Non-contact measurements and modelling of milling machine tool vibrations

Matti Rantatalo

Luleå University of Technology
Department of Human Work Sciences
Division of Sound & Vibration

Non-contact measurements and modelling of milling machine tool vibrations

Matti Rantatalo

Division of Sound & Vibration
Luleå University of Technology
SE-971 87 Luleå
Sweden

Key words: machine tool spindle, centrifugal, gyroscopic, non-contact measurement, angular contact ball bearings, machine tool chatter, laser Doppler Vibrometry, speckle noise, rotor bearing systems, FEM, magnetic excitation, inductive displacement sensors.
ACKNOWLEDGEMENTS

First of all I would like to thank my loving and encouraging family; Christina, Uno, Kerstin and Marianne, my parents Magda and Hugo and my parents in-laws Mona and Bengt - you all made it possible for me to combine a loving family life with the effort that must be invested in writing the thesis. Secondly, I would like to thank my supervisor, Professor Anders Ågren, for giving me this once in a lifetime opportunity to participate in a PhD study. All his help during my studies and his being a great boss made all the difference. I would also like to express my gratitude to Dr. Per Gren and Dr. Jan-Olof Aidanpää of Luleå University of Technology (LTU) and Bo Göransson of SKF for their help, dedication and genuine interest in my work. I would also like to thank my colleagues Peter Norman and Kourosh Tatar for their work and for making the beginning of my PhD studies a joyful and memorable time in my life. I would also like to express my gratitude to the rest of the people involved in LTU’s research project in this general area for interesting discussions, ideas and their interest: Tore Silver, Dr. Ales Svoboda, Prof. Inge Svennigson, Tommy Gunnarsson and Dr. Mikael Bäckström. Finally I would like to thank my current and previous colleagues at the Department of Sound and Vibration for sharing their knowledge and being great friends.

The work conducted was financed by the Swedish Agency for Innovation Systems (Vinnova). The funds to pay for the laser Doppler Vibrometer (LDV) used in this research came from the Kempe Foundation.
ABSTRACT

This thesis concerns the development of non-contact measurement methods and analysis of rotors. The methods have been verified and applied to milling machine spindles in order to investigate the speed dependency in the milling machine spindle dynamic. The research was financed by the Swedish Agency for Innovation Systems (Vinnova).

Turning operations like milling are common in the automotive and aerospace industries where large metal work pieces are reduced to a fraction of their original weight when creating complex thin structures. During these operations it is important that unwanted behaviours such as excessive tool vibrations be avoided (this is normally called “chatter”). Chatter causes poor surface finish and/or material damage and can expose machine operators to annoying and/or dangerous noise levels. In order to predict processes parameters for a chatter free milling operation, knowledge of the properties of the dynamic system are essential. Normally the system dynamics are measured during no rotation; in order to include the influence of the spin speed the system must be analysed for all spindle speeds intended for the milling operation. This can be done either by measurement or modelling. Non-contact measurement techniques are however, often based on displacement sensors which do not have the same sensitivity as velocity or acceleration based methods.

To improve the sensitivity in non-contact measurements of rotors a laser Doppler Vibrometry (LDV) based method has been developed. The developed LDV method is based on the reduction of the rotor surface structure and makes it possible to use single beam LDV measurements of rotors. These types of measurements were previously considered inaccurate but now have become feasible through the use of the method described in this research.

Furthermore the dynamic properties of a high-speed-milling machine spindle were studied by a contactless dynamic spindle tester (CDST) developed by SKF. The measurements were substantiated by simulations using a finite element model (FEM) which confirmed the measurement results. The CDST measurements could be performed without violating safety regulations regarding human interaction with high speed spindles through the use of a magnetic excitation method. In the measurements conducted by the CDST a speed dependency in the spindle dynamic could be detected. By performing FEM simulations the major source of this dependency could be identified. The centrifugal force of the balls in the angular contact ball bearings was shown to have the largest influence on the overall dynamics compared to the gyroscopic moment of the rotor. The study performed indicates that predictions of high-speed-milling stability must include consideration of the speed dependency in the dynamic.
CONTENTS

1 THESIS ........................................................................................................... 1

2 INTRODUCTION ............................................................................................... 5

3 LDV MEASUREMENT METHOD ....................................................................... 8

4 SPINDLE MEASUREMENTS .......................................................................... 16

5 MODELLING ................................................................................................... 22

6 DISCUSSION .................................................................................................. 28

7 CONCLUSIONS .............................................................................................. 32

8 FUTURE WORK ............................................................................................... 33

9 NOMENCLATURE ........................................................................................... 33

10 REFERENCES ................................................................................................. 34

APPENDIX A: SPINDLE DRAWING

APPENDED PAPERS

   PAPER I

   PAPER II

   PAPER III

   PAPER IV
1 THESIS

This thesis begins by introducing readers to the research problems of milling machine vibrations and spindle measurements. It then continues on by describing different non-contact measurement methods and, in particular, the theoretical background and problems associated with laser Doppler Vibrometry (LDV) when applied to rotating targets such as milling machine spindles. A method for solving these LDV problems is presented together with a method for milling machine spindle analysis using inductive displacement sensors and electromagnetic excitation substantiated by FEM simulations. Lastly, the findings are discussed along with suggestions to future work. This thesis covers work described in greater detail in the four attached papers.

1.1 Paper I

M. Rantatalo, P. Norman, K. Tatar, Non-contact measurements of tool vibrations in a milling machine, SVIB vibrations Nytt, 22 (2) (2004) 22-29

This paper presents a pre-study of two types of non-contact measurement methods for milling machine spindle vibrations. LDV on a rotating spindle and the use of an active magnetic bearing (AMB) for spindle dynamic measurements were tested. The AMB study was conducted on a low speed (0-7000 rpm) milling machine spindle. The methods and findings in this work were further studied in Papers II, III and IV.

Matti Rantatalo prepared the tool, outlined the work and performed the AMB measurements with the assistance of Peter Norman. The LDV measurements were performed by Kourosh Tatar and Matti Rantatalo and the milling machine was operated by Peter Norman. Matti Rantatalo performed the post processing and analysis of the data. Matti Rantatalo wrote the paper with the assistance of Peter Norman and presented the work at the SVIB 2004 Conference in Stockholm.

Published in SVIB’s member journal 2004-2. The paper was not subject to a review procedure.

1.2 Paper II


The paper describes a method for speckle noise removal in LDV on rotating targets. The study is an extension of the LDV study presented in Paper I. The title of the paper is a bit misleading in that a more general work is suggested than was actually
presented. A more representative title would be. “A method for speckle noise removal by surface structure reduction - an LDV application to rotating targets”.

Matti Rantatalo, Kourosh Tatar and Peter Norman outlined the work. Matti Rantatalo performed the post processing and analysis of the data with the assistance of Kourosh Tatar. Measurement data logged in Paper I was used in this paper. Matti Rantatalo wrote most of the paper with the assistance of Kourosh Tatar and Peter Norman. Correspondence with the editor of the conference proceedings and the reviewers was conducted by Matti Rantatalo.

Published in conference proceedings. The paper was subjected to a review procedure.

1.3 Paper III


The work includes an investigation in the presence of crosstalk between radial vibration components after using the method for speckle noise removal presented in Papers I and II.

Kourosh Tatar and Matti Rantatalo outlined the work with the assistance of Per Gren. Kourosh Tatar prepared and verified the dummy tool quality and performed together with Matti Rantatalo the experiments. Matti Rantatalo conducted the AMB measurements and Kourosh Tatar the LDV measurements. Kourosh Tatar performed the post processing of the data and together with Matti Rantatalo and Per Gren the data was analysed. Kourosh Tatar and Matti Rantatalo wrote most of the paper with the assistance of Per Gren. Kourosh Tatar was the corresponding author.

The paper has been accepted for publication by Mechanical Systems and Signal Processing.

1.4 Paper IV


This paper demonstrates a method for spindle analysis including FEM, contactless excitation and response measurement. The method was applied to a high-speed-milling machine with a spindle speed capacity of up to 24000 rpm. The experimental part in this paper, regarding the inductive measurement method, is a sequel to the study presented in Paper I where a low speed spindle was studied.
Matti Rantatalo outlined the work and performed the CDST measurements and tap tests with the assistance of Peter Norman. Matti Rantatalo performed the post processing and the analysis of data. Matti Rantatalo implemented a FEM representation of the rotor bearing system and performed the simulations with the assistance of Jan-Olov Aidanpää. Measurement of the physical spindle dimensions was performed by the SKF spindle service and Bo Göransson at SKF performed the bearing stiffness calculations and wrote parts about the physics behind the speed dependent bearings. Peter Norman outlined the pre-load test together with Bo Göransson and wrote about procedure and performed the measurements with the assistance of Matti Rantatalo. Matti Rantatalo performed the LDV measurements of the mode shapes. Matti Rantatalo wrote most of the paper with the assistance of Jan-Olov Aidanpää and Bo Göransson.

The paper has been submitted for publication.
2 INTRODUCTION

Turning operations like milling are common in the automotive and aerospace industries where large metal work pieces are reduced to a fraction of their original weight in the creation of complex thin structures. In Fig 1 a picture of the high speed (24000 rpm) ‘state-of-the-art’ 5-axes research milling machine used in this thesis can be seen. As with most operations, the development of more efficient processes results in increased productivity. This is the case in milling where the process can be made more efficient by optimizing performance in such a way that it better fulfils various demands such as flexibility, machine tool endurance, production time, operator work environment and production quality.

A property highly associated with these optimization parameters is machine tool vibration. Vibration can expose machine operators to annoying and/or dangerous noise levels. Regenerative machine tool vibration (chatter) will reduce the quality of a surface finish and can damage a workpiece (down to the molecular level). Chatter can also result in increased tool wear/failure and/or complex and costly machine tool failures. Therefore it is important that the vibration level be kept under control. Chatter is self-excited machine tool vibration which can be caused by various physical parameters [1]. Chatter caused by friction, thermodynamics or mode coupling in the cutting process is called primary chatter. Secondary chatter is caused
by the waviness of a machined surface and is considered to be one of the most important causes of instability in the cutting process.

The secondary chatter phenomenon is a significant issue and has been addressed and modelled by numerous authors over the past decades e.g. [2-5]. The developed models predict a specific chatter free depth of cut for a specific spindle speed. The predictions are based on the spindle speed and the frequency response function (FRF) of the tool tip; assuming a rigid workpiece. The chatter free depth of cut is calculated for different spindle speeds which can be plotted as stability lobe charts. Fig 2 shows a typical stability lobe chart where the area above the curve represents production parameters which will result in an instable process. The area below represents a stable process. Increasing the depth of cut to the wrong side of the curve will generate chatter.

