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## Investigation of the evolution of modal behavior of HSM spindle at high speed

Clément RABRÉAU<sup>a,\*</sup>, Mathieu RITOU<sup>a</sup>, Sébastien LE LOCH<sup>a</sup>, Benoit FURET<sup>a</sup>

<sup>a</sup>*Institut de Recherche en Communication et Cybernétique de Nantes (UMR CNRS 6597) – Université de Nantes, 2 avenue du Professeur Jean Rouxell, 44475 Carquefou, France*

\* Corresponding author. Tel.: +33 2-28-09-20-00; E-mail address: [clement.rabreau@univ-nantes.fr](mailto:clement.rabreau@univ-nantes.fr)

### Abstract

The cutting stability and the lifetime of High Speed Machining spindles highly depend on its modal behavior. A specific electromagnetic excitation device is used to measure the Frequency Response Function of the spindle at high speeds. The evolution with speed of the eigenfrequencies and of the radial stiffness is analyzed. A numerical model is developed to obtain the associated modes shapes; since they cannot be measured experimentally. The simulations, which are in good agreement with the experiments, enable an understanding of the evolutions of the spindle dynamic, and notably of the evolution of modal coupling at high speed.

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*Keywords:* Spindle; Dynamics; Simulation

### 1. Introduction

The dynamic behavior of High Speed Machining (HSM) spindle is nowadays still an issue. Problems caused by instabilities induce faults on machined parts and reduction of tool and spindle lifetime. The research of optimal cutting conditions to avoid unwanted vibrations is often done by empirical means. The dynamic behavior of the spindle-tool assembly is measured by a tap testing experiment, and the lobe theory is used to assess the instability zones to avoid during machining [3]. Several studies have shown the influence of dynamic effects on bearing behavior at high speeds and therefore on the spindle-tool assembly behavior [2,5,6].

The evolution of the dynamic behavior of the spindle at high speed have been studied with the classical tap testing method [2,11]. Non-contact excitation systems have also been developed to make measurement at higher speeds with better accuracy [1,4,7]. In these studies, the displacement at the tool-tip is measured with inductive sensors. Rantatalo et al., used a Laser Doppler vibrometer to make the displacement measurement [10]. A similar system was also developed at the

IRCCyN Laboratory, enabling  $\pm 80\text{N}$  up to 5kHz on industrial spindle [9].

The in-depth analysis of the experimental results is however limited due to the lack of measured points. Indeed, the accessibility is very limited inside the spindle. Therefore, the mode shapes associated to the measured eigenfrequencies cannot be measured precisely. Numerical simulation have been used to improve the spindle behavior analysis at high speed, especially the impact of bearings [5,13]. The evolution of rotor mode shapes with speed has also been studied [12].

This paper presents the study of the dynamic behavior of an industrial spindle through experimental measurements carried out with an electromagnetic excitation device. The Frequency Response Functions (FRFs) as well as the evolution of the radial stiffness of the spindle with speed are analyzed. These results were then compared to simulations made with a shaft modeled with 3D finite elements. The boundary conditions of the model are defined with an updated 5dof analytical bearing model that considers the dynamics effects. The modes shapes associated to the eigenfrequencies are presented. The measured and simulated evolutions of the frequencies with spindle speed are showed on a Campbell

diagram. The occurrence of mode coupling at high speed is investigated. The role of a numerical model on the study of the dynamic behavior of HSM spindles is discussed.

## 2. Experimentations

### 2.1. Experimental setup

An electromagnetic excitation device has been developed to apply a dynamic load to spindle shafts while rotating, see Fig. 1. The device is composed of two pairs of electromagnets placed in a perpendicular arrangement. Each electromagnet of a pair is placed to apply mutual repulsion and are supplied with a magnetization current. A blank tool with an HSK-63A attachment is mounted on the spindle. A swept-sine excitation signal is generated from PWM Amplifiers controlled by a LabVIEW interface. The radial displacement at the tool-tip is measured with inductive sensors linked to the interface with a National Instrument CompactDAQ system.

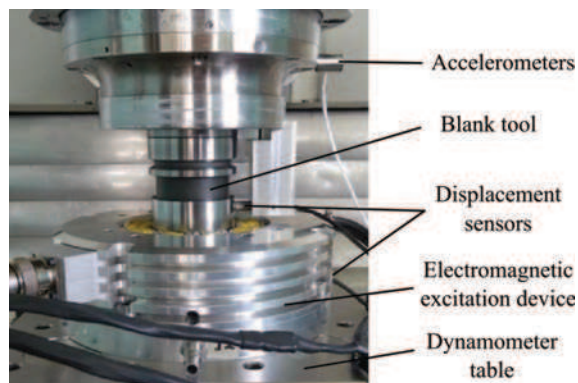


Fig. 1. Electromagnetic excitation device.

