects on the ball are considered: centrifugal force F_c and gyroscopic moment M_g. Sufficient reaction forces to the gyroscopic moment are supposed ($\frac{\lambda_k M_g}{D} \leq \mu Q_k, k = i, o$). It leads to the following equations:

$$\begin{cases} Q_i \sin\alpha_i - Q_o \sin\alpha_o + \frac{M_g}{D}(\lambda_i \cos\alpha_i - \lambda_o \cos\alpha_o) = 0 \\ Q_i \cos\alpha_i - Q_o \cos\alpha_o + \frac{M_g}{D}(\lambda_i \sin\alpha_i - \lambda_o \sin\alpha_o) + F_c = 0 \end{cases}$$

The coefficient λ_i and λ_o express the distribution of the gyroscopic moment on inner and respectively outer ring and depend on the chosen kinematic hypotheses (section 2.2). Afterwards, the global equilibrium is computed from the local equilibriums on each balls. The global loads are obtained by summing each ball's quantities.

2.2. Kinematic hypotheses

The first and commonly used kinematic hypotheses proposed by Jones is the Outer-Race Control theory [5]. It supposes that there is pure rolling on the outer raceway (i.e. $\omega_{spin,0} = 0$) and that there is rolling, slipping and spinning on the inner raceway (see Fig. 3). This hypothesis only applies if the frictional moment required to spin the ball about the normal direction of the outer raceway is higher than the one on the inner raceway (otherwise, the inner-race control theory applies). The pitch angle β can be expressed with:

$$\beta = \frac{\sin\alpha_o}{\cos\alpha_o + \gamma}$$

The second hypotheses is based on ball equilibrium from d'Alembert's principle and is called hybrid theory [11,12,8]. A new formula is used to determine the pitch angle:

$$\beta = \frac{C(S+1)\sin\alpha_i + 2\sin\alpha_o}{C(S+1)\cos\alpha_i + 2(\cos\alpha_o + \gamma) + A}$$

$$\text{with} \begin{cases} C = \frac{Q_i a_i L_i}{Q_o a_o L_o} \\ A = \gamma C[\cos(\alpha_i - \alpha_o) - S] \\ S = \frac{1+\gamma\cos\alpha_o}{1-\cos\alpha_i} \end{cases}$$

C corresponds to the ratio of frictional spinning moments at both ball-ring contacts (*a* and *L* refer to the contact ellipse semi-lengths; γ to the ball to bearing diameters ratio).

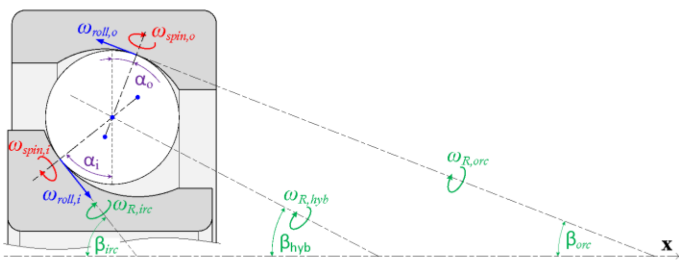

Fig. 3 Impact of kinematic hypotheses on ball spinning and rolling motions.

Table 1 Gyroscopic moment distribution.

	Outer-race control	Hybrid theory
λ_i	0	$2C/(1+C)$
λ_o	2	$2/(1+C)$

The gyroscopic moment distribution associated to each kinematic hypothesis is presented in Figure 3.
The complete stiffness matrix of the bearing is calculated as a Jacobian matrix build from the partial derivatives of loads with respect to the displacement K = [∂f/∂d].

2.3. Comparison of the models

The results obtained with three different models are investigated in this paper. The first one is based on the Jones model presented before, without consideration of gyroscopic moment and assuming the outer-race control theory (ORC). The second one is based on an enriched version of Jones model that considers gyroscopic moments on balls (ORCG). The third one considers the hybrid kinematic theory (HT) to define the distribution of gyroscopic moments and the pitch angle. The comparisons of the outer-ring contact angle and of the bearing radial stiffness obtained from these models at 0 and 10 000 rpm are presented Fig. 4 and Fig. 5. Simulations were performed considering a pure axial force (preload force).

The three models are perfectly in agreement for a static state (0 rpm). At higher speed, the different considerations of control theory and the gyroscopic moment generate noticeable variations between models.