![Stability lobes](image)

**Fig 2.** Stability lobes: The area above the stability lobe curve represents a depth of cut which will generate chatter vibrations. Values below the curve will render in a stable milling process.

### 2.1 Research problems

The FRF of the tool tip, which is used in the stability lobe calculations, is normally measured manually by tap-tests of a non-rotating (rpm=0) spindle/holder/tool system where the tool tip is excited by an impulse hammer and the response is measured by a vibration transducer. Stability lobes based on FRF measurements made at 0 rpm are calculated for all spindle speeds. The assumption in this procedure is that the dynamics of the spindle/holder/tool system is independent of the spin speed. This is
however not true for the case of high-speed-milling operations where the effect of gyroscopic moments and centrifugal forces will change the FRF. [6-8]. The value of the depth of cut, in the 0 rpm based stability lobes, is only valid for the speed 0 rpm and not for other spindle speeds unless the FRF is speed independent. In order to achieve spindle speed dependent stability lobes the FRF’s for each spindle speed must be inserted into the calculations. Tap-tests to measure the high speed FRF would add an extra risk to work environments of the operator or engineers who conducts such measurements. An alternative, safer excitation method for spindle analysis is therefore desirable. Knowledge of the origin of the speed dependency in the system dynamics can also give valuable information for spindle design.

The physical properties of a fully operating milling machine can either be analysed by various measurement methods or by modelling. To accurately model spindle dynamics speed dependency the models must include all parts which can contribute significantly. Present measurement methods use displacement based response sensors and various types of excitation methods which are all in some way in contact with the rotor during the measurement procedure. Displacement sensors have a lower sensitivity to higher vibration frequencies than acceleration and velocity based methods. No substantial research using displacement sensors for measurement of high frequencies was found during a survey of major databases. As an alternative, a velocity based measurement method would increase the sensitivity to higher frequencies. LDV is such a method and it is already a common instrument for vibration measurement. However, the method possesses limitations when applied to rotating targets because speckle noise and crosstalk between radial vibration components occurs. A method for LDV measurement of rotating rotors has been developed [9] but is based on a multiple set of continuously measuring LDV’s. The possibility of using a single beam laser would reduce the investment in measurement equipment and make it possible to measure spinning rotors in, for example, situations where measurement from two directions is not practical.

2.2 Research questions
In the previous section the research problems, which are the subject of this thesis, were outlined. When analysing the presented research problem the following research questions emerged:

- Is it possible to measure radial vibrations of a spinning rotor using a single beam LDV?
- How can a spindle be measured without violating safety regulations regarding human interaction with high-speed-milling machines?
- What contributes the most to the speed dependent spindle dynamic?
3 LDV MEASUREMENT METHOD

Vibration measurements of spinning rotors can be carried out using inductive/capacitive displacement sensors or laser sensors based on triangulation. These methods are however, limited regarding detectable frequency and amplitude and must be positioned relatively close to the measurement surface. Laser Doppler Vibrometry (LDV) offers a more sensitive velocity based vibration measurement technique than other non-contact measurements methods.

LDV is now a commonly used method for vibration measurements. This technique has many advantages compared to traditional vibration measurements. For example, it is easier to use than accelerometers plus it is often faster to use. Fig 3 shows an example of LDV measurements of four objects of different size and shape. The example shows a scan of the supporting frame of a scooter revealing an eigen-mode at 64 Hz located at the belt tunnel. The other scans show a racket, a speaker membrane and a small computer component which demonstrates measurement of small, thin and lightweight structures. The nature of the LDV system permits measurement without additional mass loading and allows a wide range of distances between the sensor head and the object (from millimetres up to several meters). Vibration measurement of hot objects can be performed as well as measurements of small and lightweight structures as e.g. the tympanic membrane in a human ear [10].

The LDV technology can also be applied in other medical applications e.g. for teeth vibration measurements during drilling [11]. Another application of laser Doppler Vibrometry is the measurement of sound wave propagation in transparent medias like gases [12, 13]. When a sound wave is propagating through a medium it changes the pressure. The changing pressure will affect the refraction index of the medium and hence modulate the laser frequency. This modulation is then interpreted as a vibration velocity by the LDV system.

3.1 LDV principal

A Polytech\textsuperscript{1} PSV 300 scanning LDV system was used in this thesis. The LDV was equipped with a scanning laser head with a scanning angle of about ±20°. The LDV used for this work has a helium neon (He-Ne) laser with a wavelength of 632.8 nm. The LDV measures the velocity component of the object along the direction of the laser beam and is based on the detection of the Doppler shift in the laser light reflected from the surface.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{fig4.png}
\caption{Sketch of an LDV. In an LDV the emitted light with the frequency $\Omega_i + \omega_b$ is allowed to interfere with the light reflected from the measured object.}
\end{figure}

\textsuperscript{1} Polytec GmbH www.polytec.com
To determine the target’s velocity the laser light is demodulated by an interferometer where the reflected light is allowed to interfere with the original transmitted light (reference light) on a photodetector. To be able to determine the direction of the target velocity a virtual constant velocity is added by a Bragg-cell which frequency modulates the reference light with a 40 MHz signal ($\omega_B$, Bragg-cell frequency), see Fig 4. The registered Doppler shift is then used to calculate the velocity of the object by using the following equation;

$$v = f_D \frac{\lambda}{2}$$  \hspace{1cm} (1)

where $f_D$ is the Doppler frequency and $\lambda$ is the laser wavelength.

### 3.2 Speckle

When illuminating an optically rough surface with a laser, a phenomenon called speckle will occur. Speckles are bright and dark spots that can be seen in a reflected laser light (see Fig 5). These spots are the result of superimposing light wavelets which are caused by differences in travelled path length, from light source to detector, due to the rough surface. A surface is considered optically rough if the surface structure exceeds the level of approximately half the wavelength of the light source, in this case 316 nm. The speckle phenomena is sometimes considered a disturbing noise but in other cases, like in TV holography [14], the speckles are used as the information carrier.

![Fig 5. Speckles on a detector. Speckles are formed at a detector due to superimposing reflected light wavelets with different travelled path lengths.](image-url)
For a non-moving target, the summarized wavelets are only seen as an added DC level to the detector signal. If the target starts to move, this DC level will start to vary according to the speckle population present on the detector. This results in speckle noise. The noise will modulate the phase of the laser light and add a noise to the detected Doppler signal. This noise level can be reduced by averaging due to its random behaviour.

### 3.3 Speckle noise on rotating targets

When measuring multiple revolutions of a rotor using LDV, the speckle noise cannot be averaged out from the signal and must be considered separately. In a static LDV measurement of a rotor, the same speckle noise occurs each revolution. If the measurement spans a number of rotor revolutions, the speckle noise will be apparent as peaks in the frequency domain at multiples of the rotational speed. If the rotor speed is constant and the measurement sequence contains many revolutions, the speckle peaks can easily be distinguished from ordinary structural vibrations due to their narrow appearance. The speckle noise has a similar appearance to ordinary out-of-roundness components which also are located in the frequency spectra at multiples of the rotational speed. The main difference between speckle peaks in the frequency domain and typical peaks originating from the out-of-roundness components is the decaying appearance with increasing frequency that the out-of-roundness components show (see Fig 6).

![Fig 6. Spectral components from LDV measurements. A: Structural vibrations. B: Rotor centre miss alignment (n=1) roundness components (n=2, n=3,…). C: Repeated speckle noise. D: n=1 illustrates the rotor miss alignment and n=2, n=3, ... illustrate the out-of-roundness components of the rotor surface structure.](image)
3.4 Speckle noise removal

As described here, it has been shown by others that the speckle noise level in LDV can be reduced or removed by different methods. By optimizing the detector size and position within a vibrometer the noise level can be reduced but not completely removed [15]. It has also been shown that the speckle noise in laser torsional vibrometry measurements can be removed by randomising the path that the laser light is undertaking during the revolutions. This is by either moving the laser along the shaft [16] or simply by adding a new surface structure. The latter strategy can be achieved by continuously applying oil or some other substances to the surface during measurement [17]. In theory, these techniques should also work for LDV measurements. Another approach to speckle noise removal is by reducing the target surface structure to below half the laser wavelength [18, 19] (Papers I and II). Fig 7 shows a comparison between LDV measurements performed on a rotor with a rough (lower graph) and a smooth surface structure (upper graph). The lower graph shows speckle peaks at multiple integers of the rotational speed (100 Hz). The upper graph shows a spectra with no peaks due to speckle. The out-of-roundness components in the upper left graph could be extracted with no disturbing speckle noise and the result was verified by a mechanical roundness tester (see Fig 8).

Fig 7. Spectra of the polished and the rough surface at a spindle speed of 6000 rpm. The spectrum of the rough rotor has been mirrored along the frequency axis down to the negative side to simplify comparison between the two. Multiple harmonics of n*100 Hz (marked by black dots) can be seen in the spectrum of the sprayed surface (n = 1,2,3…).
3.5 Crosstalk

When applying LDV to rotating optically rough targets two problems occur; the presence of speckle noise which has been described in the previous section and crosstalk between vibration velocity components. The crosstalk problem is caused by different velocity components which affect the Doppler shift in the reflected light. In this thesis (from Paper III) the crosstalk between the two radial vibration components are studied. Due to an optically rough surface, backscattered light from the tangential velocity component modulated by the in-plane displacement will be added to the out-of-plane velocity. [9, 20-22]. The measured velocities of a rotating optically rough shaft in the two orthogonal directions, $v_x, v_y$, can be expressed as [9]:

$$v_y = \dot{y} + \Omega(x - x_0)$$  \hspace{1cm} (2)

and

$$v_x = \dot{x} - \Omega(y - y_0),$$ \hspace{1cm} (3)

where $\dot{x}$ and $\dot{y}$ are vibration velocities, $x$ and $y$ are the vibration displacements, $x_0$ and $y_0$ are the distances to the spin axis due to alignment errors and $\Omega$ is the total angular velocity including torsional vibrations. $\dot{x}$ and $\dot{y}$ are the desired velocity components for each direction. The methods for speckle noise reduction/removal
described by previous authors cope with the specific speckle noise problem but are not able to neutralize the effect of crosstalk in a single beam LDV measurement. Consequently; the signal obtained during measurements under these circumstances will be a mix of the vibration components in both directions. A method for resolving the true vibrations in the two x- and y-directions using a setup of two simultaneously measuring lasers in both directions and an accurate measurement of the rotational angular velocity has been developed by Halkon and Rothberg [9]. The method does require a setup of two simultaneously measuring LDV systems in an orthogonal arrangement.