Measurement was made on an industrial Fischer MFW1709 electrospindle (24000rpm - 40kW). This spindle is equipped with two tandems of hybrid bearings in back to back arrangement with an elastic preload. Two types of experiments can be performed with this experimental setup: dynamic measurement, with a swept sine from 50 to 5000Hz to make Frequency Response Functions (FRFs) measurements at different speed and quasi-static measurement to obtain radial stiffness variation with speed. These measurements were made at speeds going from 4000 to 24000rpm after a warm up cycle to perform the experiments at a constant steady state temperature.

### 2.2. Experimental results

#### 2.2.1. Measurement post-processing

The experimental FRFs is obtained from the measured displacement at the tool tip and from the applied load identified with the measured excitation current. The  $H_1$  formulation is used as expressed in equation (1), with  $S_{FX}$  the cross power spectrum between the load signal and the

measured displacement and  $S_{XX}$  the auto spectrum of the measured displacement.

$$H_1 = \frac{S_{FX}}{S_{XX}} \quad (1)$$

A specific post-processing on the computed FRF is applied to suppress measurement noise on the result and to suppress the harmonics components caused by the runout of the measured target on the tool.

Fig. 2 shows the computed FRF of the signal without and with the post-processing steps.

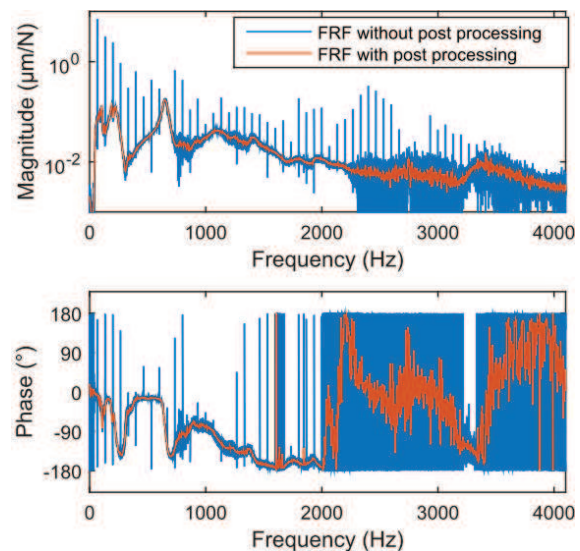


Fig. 2. Illustration of the post-processing effects on Frequency Response Functions.

The radial stiffness is obtained from the quasi-static measurement. It is the ratio of the applied load obtained from the dynamometer table over the mean measured displacement at the tool tip.

#### 2.2.2. Dynamic measurement

Fig. 4 shows the experimental FRFs measured at the tool-tip for different spindle speed. A previous analysis of the experimental results allowed to dissociate machine-tool vibration modes from spindle modes [9]. The spindle behavior is observable for frequencies upper than 300Hz (see Fig. 5). The principal mode at 650Hz at 4000rpm, is decreasing at higher speed due to dynamic effects on the bearings and on the shaft. At low speed, an other highly damped mode is visible at around 1100Hz however, it is difficult to describe its behavior without further information. A mode seems to appear at 19000rpm and end-up at 24000rpm with a frequency of 780Hz.

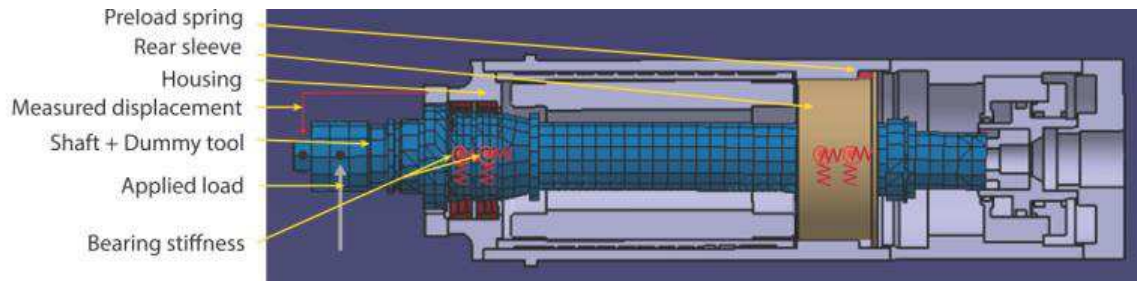


Fig. 3. Numerical model.