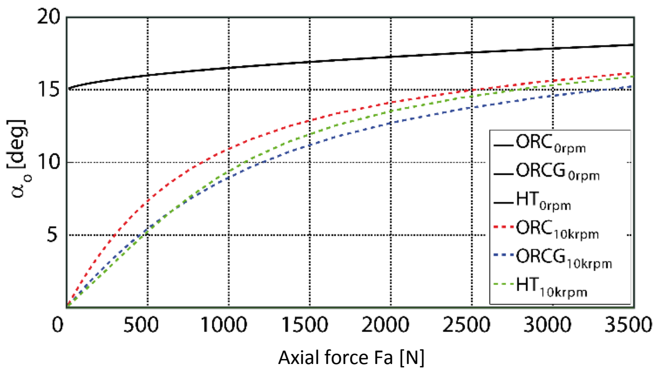

Fig. 4 Comparison of predicted outer ring contact angle for 0 and 10 000 rpm.

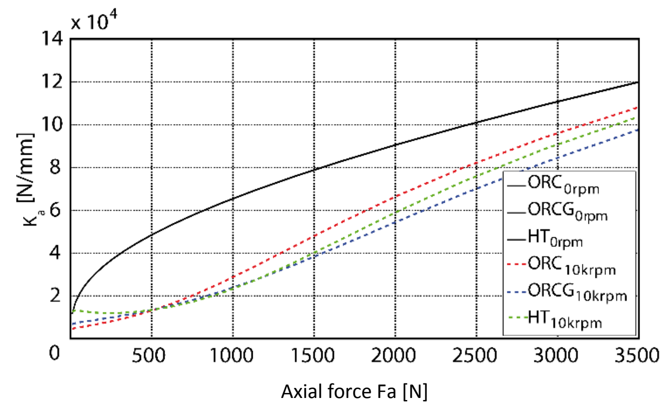

Fig. 5 Comparison of the predicted axial stiffness for 0 and 10 000 rpm.

3. Houpert bearing torque model

Houpert model of passive BB torque is an interesting phenomenological approach and was chosen for this study. Houpert description of the BB torque is presented in [4, 13]. In general the passive torque consists from several components like e.g. Harris model but the detailed approach is different. Generally several torque components are considered in torque models. Houpert model includes hydrodynamic resistance, rolling resistance, curvature shape resistance and resistance caused by pivoting effects. Each component arises from the detailed contact information (semi-major and semi-minor contact ellipses of current contacts between balls and outer/inner rings) which are computed thanks to the dynamic BB model based on kinematic hypothesis. For following calculations the EHL theory in hydrodynamic resistance was adopted. Viscous force which act to the rolling element is considered as:

$$FR_{EHL\_H} = 2{,}765.\,E^{*}.\,R^{2}_{x\,i,o}.\,k_{i,o}{}^{0{,}35}.\,U_{i,o}{}^{0{,}656}W_{i,o}{}^{0{,}466}.\,G_{i,o}{}^{0{,}022}.$$

where U, W and G are classic dimensionless parameters of speed, load and material. Internal velocities at each contact are considered in U parameter. The current contact angles and forces are implemented into the torque model. The lubricant viscosity is used according the current temperature in the bearing. Geometrical parameters of the relevant contact are contained in R_x (Hertzian radius of the curvature) and k parameter (reduced radius ratio). E^* is equivalent Young's modulus of materials in contact.

4. Closed loop spindle TM simulation model

Calculation of the BB frictional torque of a BB within a bearing structure is a complex thermo-mechanical (TM) task. For describing the transient TM interaction between the BB and a structure, a closed loop thermo-mechanical model can be used.
The principle of the TM model is introduced in [10] and a scheme the model interactions is presented in Fig. 6.

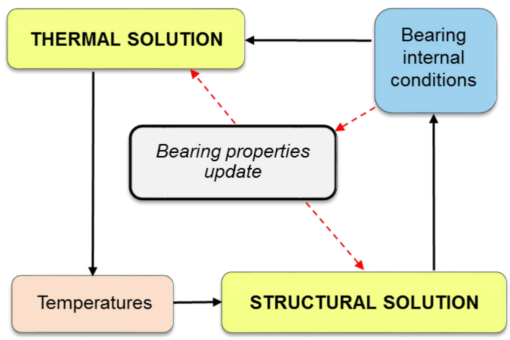

Fig. 6 Scheme of TM model

Structural solution is provided by finite element (FE) analysis, which interacts with the detailed BB kinematical and torque model. The BB model describes bearing internal conditions (kinematic state and internal forces), which allow different bearing configurations ("X" or "O") and BB preloading (force or fixed) to be considered. The FE model covers the surrounding structure (stator and shaft). Thermal solution includes a calculation of the structural thermal deformation and a BB torque model. The TM model allows running a transient simulation, in which the outputs of the FE solution are used as an input for the BB kinematical model and a calculation of a BB internal state update (internal angles, forces). Based on that, an update of the BB friction torque is calculated and consequently a thermal solution is performed. Verification of the boundary conditions of the FE model convective is done based on torque and temperature measurements. The TM model is built in Ansys software.