In Paper III the cross sensitivity in the developed method presented in Papers I and II is investigate experimentally. The crosstalk in LDV applied to a polished rotor is compared with the crosstalk present when measuring an optically rough rotor. As a reference, a set of inductive displacement sensors (DS) measured the position of the rotor in the x and y-direction (see Fig 9). The excitation of the rotor was carried out by electromagnets.

![Diagram](image)

**Fig 9.** Crosstalk test. Sketch of setup used to examine the crosstalk in LDV on rotating targets.
Simulated crosstalk based on values from the reference sensors $y$ and $x$ (DS) LDV on a rough surface LDV on a smooth surface Displacement sensors ($y$) for a smooth and rough surface

Fig 10. Effect of crosstalk in LDV: Velocities at 400 Hz for different spindle speeds 700, 1400, 2800, 5600 and 7000 rpm.

Fig 10 illustrates the effect of crosstalk on LDV measurements of a rotating rough surface for different spindle speeds and an excitation of 400 Hz orthogonal to the laser beam. The measured vibration velocity of the dummy tool after being sprayed with paint (triangle up) shows a spindle speed dependent crosstalk as expected from Eq. (2 and 3), while the same measurements on the smooth surface (triangles down) do not. The outputs from the displacement sensor (DS) in the $y$-direction for both sets of measurements (smooth and rough measurement surfaces - square and pentagram) showed no physical crosstalk during the excitation. Inserting the signals from the displacement sensors $y$ and $x$ into Eq. (2) gave the same velocity (circles) as from the vibrometer when the surface was rough (triangle up). This calculation confirms the measured crosstalk detected by the LDV when measuring the rough surface.
4 SPINDLE MEASUREMENTS

During a milling process the cutting force and the frequency response function (FRF) at the tool tip are two important parameters. The cutting force can be measured indirectly by the use of a force plate. The force plate measures the force on the workpiece by measuring the force transferred by the workpiece to the machine table. This procedure assumes that the workpiece is rigid with no interfering modal properties.

An alternate approach is to measure the force that the workpiece has on the tool. In order to do this, non-contact sensors measuring the rotor vibrations are used. Non-contact measurement of rotating milling machine spindles customarily use inductive and capacitive displacement sensors together with laser based displacements probes [1]. Albrecht et al. [23] describe an indirect method of force measurement when milling that uses capacitive or inductive displacement sensors. In this method, the transfer function between the force applied on the tool tip and the displacement from capacitive sensors mounted on the spindle close to the housing is measured for different spindle speeds. Different Kalman filters were then calculated for each speed and applied to the displacement sensor signal to produce real time cutting force measurement data. Tap tests on a ball bearing mounted on the tool tip were used to excite the structure (see Fig 11A). Spiewak [24] presented an alternative accelerometer based cutting force measurement method where a milling cutter was...

---

2 www.kistler.com
instrumented with a 3-axial accelerometer inside the tool close to the tip. This
method requires specially manufactured tools.

For spindles equipped with active magnetic bearings (AMB), Auchet et al. [25] have
outlined a another method for indirect cutting force measurement based on command
voltage of the AMB. The method used the relationship between cutting force and an
increasing command voltage in the magnetic bearings in order to keep the rotor in
place (see Fig 11B). The relationship was established by measuring the FRF between
a force (tap test) applied to the tool tip and the command voltage of the AMB. The
FRF measured at 0 rpm as a predictor of the cutting force at high speed machining
will, in the view of this author, be a source of error. Using AMB’s for measurement
and chatter control purposes has been investigated. Knospe [26] looked at active
chatter suppression through the use of AMB. Chen and Knospe [27] estimated
cutting dynamics by both exciting the system and increasing the damping of the lathe
tool using an AMB. Similar to exploiting the relationship between the AMB
command voltage and the cutting force, methods which use the current in motorised
spindles have been studied. For example, Jeong and Cho [28] developed a method
where they improved the frequency range from earlier methods by a factor of two up
to 130 Hz.

4.1 Speed dependent stability lobes

Knowledge about the tool tip FRF is important when calculating the stability lobes
and optimizing the maximum depth of cut. When dealing with rotating dynamic
systems like a milling machine spindle, especially during high speed machining, the
FRF depends on the spindle speed. To achieve the spindle speed dependent FRF, the
machine tool must be analysed in a rotating state that spans the whole range of
spindle speeds intended for the operation. An experimental method for the prediction
of stable cutting regions was presented by Schmitz et al. [29] which took into
account the dynamic change that a rotating spindle undergoes. The method is based
on impulse hammer excitation and capacitive probe response measurement of a tool
rotating during different spindle speeds (see Fig 11C). Stability lobes for a discrete
number of spindle speeds are calculated and the limit value corresponding to the
actual spindle speed used during the measurement is picked out to form a spindle
speed dependent stability lobe chart. Experimental tests found that there is a
changing stable limit of cut above 16000 rpm. Sims et al. [30] demonstrated a
method for tool tip FRF prediction based on piezoelectric actuators and sensors
mounted near the base of the tool (see Fig 11D). The predicted FRF was compared
with ordinary impulse hammer tap tests at the tool tip.

4.2 Spindle measurements

For this thesis, a contactless dynamic spindle testing (CDST) instrument for
measuring the speed dependency was used. The CDST uses inductive response
measurement and electromagnetic excitation of the tip of a dummy tool (Papers I and
IV) for spindle testing. The tool tip FRF can be measured in the radial directions x and y without violating safety regulations regarding human interaction with high speed rotating spindles.

Fig 12. Setup of the CDST measurement performed on the high-speed-milling machine.

The rotor excitation was caused by electromagnets which were fed by a frequency step vice sine sweep coil current which generated a magnetic force that acted on the rotor. The rotor consisted of a specially manufactured dummy tool with a laminated rotor part designed to reduce the energy losses caused by eddy current effects. In each radial direction two electromagnets (on opposite sides of the rotor) worked together in attracting the rotor to cause the excitation. Two types of milling machine spindles were tested using this measurement method. Fig 12 shows the measurement setup of the CDST measurement of the high speed spindle. During the measurement procedure the spindle was lowered, thereby inserting the dummy tool into the CDST measurement unit.

4.3 Low speed spindle testing
In an initial study presented in Paper I a 3-axes Dynamite milling machine with a spindle speed capacity up to 7000 rpm was measured (see Fig 13).
Fig 13. Measured FRF’s for different spindle speeds. Bright colours correspond to high magnitudes. Arrows point out detected structural modes and the rotor unbalance.

In Fig 13 the influence of the rotor unbalance can be seen in the upper left corner of the figure. Three structural modes of the rotating spindle can be seen in the figure between 200 and 600 Hz. These modes do not seem to possess any visible speed dependency except for the step between 0 and 350 rpm. This change could be due to the different dynamic properties of non-rotating and rotating bearings.

4.4 High speed spindle testing

In a more extensive study presented in Paper IV, a Liechti Turbomill ST1200 ‘state-of-the-art’ machining centre (capable of multiple movement – up to 5-axes) equipped with a Fischer spindle (MFWS-2305/24/8) capable of speeds of up to 24000 rpm was studied. The FRF measured in this study were recalculated from [m/A] to [m/F] and the measurements were performed in the intervals 0, 2000, 4000,…,24000 rpm. The detected eigen-modes in the measurements showed a clear speed dependency especially above 10000 rpm (see Fig 13).
4.5 Stability lobes for different spindle speeds

By using the speed dependent FRFs, stability lobes could be calculated for each spindle speed. When applying the method presented by [29] the depth of cut of a certain spindle speed can be picked out. Fig 15 shows stability lobe calculations based on different FRFs measured during different spindle speeds. The thick red curve represents the stability limit calculated when based on the FRF measured at 0 rpm. Black thin curves are based on FRFs measured during speeds in the intervals 2000, 2000, 4000,…,24000 rpm. Green dots mark the speed dependent depth of cut for the speeds 10000, 12000, 14000,…,24000 rpm which were picked out from each curve. For milling, based on 0 rpm stability load predictions, green dots located above the red curve indicate that the depth of cut could be increased for that speed while dots below indicate that chatter vibrations will occur at that speed. The example in Fig 15 illustrates the importance of investigating the speed dependency in FRFs when performing predictions of high-speed-milling machine stability. Correct predictions can mean avoidance of chatter vibrations or identification of speeds where the depth of cut can be increased.

3 The stability lobes were calculated by Sandvik Coromant using CutPro.
Fig 15. Stability lobes calculated for different spindle speeds. The red curve is based on 0 rpm FRF. Black curves are based on FRFs measured at the speed intervals of 2000, 4000, 6000, …, 24000. Green dots represent the speed dependent depth of cut picked out from stability lobes based on FRFs measured at that specific speed.
5 MODELLING

The speed dependency shown in the high speed spindle measurements will alter the stability lobe predictor criterion. Paper IV describes investigations into the spindle speed dependency described in the previous section through the use of numerical simulations. The simulation examined the influence of the gyroscopic moment of the rotor and the centrifugal effects in the ball bearings on the eigen-frequencies of the spindle.

In their analysis of a milling machine spindle, Wang and Chang [31] presented a modelling method based on FEM. The model did not include rotation and therefore centrifugal forces and gyroscopic moments were not considered. In 1976 Nelson and McVaugh [32] presented a FEM formulation of a rotor bearing system based on the Euler Bernoulli beam theory where the effect of gyroscopic moments and centrifugal forces is included. Zorzi and Nelson [33] later added internal damping and in 1980 Nelson [34] presented another formulation based on the Timoshenko beam theory which factored in shear deformation effects. Xiong et al. [7] presented a way of combining this FEM representation and the milling cutting force model formulated by Altintas [3]. The model, which only consisted of the rotor, predicted that the gyroscopic moment would not affect the stability regions in milling but would increase the real part of the eigen-frequency and thus reduce the axial depth of cut. The model also predicted a change in spindle resonance frequencies of about ±10 Hz. Chi-Wei Lin et al. [35] integrated a thermo-mechanical-model into the Timoshenko FEM description. Numerical and practical experiments verified an increase in bearing stiffness with increasing bearing preload. The work also predicted a softening of the spindle shaft with increasing spindle speed. It was suggested that the softening of the bearing radial stiffness due to speed could be compensated for by the thermally-induced preload. Cao and Altintas [8] presented a general method for the modelling of a spindle bearing system which included the axial coordinate plus a corresponding spindle speed and preload dependent five degree of freedom bearing stiffness matrix. In the spindle model a rotor related centrifugal force was modelled by subtracting a \( \Omega^2 \) term\(^4 \) from the stiffness matrix.