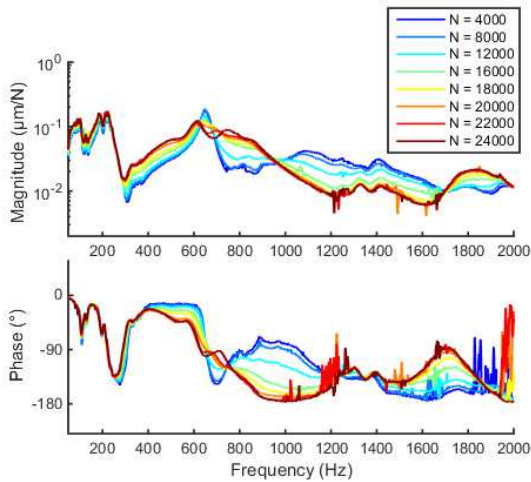


Fig. 4. Frequency Response Functions measured at the tool-tip for different spindle speed.

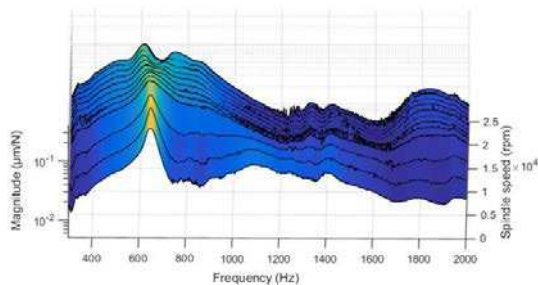


Fig. 5. Evolution of Frequency Response Function measured at the tool-tip with spindle speed.

### 2.2.3. Quasi-static measurement

Fig. 6 shows the evolution of the radial stiffness of the spindle with speed. There is an important drop of the stiffness of about 60%, which is due to the dynamic effects on the bearing balls.

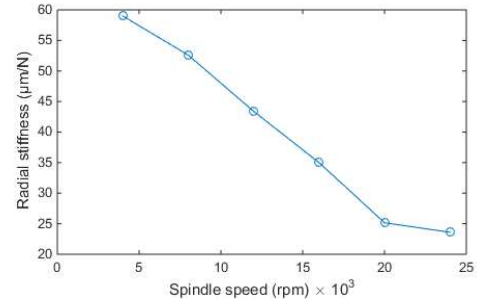


Fig. 6. Evolution with spindle speed of the quasi-static stiffness at tool-tip.

## 3. Numerical simulations

In order to have a better understanding of the experiments results and therefore of the evolution of the spindle behavior with speed, a numerical model have been developed.

### 3.1. Model development

The main components of the numerical simulations are presented in Fig. 3. The shaft and the blank-tool are modeled by 3D finite-elements. Mass and stiffness have been added on the location of the motor with 2D finite elements. The HSK interface is considered rigid. The bearings behavior is taken into account through  $5 \times 5$  stiffness matrices depending on the spindle speed. These matrices are obtained from the updating of an analytical model of the axial behavior of the spindle that includes a 5DoF model of bearing [8]. This axial model computes the axial equilibrium of the preloaded system with regard to the speed and the axial load. The updating gives the operating preload and other bearing parameters required to precisely predict the axial behavior of the spindle as well as the bearing stiffness. The relevance of the radial stiffness obtained with the updating is assumed to be the same as the axial one. This initial model takes into account the dynamic effects in the bearing: centrifugal forces and gyroscopic moments on the balls, and also macroscopic deformation of the shaft due to centrifugal forces. The presence of a stroke limit and of friction between the rear sleeve and the housing is also considered. The rear sleeve and the housing are assumed to be rigid. The movement of the rear sleeve is limited to a translation in the axial direction. An axial stiffness between the housing and the sleeve models the

preload stiffness, whose value is also obtained with the initial axial model. The behavior of the machine-tool is neglected in a first approach. The LMS software Virtual.Lab was used for the simulations.

3.2. Model validation

Fig. 7 shows a comparison between the numerical model and experiments results at 4000, 16000 and 24000rpm. The shaft and bearing damping values are set manually as it is not the focus of this study. The principal modes are visible in the simulation results at the different speeds: 650Hz and 1090Hz at 4000rpm, 645Hz and 1850Hz at 16000rpm and 640Hz, 740Hz and 1825Hz at 24000rpm. The mean error between the simulation and the experiments for these frequencies is about 4%. The simulated frequencies, even if they are not perfectly predicted are accurate enough to assess the correspondence between simulated mode shapes and real mode shapes.