5. Experimental setup

Validation of the bearing torque and kinematic models is performed through a specific test rig, presented in Fig. 7. The mechanical structure consists of a spindle housing (3) with a couple of FAG B7214-C-T-P4S bearings in “X” configuration with spring preload. Values of preload and, consequently, bearing torque can be modified through an actuator (4) that pushes on one of the outer rings. Oil-air lubrication using a VG68 oil is provided by an oil-air mist system (dosing of 0.1 g of oil per hour and bearing is used). The shaft is driven by an electric motor (1), through the torque sensor (2). Preloading force, speed, torque, temperatures of the outer rings and the structure are measured continuously.

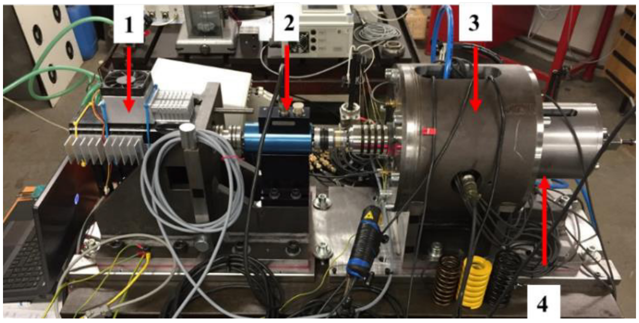

Fig. 7 Measurement of bearing torque on a test bench.

6. Results and discussion

Comparison of the BB friction torque prediction by using three BB kinematical models, discussed in this paper, and a comparison with the measured data is given in Fig. 8. Friction torque is calculated by using the TM model of the test rig, introduced in the section 5. Conditions applied are 10 000 rpm and 800 N axial preload, which correspond to the typical operating conditions of the BB used.

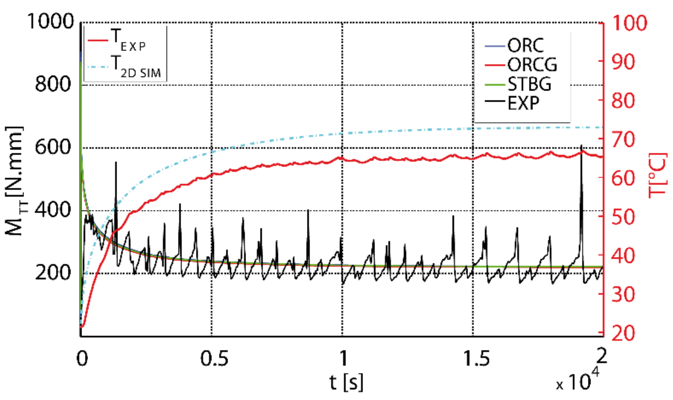

Fig. 8 Comparison of bearing torque and temperature, between simulations and experiments at 10 000 rpm.

The BB kinematical models are marked as ORC (Outer race theory without gyroscopic effects), ORCG (ORC with gyroscopic effects) and STBG (hybrid model). It may be seen that all of the BB torque models predict almost the same level of the friction torque (matching within 2%). Besides, a good agreement of the predicted torque values with the average experimental data can be noticed. Sharp peaks in the experimental data result from oil injections into the bearing, which is a transient effect not covered by the BB torque model. Fig. 8 also illustrates the temperature raised obtained experimentally (in red) and by simulation (in dashed blue). The difference between the simulated and measured temperature is approximatively 12 %.

7. Conclusion

Three ball bearing kinematical models with different gyroscopic moment distribution hypothesis have been investigated with respect to the ball bearing friction torque prediction. The bearing torque models have been used together with the Houpert bearing torque model. Validation has been done using a specialized test rig. It has been shown that different kinematical models lead to noticeable variation of the bearing internal state (contact angles, stiffness) prediction especially for higher preload values. Impact of the bearing kinematical models on the torque prediction has been tested for lower level of the bearing preload and speed, which showed almost no effect. Effect of higher speeds and bearing preload is a perspective of this work and might lead to higher variations between kinematic models.

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