The numerical simulations presented in Paper IV were based on the FEM formulation described by Nelson [34] and were extended by including a separate mass and stiffness radius together with a stiffness radius dependent shear deformation factor [36]. A second order homogenous differential equation was used to describe the dynamical system;

\[
[M]\ddot{y} + [D]\dot{y} + [K]y = [0]
\]

\(4\)

\(^4\) \(\Omega^2\) multiplied by a radial version of the translational mass matrix.
where \([M], [G],\) and \([K]\) are the system matrixes of a shaft element (see the Appendix in Paper IV). \(q\) is the generalized coordinates of the FEM assembly. The assembled second order homogenous differential equation was transformed into a first order differential equation, using the state vector notation described in [37]. The equation of motion could then be rewritten as:

\[
\begin{bmatrix}
\Omega & [M] \\
[-M] & [0]
\end{bmatrix}\begin{bmatrix}
\dot{p} \\
[0]
\end{bmatrix}+
\begin{bmatrix}
[K] \\
[0]
\end{bmatrix}\begin{bmatrix}
0 \\
[M]
\end{bmatrix}\begin{bmatrix}
\dot{h} \\
[0]
\end{bmatrix}=egin{bmatrix}
0 \\
0
\end{bmatrix},
\] (5)

where

\[
\begin{bmatrix}
\dot{p} \\
\dot{q}
\end{bmatrix}=egin{bmatrix}
0 \\
0
\end{bmatrix},
\] (6)

5.1 Spindle model

In the simulation of the milling machine spindle a reduced model without the machine foundation and spindle housing was used. The model was based on the drawings shown in Appendix A. The model consisted of the rotor and a pair of supporting ball bearings.

![Spindle model including the rotor and the bearings.](image)

The rotor included the spindle shaft, motor package, inner bearing rings, tool holder and dummy tool. The rotor was divided into FEM elements and speed dependent
bearing stiffness was added to nodes corresponding to the bearing positions. Fig 16 shows the rotor and the bearings modelled as springs. The figure is a simplified illustration of modelled bearing stiffness. Apart from the illustrated radial stiffness of the bearings, an angular stiffness component and a cross coupling term are included in the model.

5.2 Preload measurement

To be able to calculate the bearing stiffness the preload force of the bearings needs to be known. However, when dealing with a real milling machine spindle the bearing preload is normally not known to the operator and must be measured. On-line measurements which give continuous information about the preload status would be preferable. Studies of on-line measurements and control of the bearing load have been described by Chen and Chen [38]. Most spindles now used for production work do not have and can not be retrofitted for on-line measuring. The preload must therefore be measured or be provided by the spindle designers. The spindle used in this study was designed to use spring-loaded bearings as shown in Fig 17. It consists of a rotor with two hybrid angle-contact bearings placed back to back. The preload is measured by pulling the spindle towards \(-z\) while the displacement is measured using a dial indicator. The force needed to unload the front bearing, hence the preload, was estimated as 1450N.

5.3 Bearing stiffness calculations

Using the result from the preload measurement the bearing stiffness within the speed interval 0, 2000, 4000, …, 24000 rpm were calculated. The bearing stiffness calculations were performed by SKF with their in-house developed software “Bearing Beacon”. The bearing stiffness for different spindle speeds in relation to the stiffness at 0 rpm are plotted in Fig 18. The plot shows that the stiffness in the radial direction of the back bearing decreased to a level of 62% of its original (0 rpm) value.
when the speed increased to 24000 rpm. The corresponding value for the front bearing was 38%. This bearing softening was caused by the centrifugal force which acted on the balls by displaced them axially and radially. In doing this, an extra spring is added in series with the normal Hertzian contact spring. The shape of the stiffness variation of the bearings can be seen in the eigen-frequency variation of the bearing related modes measured by the CDST (see Fig 14).

![Graph showing changes in bearing stiffness with increasing spindle speed.](image)

**Fig 18.** Changes in bearing stiffness in the x direction with increasing spindle speed. 100% represents the bearing stiffness at 0 rpm.

### 5.4 Simulations

The eigen-frequencies of the modelled spindle can be seen in Fig 19. In Graph A the gyroscopic effect of the rotor can be seen; each mode splits up into a backward and a forward mode. Graph B, shows the same simulation with speed dependent bearing stiffness. The speed dependency in the bearing stiffness originates from the centrifugal force that acts on the balls in the bearing. In this case, the effect of the centrifugal force acting on the balls in the bearings had a more significant influence on the eigen-frequencies than the gyroscopic moment. With the speed dependent bearing stiffness a reduction of the eigen-frequencies of the first and second modes of about 40% and 37% could be detected. In comparison, the reduction was approximately 1% when only the gyroscopic effect was present. When including the speed dependent bearing stiffness, the same shape - as seen in the bearing stiffness variation (Fig 18) - could be seen in the variation of the eigen-frequencies of bearing related modes.
Fig 19. Simulated eigen-frequency for different spindle speeds. A: Influence of the gyroscopic moment of the rotor. B: Influence of both the gyroscopic moment of the rotor and the centrifugal effect in the bearings.
5.5 Mode shape and eigen-frequency identification

The rigid body modes of the rotor are governed by the mass distribution and the bearing stiffness [39]. The rigid body mode governed by the front bearing has a node point at the back bearing position which causes a larger displacement at the front bearing position. This mode will form a conical shape and vice versa for the back bearing. The radial bearing stiffness mainly contributes to the rigid body modes while the stiffness of the rotor shaft dictates the flexural modes. In reality, the modes often appear as a mix of rigid body modes and flexural modes. The mode shape analysis of the simulation revealed that the first mode shape was governed by the back bearing and the second mode by the front bearing. The third mode was a pure flexural mode and the forth was a mix of a flexural mode and the front bearing. Due to the reduced bearing stiffness, the first and second mode shapes intersected (see Fig 19B) at approximately 14000 rpm. The simulated mode shapes were verified by an LDV line scan of the visible part of the rotor at 0 rpm (see Fig 20).

Fig 20. Mode shapes of the spindle at 0 rpm. Simulations compared to LDV measurements.
6 DISCUSSION

In this thesis a method for single beam LDV measurements on rotating spindles has been presented together with a spindle analysis method based on inductive displacement measurement, electromagnetic excitation and FEM. The LDV method removed the speckle noise and the cross sensitivity in the measurements and the miss alignment and out-of-roundness components could be extracted from the measurements signal. By applying this method, single beam LDV measurements can be conducted on rotating spindles. The second method for spindle analysis identified the bearing stiffness as the weakest link in the spindle design regarding the speed dependent spindle dynamics. A simplified model of the spindle system showed good agreement with the measurements.

6.1 LDV on rotating targets

The developed single beam LDV method for radial rotor vibration measurements requires that the measurement surface be polished. When using this method the measurement surface must be kept smooth and clean during the measurement procedure. When choosing material for the dummy tool, stainless steel is preferable. This type of material is non-corrosive and thus avoids surface problems which would lead to inaccurate measurement. The alignment of the laser is also a parameter that must be considered when using this method. The alignment was however, found to be an easy procedure. The biggest issue in alignment is that too large a displacement during the measurement could reflect the returning light away from the path of the emitted light. This effect was however, not detected in the measurements performed on the milling machines studied.

6.2 Modelling

Excluded parts: The model, used in Paper IV, is a reduced milling machine model. The machine foundation, spindle housing and parts belonging to the pre-load mechanism were not included. Consequently the bearing support was modelled as a rigid support. Asymmetry in the spindle housing or in the mounting of the spindle was not included in the model. Rotor unbalance was also not included in the simulation.

Coupling: The coupling between the holder and the spindle shaft and the coupling between the holder and the tool was not considered. The coupling was simply modelled as a rigid connection. This simplification resulted in a slightly stiffer spindle being simulated.

Drawbar and springs: Inside the spindle shaft, mechanical components like the drawbar and springs are mounted. These parts are used to pull the tool and connect it to the spindle shaft. The content of the hollow spindle shaft could not be determined
with a high degree of confidence and therefore the inner mass radius of the hollow spindle shaft was set to zero. The eigen-frequencies and gyroscopic moments could be affected by an inaccurate model of the mass distribution inside the spindle.

**Motor:** The part of the integrated motor which is mounted on the spindle shaft was modelled as a mass without any stiffness properties. In reality, it may be that the motor part adds some stiffness to the shaft; meaning that this assumption would result in a softer model rotor.

**FEM description:** A FEM formulation based on the Euler Bernoulli beam theory presented by Nelson and Mc Vaugh [32] was compared to the formulation based on the Timoshenko beam theory [34]. Eigen-values simulated by the use of the Timoshenko formulation are marked by circles and simulations made by the Euler Bernoulli formulations are marked by asterisks in Fig 21. The absence of the shear deformation in the Euler Bernoulli formulation means that a stiffer spindle with higher eigen-frequencies is depicted. The difference is greater when the frequency increases. Mode 1-6 showed a change in eigen-frequency at 0 rpm of about 4%, 3%, 4%, 4%, 8% and 13%. It can be argued that the impact of the difference between the two beam theories could be ignored and that the shear deformation is of little importance. In this case, the difference for the 1st and 6th mode corresponded to 27 Hz and 419 Hz respectively. The amount of difference is dependent on the actual geometry of the rotor and must be considered for each case. Damping, gravity and the centrifugal effects of the rotor described by [8] were not included in the model. The axial load during free run was considered negligible and therefore excluded from the simulation.

**Shear deformation factor:** The shear deformation factor included in the shear effect in the Timoshenko beam theory is normally determined experimentally and a typical value of this variable is 0.9 for a solid circular shaft. In the study presented in Paper IV an analytical method [36] to achieve the shear deformation factor of a circular hollow shaft was used. The use of the analytical value resulted in lower eigen-frequencies of the modes (see Fig 22). The difference between the two methods increased for higher frequencies but the influence of the shear deformation factor was not as evident as the influence of the actual shear deformation effect. Mode 1-6 shows a change in eigen-frequency at 0 rpm of about 1%, 1%, 1%, 1%, 2% and 3%. For modes 1 and 6 the percentage corresponds to 8 Hz and 105 Hz respectively.
Fig 21. Simulated eigen-values. The FEM formulations based on Euler Bernoulli (asterisks) and Timoshenko (circles) beam theory are compared.

Fig 22. Simulated eigen-frequencies regarding the shear deformation coefficient effect. The effect of the analytically obtained shear deformation coefficient of a hollow shaft (circles) compared to the solid circular shaft value 0.9 (asterisks)
6.3 Centrifugal effects of the shaft

According to Cao and Altintas [8] the eigen-frequency of a rotating spindle is affected by the centrifugal effects of the rotor which would then lead to a softening of the rotor shaft. This softening was introduced by a $-\Omega^2$ term added to the stiffness matrix. This term will indeed weaken the stiffness, if present. This term is however not included in the original FEM formulation presented by Nelson [34] in 1980. A comparison between two simulations with and without this term can be seen in Fig 23. Looking at Mode 3 the softening caused by the $-\Omega^2$ term reduces the eigen-frequencies which is not the case in the absence of that term. The softening can not be seen in the CDST measurements (see Fig 14) which supports the FEM formulation without the $-\Omega^2$ term in the stiffness matrix.