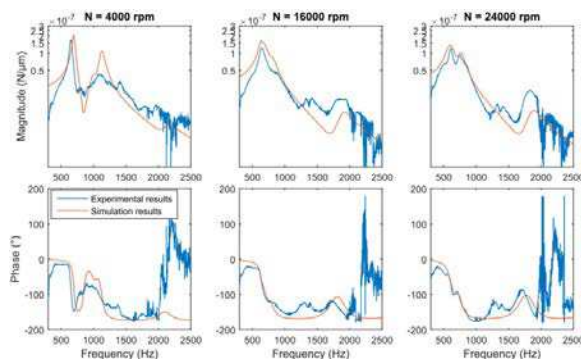


Fig. 7. Comparison between simulated and experimental FRFs at the tool tip.

3.3. Analysis of simulation results

Fig. 8 presents the simulated evolution of the FRFs at the tool tip with spindle speed. The main mode at 650Hz is well predicted by the model, as well as the mode at 1900Hz that have a similar evolution of amplitude. Two modes between 900Hz and 1200Hz appear in the simulation at lower spindle speed and seems to be highly damped on the experimental results. At higher speed the simulations and the experiments both shows a similar mode at 800Hz.

Table 1. Basic mode shapes designation and associated phenomena.

Name	Description
<i>Bend1</i>	1 <sup>st</sup> bending mode of the shaft
<i>RearB</i>	Vibration of the rear bearing
<i>FrontB</i>	Vibration of the front bearing
<i>Bend2</i>	2 <sup>nd</sup> bending mode of the shaft

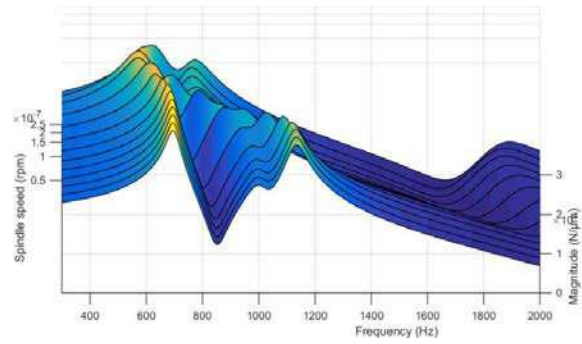


Fig. 8. Evolution of simulated FRFs with speed.

The simulated mode shapes are presented on Fig. 9 for the spindle at low speed (4000rpm) and high speed (24000rpm). The displacements of 11 points distributed on the shaft axis are used to reconstruct the mode shapes and to analyze which phenomenon is mainly associated to a given mode. Four different basic mode shapes can be identified and are presented in Table 1. The description of the phenomena involved in each basic mode shape is also given.

Due to the variation of the frequencies of the simulated modes with spindle speed, the dominant phenomenon visible on each mode shape evolves with speed.

4. Discussion

The observation of the evolution of the simulated eigenfrequencies enables a better understanding of the experimental results. For example, the dominant mode at 650Hz at 4000rpm, whose frequency is slightly decreasing with speed, corresponds to the first bending mode of the shaft (see Fig. 5).

The Campbell diagram presented in Fig. 10 shows the evolution of the frequencies of all the four simulated modes with spindle speed. They are compared with experimental frequencies extracted by modal analysis. The colors highlight which of the four basic mode shapes dominates a given mode.

The main mode at 650Hz is well correlated with the simulation and corresponds to the first bending mode *Bend1* up to 16000rpm. A little decrease of the mode frequency can be observed both on simulations and experiments. It can be explained by the bearing stiffness loss that has an impact on bending modes.

At higher speed, due to the significant drop with bearing modes, the dominant mode shape becomes a front bearing mode (*FrontB*).

The second main mode that was previously associated to a rear bearing mode (*RearB*) becomes visible at 20000rpm on experimental results and turned into a bending mode.

As it seems to be highly damped, this mode doesn't appear clearly on the experiments.