Fig 23. Comparison between FEM simulations. Circles denote simulated eigen-frequencies with no $-\Omega^2$ term in the stiffness matrix. Asterisks denote eigen-frequencies of a simulation with the $-\Omega^2$ term.
7 CONCLUSIONS

LDV method: A method for single beam LDV as presented in Papers I and II is based on the reduction of the rotor surface structure. The speckle noise in the measurement signal could be removed and the out-of-roundness components (form error) could be extracted. A crosstalk sensitivity study (Paper III) confirmed that the theory of crosstalk in LDV on rotors and verified this method’s ability to eliminate crosstalk from in-plane vibrations. The elimination also implies that the sensitivity to torsional vibrations is eliminated. The developed single beam LDV vibration measurement method makes it possible to carry out LDV measurement of spinning rotors without speckle noise and speed dependent crosstalk from in-plane vibrations. This method can, in contrast to other displacement-based methods, be used when a long distance between the measurement object and the sensors is necessary. The method can, as compared to other LDV methods, be applied where only one radial direction of the rotor can be accessed. The capacity of the LDV system in this method makes it possible to measure vibration of rotors up to frequencies of 1 MHz. The method requires that the measurement surface be polished to an optical smoothness; a level less than half the laser wavelength.

CDST and FEM analysis: The method presented in Paper IV describes a measurement method for rotors that functions in combination with speed dependent FEM simulations. By using the CDST measurement method, spindle analysis can be performed without any violation of applicable safety regulations regarding human interaction with rotating milling machine spindles. The centrifugal force of the ball bearings was shown to have a more significant effect than the gyroscopic moment of the rotor on the speed dependent dynamic of the studied milling machine spindle. It must however, be noted that this conclusion is based on data obtained from the specific spindle used in the study. Other results may be possible with other spindle designs. The studies reported on in this thesis do indicate that predictions of high-speed-milling stability based on a 0 rpm tap-test can be difficult due to the speed dependency of the system dynamics. A rotor softening reported by previous authors could not be detected in the CDST measurements. The measurement supports the simulation method chosen for Paper IV. This disagreement between the presented results in this thesis and the results presented by previous authors must be investigated further.

Research questions: According to the conclusions drawn, the research questions that initiated the presented studies could be answered. Radial vibrations of a spinning rotor can be measured using a single beam LDV setup. When modelling a milling machine spindle the suggested method in this thesis is based on the FEM description initially presented by Nelson [34] together with a spindle speed sensitive bearing stiffness, a separate mass and stiffness radius, and a stiffness radius sensitive shear deformation factor. The most significant part contributing to the speed dependency
of the spindle was found to be the bearing stiffness sensitivity to centrifugal forces acting on the balls in the bearings. To verify the simulated spindle behaviour a method for spindle analysis was used which included inductive displacement sensors and magnetic excitation. The electromagnetic excitation of the spindle simplified and removed hazardous elements in the measurement procedure normally performed by an impulse hammer.

8 FUTURE WORK

If only one radial direction of the rotor is accessible, the developed single beam LDV method is suitable. The development of a method for measuring the hidden radial direction is a possible avenue for future work. If such a method could be developed, both radial directions could be measured without moving the laser. An investigation into crosstalk caused by miss alignment and out-of-roundness components together with high frequency measurements of milling machines also suggests itself for future work. The absence of rotor softening in the measurements presented in Paper IV should be investigated further. Further studies can include the conversion of the CDST measurements from a dummy tool setup to a real cutting tool setup. The impact of the forward and the backward modes on the milling stability could also be investigated.

9 NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>$v$</td>
<td>Velocity</td>
<td></td>
</tr>
<tr>
<td>$\lambda$</td>
<td>Laser wavelength</td>
<td></td>
</tr>
<tr>
<td>$f_D$</td>
<td>Doppler frequency</td>
<td></td>
</tr>
<tr>
<td>$\omega_B$</td>
<td>Bragg-cell frequency</td>
<td></td>
</tr>
<tr>
<td>$\Omega$</td>
<td>Rotor spin speed</td>
<td></td>
</tr>
<tr>
<td>$[M]$</td>
<td>Mass matrix</td>
<td></td>
</tr>
<tr>
<td>$[K]$</td>
<td>Stiffness matrix</td>
<td></td>
</tr>
<tr>
<td>$[G]$</td>
<td>Gyroscopic matrix</td>
<td></td>
</tr>
<tr>
<td>$q$</td>
<td>Generalized coordinates</td>
<td></td>
</tr>
<tr>
<td>$h$</td>
<td>Transformed generalized coordinates</td>
<td></td>
</tr>
</tbody>
</table>
10 REFERENCES


APPENDIX A: SPINDLE DRAWING
Paper I
NON-CONTACT MEASUREMENTS OF TOOL VIBRATIONS IN A MILLING MACHINE

Matti Rantatalo1 Peter Norman2 Kourosh Tatar3

Luleå University of Technology  
1Div. of Sound and Vibrations  
2Div. of Manufacturing Systems Engineering  
3Div. of Experimental Mechanics  
Luleå  
Sweden

Abstract

Empirical knowledge of the machining process is now being complemented by more scientific methods to determine machine tool vibration properties that are associated with stable machining. Current methods used to predict the behaviour of a milling process (e.g. “impact-test”) use measurements made on non-rotating systems. However, such measurements do not detect the effects of gyroscopic and centrifugal forces present in a rotating system. To be able to fully investigate the behaviour of a high speed rotating system one need to use non-contact measurement methods e.g. the Laser Doppler Vibrometry (LDV) method, which is becoming common for vibration measurements; although the method is not without its problems when applied to rotating objects. The work presented here consists of an investigation into the use of LDV and inductive position sensors with magnetic excitation to measure vibrations of a tool in a milling machine. The ability to perform non intrusive measurements should make it possible to analyse changes in machine tool dynamics during the cutting process and in the long term find ways to stable machining.

1 INTRODUCTION

Manufacturers of modern machine tools are increasingly implementing advanced process monitoring and supervisory process control [1] to complement the basic functionality of the machine tool control system. At their simplest, process monitoring systems are used to help prevent or limit the effects of catastrophic events such as tool breakage [2] or spindle failure. Such events can be detected by monitoring the current drawn by axis drives and spindle motor [3], or by more advanced techniques such as
cutting force monitoring or measurements of vibrations using accelerometers or acoustic emission using sensitive transducers and signal conditioning software[4-8]. By setting safe limits for the monitored parameter(s) based on experience or trials, unusual or unexpected events which may indicate a catastrophic failure can be used as a trigger to stop the machine. To be able to fully investigate the behaviour of a high-speed rotating system, such as a machine tool spindle, it is necessary to use non-contact measurement methods. Several approaches to the non-contact measurement of rotating objects have been developed. These include optical techniques such as Pulsed Laser TV-Holography [9] and Laser Doppler Vibrometer techniques (LDV) [10] which all have their difficulties when applied to rotating objects. [11] Another way of measuring vibrations is with the use of inductive position sensors which is commonly used for displacement measurement. E.g. in SKF’s (www.skf.se) magnetic bearing product; position sensors are used in order to keep track of the axle position while it is being controlled by magnets keeping it hovering inside the bearing free of mechanical contact. This paper describes an investigation in the possibilities of measuring vibrations and Frequency Response Functions (FRF) of a tool in a milling machine using LDV or inductive position sensors. The two methods have been tested separately and no comparison between the two has been made.

2 LDV MEASUREMENTS OF A POLISHED DUMMY TOOL

A dummy tool with a radius of 10 mm and a length of 100 mm was manufactured from a solid stainless steel tool blank. The shaft was mounted in a lathe and polished using emery paper, diamond paste with particles ranging from 9 µm to 0.25 µm and with a chemical polishing fluid. In the actual experiment a spray which is normally used for crack detection was used to create a removable diffuse (optically rough) surface on the polished dummy tool. Both the polished tool surface and the sprayed surface were measured by the optical profiler, and a representative area of the tool (of 304 x 199 µm) was sampled in steps of 414 nm. The measurements showed a normally distributed surface structure with Ra = 11.29 nm, implying that the polished surface was optically smooth compared to the laser wavelength of 633 nm. The sprayed surface showed substantially higher values, Ra = 21.02µm, giving an optically raw surface.

For the vibration measurements, a PSV 300 LDV system from Polytec GmbH (www.polytec.com) including a displacement decoder was used. The LDV scanning head was mounted on a sturdy tripod and placed approximately 2 m from the tip of the dummy tool on a soft damped material to reduce the influence of structural floor vibrations. The maximum detectable frequency was set to 16 kHz.

The LDV was used to measure the vibrations at the tip of the polished dummy tool in the radial direction at a spindle speed of 6000 rpm. After this the same measurement was carried out on the tool after being sprayed to give an optically raw finish.
3 FRF MEASUREMENTS USING INDUCTIVE SENSORS AND MAGNETIC EXCITATION

FRF measurements using inductive sensors and magnetic excitation were carried out by the use of a standard magnetic bearing manufactured by SKF Revolve and a specially made dummy tool with magnetic properties. (The inductive sensors in the bearing have 1% linearity, a sensitivity of 50µm/V, a range of 1.5 mm, a resolution of 30 nm and are immune to shifts in temperature). The bearing was mounted in a milling machine on the machine table with its W axis along the y direction of the machine (Figure 1). A modal test of the mounted bearing was performed which revealed no significant modal properties in the frequency band of interest, implying that the FRF measurements of the spindle could be performed without unwanted influences from the mounted bearing structure. A dummy tool with magnetic properties was mounted in the machine spindle and lowered down into the bearing. The FRF of the mounted dummy tool was then calculated by measuring the response from the inductive position sensors while exciting the structure with the magnets. The magnetic bearing control software MBscope was set up to perform an analysis of the system in the interval 50-1000 Hz with 600 points/decade and 30 s/decade using a sine sweep with an amplitude of 1A. Measurements were made while exciting the structure in the W13 direction for 21 different spindle speeds from 0 to 7000 rpm in steps of 350 rpm. The measurements started with 0 rpm and with a machine structure temperature of approximately 18º C.

Figure 1 Milling machine with a magnetic bearing mounted on the machine table for spindle FRF measurements. The bearing W direction is along the normal y direction for the milling machine.
4 RESULTS

4.1 LDV

The results from the LDV measurements at 6000 rpm are presented in a velocity spectrum chart where the polished and rough dummy tool measurements are displayed together Figure 2. The spectrum of the rough surface has been flipped down to the negative side in the chart to simplify comparison of the two spectra. In the charts it can be seen that the spectrum of the rough dummy tool, in opposite to the polished one, contains peaks at $f \times n \text{ Hz}$ where $f$ is the rotational speed of 100 Hz (6000 rpm) and $n = 1, 2, 3 \ldots$. These peaks are expected due to the presence of a speckle noise repeated for each dummy tool revolution in the sampled data.

![Figure 2 Spectra of the polished and the sprayed surface at a spindle speed of 6000 rpm. The spectrum of the rough dummy tool has been mirrored along the frequency axis down to the negative side to simplify comparison between the two. Multiple harmonics of $n\times100$ Hz where $n=1,2,3\ldots$ can be seen in the spectrum of the sprayed surface.](image-url)
5.2 Inductive sensors

The results from the FRF measurements with the magnetic bearing are presented in Figure 3 and Figure 4. Figure 3 displays the magnitude response and the phase response for a non rotating spindle. Figure 4 shows the measured magnitude response functions for all spindle speeds where the response functions are displayed along the x axis from 50-1000 Hz. (High response values are represented by bright pixels in the image). A resonance frequency can be seen at 500 Hz in Figure 3 and Figure 4 for the non rotating case. The difference between the FRF of the non rotating spindle and the rotating ones is obvious especially regarding the resonance frequency at 500 Hz. On the upper left hand side in Figure 4 the spindle speed can be seen as a disturbance in the measurement.

Figure 3  Frequency response function for 0 rpm.
DISCUSSION AND CONCLUDING REMARKS

By polishing the tool to a level where the surface could be considered as optically smooth; the speckles noise is removed and the measurements of axial vibrations can be performed. If this method is to be used for vibration measurements of a rotating tool when it is cutting, several other difficulties must be tackled. The laser beam must have a clear path to the tool with no interfering metal chip, oil or cooling fluid from the cutting process. The polished part of the tool surface, where the measurement will be conducted, has to be free of particles such as dust or process fluids. Any disturbances on the surface will cause speckle noise and unwanted peaks in the spectrum.

The test with the magnetic bearing showed that it is possible to measure the FRF during different spindle speeds with a magnetic bearing, and therefore making it possible to detect e.g. speed dependent components in the FRF. The difference detected in the measurements of the non rotating spindle and the rotating ones can have its origin in temperature or speed dependent components in the spindle structure. More tests will be conducted with other test signals and during other machine conditions. The influence of the unknown excitation force from the machine itself must also be considered in future measurements. Comparisons between LDV and inductive position sensors will also be carried out.
6 ACKNOWLEDGEMENTS

The financial and technical support for this work was provided by the Swedish Agency of Innovation Systems (Vinnova) and SKF Nova, respectively. The LDV system and auxiliary equipment have been financed by the Kempe Foundations.

7 REFERENCES


Paper II
Laser doppler vibrometry measurements of a rotating milling machine spindle

Matti Rantatalo
Kourosh Tatar
Peter Norman
Luleå University of Technology
1Div. of Sound and Vibrations
2Div. of Experimental Mechanics
3Div. of Manufacturing Systems Engineering
Luleå
Sweden

ABSTRACT
Finding an optimum process window to avoid vibrations during machining is of great importance; especially when manufacturing parts with high accuracy and/or high productivity demands. In order to make more accurate predictions of the dynamic modal properties of a machining system in use, a non-contact method of measuring vibrations in the rotating spindle is required. Laser Doppler Vibrometry (LDV) is a non-contact method, which is commonly used for vibration measurements. The work presented consists of an investigation into the use of LDV to measure vibrations of a rotating tool in a milling machine, and the effects of speckle noise on measurement quality. The work demonstrates how the axial misalignment and the roundness of a polished shaft can be evaluated from LDV measurements.

1 INTRODUCTION
Manufacturers of modern machine tools are increasingly implementing advanced process monitoring and supervisory process control (1) to complement the basic functionality of the machine tool control system. At there simplest, process monitoring systems are used to help prevent or limit the effects of catastrophic events such as tool breakage (2) or spindle failure. Such events can be detected by monitoring the current drawn by axis drives and spindle motor (3, 4), or by more advanced techniques such as cutting force monitoring or measurements of vibrations using accelerometers or acoustic emission using sensitive transducers and signal conditioning software (5-9). By setting safe limits for the monitored parameter(s) based on experience or trials, unusual or unexpected events which may indicate a catastrophic failure can be used as a trigger to stop the machine.
Since vibrations are the result of relative movement between the cutter and work piece, the dynamic behaviour of both the machine structure and rotating spindle/cutter together with the behaviour of the component being machined has to be considered. In most situations, the work piece can be considered a solid part fixed to the machine table with no significant modal properties of its own. This assumption tends to weaken, however, when machining components with relatively thin walls (10).

Regenerative machine tool chatter is a fundamental type of vibration that can occur during milling. These vibrations have their origin in the closed loop nature of the cutting process and are dependent on the structural vibration modes, described by the frequency response function (FRF) of the machine tool. The FRF is normally measured on a non rotating/static system from which the limits for chatter free machining can be calculated (11). In modern machine tools, spindle speeds of 20,000 rpm and upwards are not uncommon, since the dynamic characteristics of the spindle such as damping change, this causes the FRF of the system as a whole to change.

To be able to fully investigate the behaviour of a high-speed rotating system, such as a machine tool spindle, it is necessary to use non-contact measurement methods. Several approaches to the non-contact measurement of rotating objects have been developed. These include optical techniques such as Pulsed Laser TV-Holography (12) and Laser Doppler Vibrometer techniques (LDV) (13).

LDV is a well-established technique for measuring the velocity of a moving object. It is based on the Doppler effect, which explains the fact that light changes its frequency when detected by a stationary observer after being reflected from a moving object. The vibrating object scatters or reflects light from the laser beam and the Doppler frequency shift is used to measure the component of velocity which lies along the axis of the laser beam. As the laser light has a very high frequency, direct demodulation of the light is not possible and optical interferometry is therefore used.

When a coherent light source illuminates a surface that is optically rough, i.e. the surface roughness is large on the scale of the laser wavelength, a granular pattern called speckle which has random amplitude and phase is seen. This is due to interference between the components of backscattered light. The intensity of a speckle pattern obeys negative exponential statistics and their phases are uniformly distributed over all values between -π and π (14). If the speckle pattern changes during LDV measurement the rate of change in the resulting phase will be nonzero, and the frequency spectrum will contain peaks. These kinds of speckle fluctuations are induced by non-normal target motions, such as tilt, in-plane motions or rotation (15). Speckle fluctuations due to target rotation are periodic and will repeat for each revolution. This leads to peaks in the spectrum at the fundamental rotation frequency and higher order harmonics. These modulations are difficult to distinguish from the true vibrations and in the worst case, can almost completely mask the vibration pattern. It is therefore important that the target to be measured has a surface smooth enough so that the speckle noise is avoided.
2 EXPERIMENTAL SET-UP AND PROCEDURE

2.1 Preparation of the dummy tool
A dummy tool with a radius of 10 mm and a length of 100 mm was manufactured from a solid stainless steel tool blank. The shaft was mounted in a lathe and polished using emery paper with grades ranging from 400 (grains/mm) to 1200. The shaft was finally polished using diamond paste with particles ranging from 9 µm to 0.25 µm and with a chemical polishing fluid. Quality control of the polished surface was performed using non-contact optical surface profile measurement (www.veeco.com). In the actual experiment a spray which is normally used for crack detection was used to create a removable diffuse (optically rough) surface on the polished dummy tool. Both the polished tool surface and the sprayed surface were measured by the optical profiler, and a representative area of the tool of 304 x 199 µm was sampled in steps of 414 nm. The measurements showed a normally distributed surface structure with Ra = 11.29 nm, implying that the polished surface is optically smooth compared to the laser wavelength of 633 nm. The sprayed surface showed substantially higher values, Ra = 21.02µm, giving an optically raw surface.

2.2 The milling machine
The LDV measurements were made on a Liechti Turbomill ST1200 ‘state-of-the-art’ machining centre offering multiple (5-axis) movement and a spindle capable of speeds of up to 24,000 rpm. The polished dummy tool was mounted in a Corogrip holder with an HSK shank which was in turn mounted in the machine and was not removed until all the measurements had been made.

2.3 Setting up the LDV
For the measurements, a PSV 300 LDV system from Polytec GmbH (www.polytec.com) including a displacement decoder was used. The LDV scanning head was mounted on a sturdy tripod and placed approximately 2 m from the tip of the dummy tool on a soft damped material to reduce the influence of structural floor vibrations. Care was taken to align the laser beam so that it’s centre line passed through the centre line of the shaft and was perpendicular to the shaft’s axis of rotation. This was necessary to ensure that the true velocity vector associated with the vibrations was along the incident direction of the laser beam. The LDV system was set up to perform sampling with a frequency of 40.96 kHz. The maximum detectable frequency was set by the system to 16 kHz. The LDV system produced frequency spectra with a standard FFT algorithm using a complex averaging method with 100 averages of 800 ms each giving a frequency resolution of 1.25 Hz and a total measuring time of 12.8 s.

2.4 LDV measurements
A series of experiments were carried out to establish whether vibrations of a rotating tool could be measured using the LDV system. Four different spindle speeds 2700, 4200, 6000 and 7200 rpm were studied. The LDV was used to measure the vibrations at the tip of the polished dummy tool in the radial direction at these speeds. The same set of measurements was carried out on the tool after being sprayed to give an optically raw finish.
Figure 1. LDV measurement of the rotating optically raw (sprayed) dummy tool.

Logged data was exported from the LDV as ASCII files and then imported into Matlab 6.0 where more detailed analysis and filtering of the data was carried out. The large-scale profile around the circumference of the dummy tool was measured using a mechanical roundness tester from C E Johansson (www.cej.se), with an accuracy of ± 0.3 µm. This was performed after that the LDV measurements were carried out.

3 RESULTS

In this section the results from the measurements at 6000 rpm are presented. The velocity spectrum of the polished and rough dummy tool measurements are displayed in the same chart for different frequency bands, Figure 2-5. The spectrum of the rough surface has been flipped down to the negative side in the charts to simplify comparison of the two spectra. In the charts it can be seen that the spectrum of the rough dummy tool contains peaks at f * n Hz where f is the rotational speed of 100 Hz (6000 rpm) and n = 1, 2, 3… These peaks are expected due to the presence of a speckle noise repeated for each dummy tool revolution in the sampled data.

A zoomed part of the spectrum covering the frequency band 8.8-10 kHz shows clearly the speckle noise in the form of peaks at integer multiples of the rotational speed of 100 Hz. These are marked with circles along the frequency axis, Figure 3. These peaks could not be seen in the graph of the polished tool. Between 1.1-1.5 kHz contains both speckle noise peaks
and ordinary vibrations, Figure 4. Note that the vibrations are present in both curves but the peaks are only present in the spectrum of the rough surface.