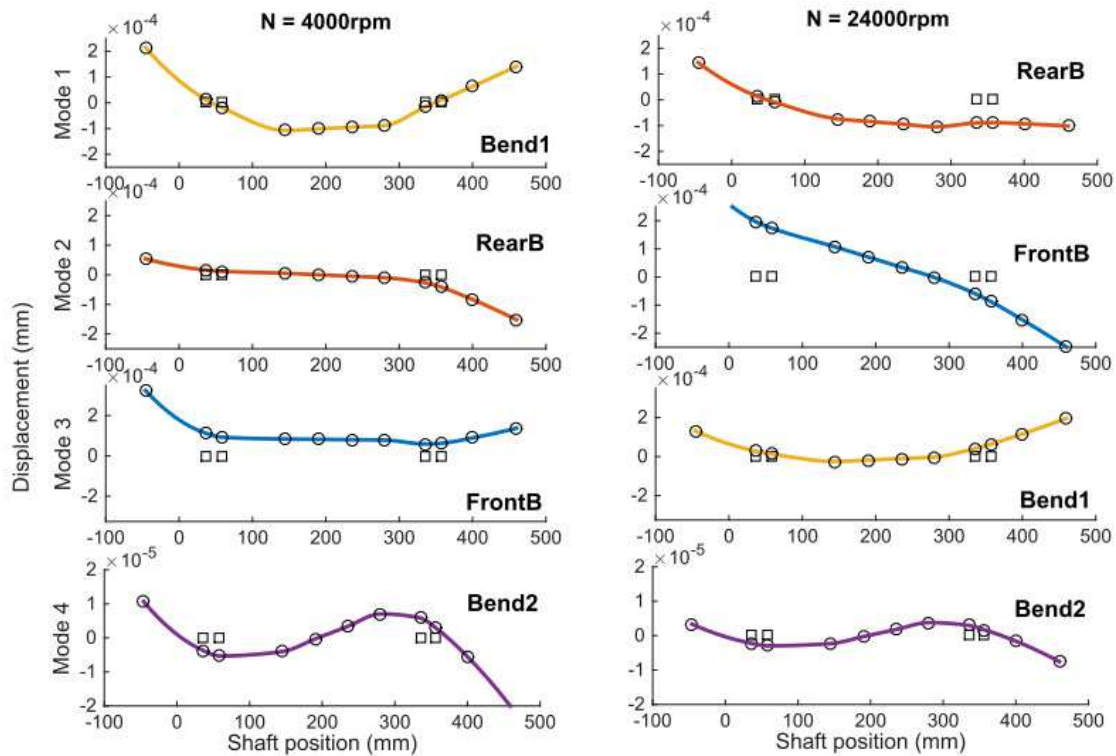


Fig. 9. Spindle mode shapes with indication on the dominant associated phenomena.

At low speed, at 1100Hz the front bearing mode is visible on the experiments but is also very damped and disappear as the two bearing modes get closer.

The second bending mode *Bend2* at 2kHz is visible on the experimental measurement, however it's frequency at low speed might not be well extracted due to the high damping on this mode.

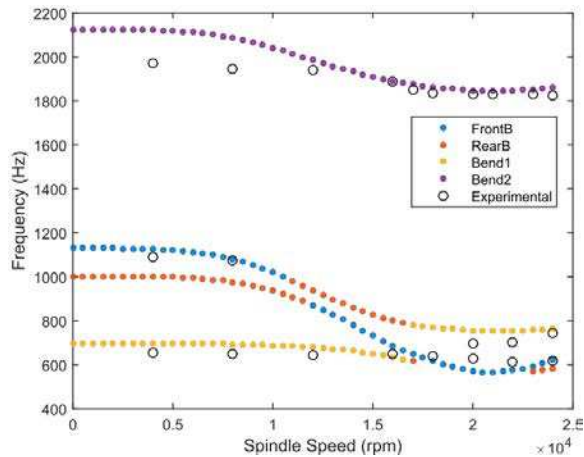


Fig. 10. Comparison of the evolution of simulated and experimental frequencies with dominant phenomena.

Both the frequencies dominated by the bending of the shaft and vibration of the bearings tends to decrease with speed. This can be explained with the stiffness loss of the bearings due to the dynamics effects on the balls. When the frequencies are distant, one basic mode shape relative to one phenomenon is dominant. As the frequencies of the modes get closer to each other, its mode shapes evolve due to the contribution of the different phenomenon.

## 5. Conclusion

The experimental setup measuring the evolution of the dynamic behavior of HSM spindle has been presented. The results of the experiments have been shown: the quasi-static stiffness decreases with spindle speed and the frequencies have a complex behavior. In order to analyze the frequencies behavior, a spindle model was developed based on a finite element model of the shaft, associated to an analytic bearing model that consider dynamic effects. The evolutions of simulated FRFs with speed have been compared to the experiments, which validates the model. For each frequency, the mode shapes have been studied and a dominant phenomenon has been identified. The evolution with spindle speed of the dominant phenomena for each mode helped to explain the complex evolution of the FRFs of the spindle. The impact of the bearing stiffness loss with speed on both the spindle quasi-static stiffness and main frequencies has been highlighted. The use of simulations allowed to explain the

vanishing of certain modes as the speed increase and the emergence of new modes at higher speed.

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