The frequency band covering 0-1 kHz shows harmonic peaks in both FFT graphs, see Figure 5. However, the first peak at 100 Hz in the polished measurement spectrum was detected as the dummy tool axial misalignment and the other six harmonics as the roundness profile. Figure 6 shows the signal from the displacement decoder. This signal is band-pass filtered between 0.15-0.75 kHz, thus filtering out the roundness profile. The result is shown in Figure 7, where the filtered time signal for one revolution is presented in a polar plot (dashed line), together with an independent mechanical measurement of the roundness made by the roundness tester (solid line). The difference between the curves is less than the error given by the manufacturer of the roundness tester (+0.3 µm). For the sprayed dummy tool the roundness could not be measured properly due to speckle noise caused by the rough surface. Similar results were achieved for measurements made at spindle speeds of 2700, 4200, and 7200 rpm.

![Velocity FFT, 6000 rpm](image)

Figure 2. Spectra of the polished and the sprayed surface at a spindle speed of 6000 rpm. The spectrum of the rough dummy tool has been mirrored along the frequency axis down to the negative side to simplify comparison between the two. Multiple harmonics of n*100 Hz where n=1,2,3… can be seen in the spectrum of the sprayed surface.
Figure 3. Zoomed part of the spectrum, 8.8-10 kHz. Frequencies where peaks are expected due to speckle noise are marked with a ring on the frequency axis.

Figure 4. Zoomed part of the spectrum, 1.1-1.5 kHz. Frequencies where peaks are expected due to speckle noise are marked with a ring on the frequency axis. It can clearly be seen that no peaks is present in the spectrum of the polished surface at the marked positions. Note that the vibration signal is present in both measurements.
Figure 5. Zoomed part of the spectrum, 0-1 kHz. Frequencies where peaks are expected due to speckle noise are marked with a ring on the frequency axis. In this graph peaks in both spectra are present at the marked frequency positions. For the polished case the peaks are identified as tool axial misalignment and roundness.

Figure 6. Displacement measurement at 6000 rpm.
Figure 7. Polar plot of the roundness of the dummy tool. Solid line: Measured with a mechanical roundness tester. Dashed line: Measured with LDV at a rotational speed of 6000 rpm. The distance between two circles in the plot is 2 µm.

4 DISCUSSION AND CONCLUDING REMARKS

Speckle noise interference was avoided by polishing the surface of the dummy tool (in effect a rotating shaft) until an optically smooth surface was achieved. The optically smooth surface of the rotating tool generated no repeated speckle noise and hence no unwanted peaks at integer multiples of the rotational frequency. This allowed the axial misalignment and roundness of the dummy tool to be measured at speeds of up to at least 7200 rpm using an LDV. This implies that radial vibration measurements of the tool can also be conducted; for example, when investigating dynamics of the cutting process.

The possibility of extracting roundness and alignment information is based on the fact that at a rotational speed of 6000 rpm (100Hz) any misalignment would be seen as a 100Hz signal. Since the tool is not perfectly round, harmonics of 100Hz will be present. The first component of the out of roundness is an elliptical form and results in a frequency peak at 2*100Hz (200 Hz). The second roundness component, a tri-lobed form, would be seen as a peak at 3*100Hz (300Hz) and so on. In the experiments, no significant peaks at integer multiples of the
rotational speed could be detected above the 6th component at 7*100Hz. To eliminate the possibility of structural vibrations being misinterpreted as misalignment or out of roundness, measurements were made at a number of rotational speeds (2700, 4200, 6000 and 7200 rpm). At each of these speeds, the misalignment data would be seen as a peak at a different frequency; namely 45Hz, 70Hz, 100Hz and 120Hz. Out of roundness data would be at multiples of the primary frequency. When measuring at speeds other than 6000rpm, any structural vibration around 100Hz would become clear, and the peak due to misalignment shifted. This makes it possible to analyse the presence of structural vibrations overlaying axial misalignment and out of roundness. No significant structural vibrations overlaying the roundness and misalignment data were detected in the experimental data.

However, since some frequency components of structural vibrations are spindle speed dependant it is not possible to draw firm conclusions about misalignment and roundness based solely on the LDV measurements. This effect must be investigated further. Good correspondence was however seen between LDV measurements and direct mechanical measurements of misalignment and out of roundness made using a dial test indicator and roundness tester. This indicates that no significant structural vibrations are overlaying the pitch and roundness data at the spindle speeds investigated.

Several problems must be overcome if LDV measurements are to be made of a rotating tool when it is cutting. Firstly, the laser beam must have a clear line of sight to the target surface on the tool without interference from cooling fluid or metal chips generated by the cutting process. The target surface must also be kept free of particles such as dust or process fluids. In milling machines where the tool moves relative to the machine base it must be possible to track the moving tool either by physically mounting the LDV on the moving axis of the machine or by some other tracking system. Finally, the alignment of the laser beam relative to the optically smooth target surface on the tool is also an issue that has to be considered. Axial misalignment, roundness, in-plane vibrations or applied cutting forces can affect the direction in which the laser beam is reflected from the target surface which could lead to a poor signal level or drop outs. In plane vibrations and deflection due to cutting forces together with the geometry of the shaft can also lead to misinterpretation of vibration data due to cross sensitivity. This has not been investigated in this work.

Different spindle / cutting speeds, cutting forces and changing component geometry affect the dynamics of a machining system. Measurement techniques based on physically mounting sensors, such as accelerometers, on the machine or workpiece can also affect the system dynamics. The ability to perform non-contact measurements of vibrations will allow measurement of changes in machine dynamics to be made during the cutting process without affecting the process itself. This is the subject of ongoing work.

5 ACKNOWLEDGEMENTS

The financial and technical support for this work was provided by the Swedish Agency of Innovation Systems (Vinnova) and SKF Nova, respectively. The LDV system and auxiliary equipment have been financed by the Kempe Foundations.
6 REFERENCES

Paper III
Laser vibrometry measurements of an optically smooth rotating spindle

Kourosh Tatar*, Matti Rantatalo** and Per Gren*

Luleå University of Technology, SE-97187 Luleå, Sweden
* Div. of Experimental Mechanics
** Div. of Sound and Vibrations

Abstract

Laser Doppler Vibrometry (LDV) is a well-established non-contact method, commonly used for vibration measurements on static objects. However, the method has limitations when applied to rotating objects. The LDV signal will contain periodically repeated speckle noise and a mix of vibration velocity components.

In this paper the crosstalk between vibration velocity components in laser vibrometry measurements of a rotating dummy tool in a milling machine spindle is studied. The spindle is excited by an active magnetic bearing (AMB) and the response is measured by LDV in one direction and inductive displacement sensors in two orthogonal directions simultaneously. The work shows how the LDV crosstalk problem can be avoided if the measurement surface is optically smooth, hence the LDV technique can be used when measuring spindle dynamics.

Keywords: Laser vibrometry; Crosstalk; Speckle noise; Spindle dynamics

1 INTRODUCTION

To be able to fully investigate the behaviour of a rotating system, such as a milling machine spindle, it is necessary to make measurements directly on the spindle during rotation. This can be done either by electronically or optically based non-contact measurement methods such as capacitive displacement sensors [1], Laser distance sensors [2] or Laser Doppler Vibrometry (LDV) [3]. The laser vibrometer is a powerful tool for measuring vibration velocities. The nature of the LDV system renders measurements without additional mass loading and allows a wide range of distances between the sensor head and the object (from millimetres up to several meters and scanning angles of about ±20°). However, two major problems occur when performing
LDV measurements on rotating objects; the presence of speckle noise and crosstalk between vibration velocity components. In some cases a tracking system can be used, where the laser beam follows the rotating surface. Measurements on propellers and tiers have been demonstrated [4, 5].

Speckles are a random pattern of dark and bright spots formed in space when a diffusely reflective surface is illuminated by coherent light (laser light). This is a result of superimposing wavelets of light with different traveled path length due to the surface structure. The speckle noise from a rotating shaft is generated by the moving speckle pattern on the LDV detector. This pattern is repeated for each revolution and will create a repeated noise in the measurement signal, called pseudo vibrations [6]. Larger surface structures than half the laser wavelength will result in fully developed speckles. It has been shown by prior authors that the speckle noise level can be reduced or removed with different methods. By optimizing the target-detector separation within a laser vibrometer the noise level can be reduced but not completely removed [7]. The speckle noise in laser torsional vibrometry measurements can be removed by randomising the path that the laser light is undertaking during the revolutions, either by moving the laser along the shaft [8] or simply by adding a new surface structure. The latter can be achieved by continuously applying e.g. oil or some other substances to the surface during the measurement [9]. In theory this technique should also work in laser doppler measurements.

The crosstalk problem can be described as an error-term in the measurement caused by a velocity component due to the rotation [10-13]. The measured velocities of a rotating optically rough shaft in the two orthogonal directions, \(v_x, v_y\), can be expressed as [13]:

\[
v_y = \dot{y} + \Omega(x - x_0)
\]

(1a)

and

\[
v_x = \dot{x} - \Omega(y - y_0),
\]

(1b)

where \(\dot{x}\) and \(\dot{y}\) are vibration velocities, \(x\) and \(y\) are the vibration displacements, \(x_0\) and \(y_0\) are the distances to the spin axis due to alignment errors and \(\Omega\) is the total angular velocity including torsional vibrations. \(\dot{x}\) and \(\dot{y}\) are the desired velocities to measure.

The methods for speckle noise reduction/removal described above cope with the specific speckle noise problem but are not able to neutralize the effect of crosstalk in a single beam LDV measurement. Consequently; the signal obtained during measurements under these circumstances will be a mix of the velocities in both directions.

A method for resolving the true vibrations in the two \(x\)- and \(y\)-directions using a setup of two simultaneously measuring lasers in both directions and an accurate measurement of the rotational angular velocity has been developed by Halkon and Rothberg [13].
In [3] it is shown that the speckle noise in laser vibrometry can be avoided by polishing the surface optically smooth, i.e. the surface roughness is much smaller than the laser wavelength. The out of roundness was measured and showed a good agreement with a mechanical roundness measurement. However, the crosstalk was not investigated.

In this work we investigate experimentally the crosstalk between the two directions (x and y), in laser vibrometry measurements of a rotating spindle with a polished surface. The spindle is excited by an active magnetic bearing (AMB) manufactured by SKF Revolve, and the response in the cross direction is measured by LDV and inductive displacement sensors.

2 EXPERIMENTAL SETUP AND PROCEDURE

Fig. 1. (a) Photo of the setup. Active magnetic bearing (AMB), Collet holder (CH), Dummy tool (DT), LDV measurement point (LMP) and spindle (S). (b) Schematic representation of the AMB. Displacement sensor (DS), Electromagnet (EM), Ferromagnetic material (FM).

Fig. 1(a) is a photo showing the AMB mounted in the milling machine table and Fig. 1(b) is a sketch of the AMB. The measurements were made in a Dynamite; 3-axis vertical table-top milling machine. The spindle (S) is capable of speeds of up to 7000 rpm. The dummy tool (DT) was mounted in a Collet holder (CH) with a Morse taper which was in turn mounted in the machine. The surface of the LDV measurement position (LMP) was polished optically smooth. The surface roughness was measured to $Ra = 0.021 \, \mu m$, using a Wyko NT1100 optical profiler (www.veeco.com). The surface could be made temporary rough by spraying it with a developer for crack testing (paint). This paint could easily be removed without scratching the surface. The spindle was harmonically excited by the AMB in the x- or y-direction with electromagnets (EM) at a cylindrical segmented part of the dummy tool consisting of a ferromagnetic material (FM). The electromagnets are arranged in two pairs opposite to each other. The vibrations of the spindle in the y-direction were measured by the vibrometer at LMP. Inductive displacement sensors (DS) within the AMB measured the displacements in the x- and y-directions simultaneously. The inductive displacement sensors are arranged in pairs, one pair measure the displacement in the x-direction and the other pair in the y-
direction. The sensitivity of the inductive displacement sensors are 110µmV⁻¹. The measurement range is 0-5 kHz and the limiting gap between the rotating dummy tool and the displacement sensors is 150 µm.

A PSV 300 LDV system from Polytec GmbH including a displacement decoder was used. The LDV scanning head was mounted on a sturdy tripod and placed 1 m from the polished dummy tool. Specular surfaces obey the law of reflection; angle of incidence = angle of reflection. For successful measurements on a specular surface the laser vibrometer must be aligned properly with the target rotation axis. In practice the arrangement was aligned by looking at the reflected laser light. A paper sheet with a hole for passing the laser beam was mounted on the scanning head. The laser beam was focused on the polished surface behaving like a cylindrical mirror. The reflected light sheet was adjusted so that the central part passed the aperture of the scanning head. The signal quality indicator of the LDV system showed a very high value at all spindle speeds so no dropout errors were present at the measurements. The LDV system was set up to perform sampling with a frequency of 32 kHz. The LDV sensitivity was set to 5mms⁻¹V⁻¹ during the measurements on the smooth surface. The sensitivity was then decreased to 25mms⁻¹V⁻¹ when measuring on the rough surface to avoid overloads due to the increased signal energy caused by the speckle noise. The measurement ranges were 10V and 31.6V respectively.

Five different spindle speeds, 700, 1400, 2800, 5600 and 7000 rpm, were studied. The presence of speckle noise was examined in the frequency domain of the LDV output during free run, which means that no forces where applied by the AMB. The eigenfrequencies of the dummy tool/spindle were extracted from the frequency response functions shown in Fig. 2, using the AMB controlling system software. Three different excitation cases were examined; excitation close to the first eigenfrequency of the tool spindle system (400 Hz), excitation above the first eigenfrequency (700 Hz), and excitation at the rotation frequencies.
The fact that the LDV and the displacement sensors measure at different positions along the z-axis (Fig. 1(b)) will give different signal amplitudes. The dummy tool was exited in the y-direction and the displacement sensor output and the LDV output where compared. The velocity (differentiated displacement) at the displacement sensor is about four times greater than at the laser measurement point (LMP) at all measured spindle speeds during 400 Hz and 700 Hz excitation. When the excitation frequency coincides with the rotation frequency the recalulation factor is about seven. Henceforth, the data from the displacement sensor is recalculated to the LMP.

The crosstalk in the LDV measurements of a rough and polished rotating shaft was studied. For each case the shaft was excited with different frequencies in the x-direction for the complete set of spindle speeds.

3 RESULTS

Measurements were first performed at free run. Fig. 3 shows the displacement amplitudes of the second to nineteenth rotational harmonics of the polished dummy tool. The amplitudes are independent of spindle speed and are rapidly decaying, which indicates that they are not caused by random speckle noise. The profile of the polished measurement surface is not perfectly round. The actual deviation from a perfect circle will be recorded by the LDV. These roundness components are seen in the FFT as rotation harmonics. The third harmonic has the highest value, which means that the triangular component is the dominant one. By band-pass filtering the displacement signal it is possible to reconstruct the profile of the surface [3]. An Independent
measurement (at a later time) of the surface using a mechanical roundness tester (C E Johansson) confirms these peaks and some scratches. The scratches are also confirmed by the surface profile measurements performed by the Wyko equipment. Otherwise the surface was optically smooth (Ra=21nm).

Fig. 3. The displacement amplitudes of the second to nineteenth rotational order of the polished dummy tool.

To examine the crosstalk, the dummy tool was excited in the x-direction (cross direction to the LDV) at 400 Hz, 700 Hz and at the rotational frequencies. In each case, two sets of measurements were made; firstly on the polished dummy tool with an optically smooth surface and secondly on the dummy tool after being sprayed with paint to give an optically rough surface. Fig. 4 illustrates the effect of crosstalk in LDV measurements of a rotating rough surface for different spindle speeds. The vibration velocity measured on the dummy tool after being sprayed with paint (triangle up) shows a spindle speed dependent crosstalk as expected from Eq. (1), while the same measurements on the smooth surface (triangles down) does not. The outputs from the displacement sensor (DS) in the y-direction for both sets of measurements (smooth and rough measurement surface) are also presented in the graph (square and pentagram). The signal to noise ratio was checked to be 43dB in a typical case for the inductive displacement sensors. The differentiation of the signal was performed in the frequency domain at the excitation frequency. Inserting the signals from the displacement sensors \( \dot{y} \) and \( x \) into Eq. (1a) results in the expected velocity from the vibrometer when the surface is rough (circle). Numerically there is a good agreement between the calculated and the measured velocity. In (a) the excitation frequency is at 400 Hz and the displacement and velocity amplitudes in the x-direction are about 6 \( \mu m \) and 15100 \( \mu m/s \cdot 1 \) at the LMP respectively.
In (b) the excitation frequency is at 700 Hz and the displacement and velocity amplitudes in the x-direction are about 4.5 µm and 19800 µm/s at the LMP respectively. Despite the different excitation level and frequency the results in (a) and (b) are consistent. There are some differences between the LDV output from the measurements on the optically smooth surface and the velocities obtained by the displacement sensor in the y-direction (triangle down and square). These differences are due to a small misalignment (few degrees) between the LDV and the displacement sensor in the y-direction. The sensitivity to misalignment has been checked by calculations. Since the excitation in the x-direction is comparatively large, even small misalignment angles do change the level of the output considerably. For example the difference in (a) can be compensated by an angle of about 4.5 degrees. In (c) the dummy tool was excited at the rotation frequencies, 11.7, 23.4, 46.7, 93.3 and 116.7 Hz. The vibrometer output from the measurements on the smooth surface shows no crosstalk and follows the differentiated displacement sensor output. The crosstalk effect lowers the vibration amplitude level in this case contrary to the previous two excitation frequencies. This shows that the crosstalk can result in either a higher vibration level or a lower one than the correct one.
Fig. 4. Crosstalk in LDV measurements for different spindle speeds. Excitation at (a) 400 Hz, (b) 700 Hz and (c) at rotation frequencies.
4 DISCUSSION

The laser beam alignment is one of the most important and critical step in rotating components measurement. In this study the laser vibrometer must also be aligned with the inductive displacement sensors for comparison. Further the amplitude of the orthogonal excitation displacement is even more important when studying the crosstalk problem, which means that the inductive displacement sensors measuring the displacements in the cross direction must be perfectly orthogonal to the laser vibrometer. This task showed to be difficult to overcome. The inevitable instrument angular misalignment was though minimized by try-and-error which was time-consuming. The actual value of the angular misalignment is unknown and difficult to control now afterwards. Backward calculations estimate the misalignment to be about 4.5 degrees in Fig. 4(a) which is not unreasonable. However, the objective of this paper is to investigate the crosstalk in laser vibrometry measurements on an optically smooth rotating spindle. The crosstalk is angular velocity dependence and even small displacements in the orthogonal direction result in amplitude differences for different but still high spindle speeds. Despite the systematic misalignment error the laser vibrometry measurements show no such spindle speed dependence. The differences between the level of the laser vibrometry measurements and the level of the displacement sensor outputs associated with the instrument misalignment, noise and measurement error should not interfere with the crosstalk investigation. It is shown that the crosstalk can influence the vibration measurement producing a vibration level increase or decrease with respect to the correct level. The latter trend can be explained by the following example. Suppose that the displacements are

\[ x = A_x \sin(\omega t), \]  

(2)

And

\[ y = A_y \sin(\omega t + \phi), \]  

(3)

where \( \omega \) is the vibration frequency and \( A_x \) and \( A_y \) are the displacements amplitudes. Differentiating (3) gives the translational velocity in the y-direction

\[ \dot{y} = A_y \omega \cos(\omega t + \phi). \]  

(4)

Inserting expression (2) and (4) into (1a) and neglecting the alignment error \( x_a \) gives the measured velocity

\[ v_y = A_y \omega \cos(\omega t + \phi) + \Omega A_y \sin(\omega t). \]  

(5)

If the vibration frequency is equal to the rotation frequency, \( \omega = \Omega \) and if the phase difference between the displacements in the x- and the y-direction is 90˚, the measured velocity (5) becomes
\[ v_y = \Omega (A_x - A_y) \sin(\Omega t), \] (6)

and in the worst case; when the amplitudes \( A_x \) and \( A_y \) are equal the LDV output becomes zero.

5 CONCLUSIONS

Laser vibrometry is normally used on stationary vibrating objects, although synchronous tracking of the laser beam with the moving surface has been tried in number of cases by others. But to measure vibrations of a rotating spindle necessitates that the laser beam is stationary in space. If the rotating surface is optically rough, a moving speckle pattern will occur on the detector, which gives a repeatable speckle noise in the measurement signal and also crosstalk from other velocity components.

By using a polished surface, the crosstalk in LDV measurements is avoided and the desired vibration can be measured. The fact that the cross vibrations applied by the AMB is removed from the LDV measurements automatically imply the removal of torsional vibration contributions.

The frequency content in the signal due to roundness components did not indicate in any detectable crosstalk.

The scanning LDV also provides the possibility to measure vibrations on different parts of a rotating machine, i.e. the milling machine spindle housing and other significant components during machining, which will give an overall picture of the system.

6 ACKNOWLEDGEMENTS

The work is supported by the Swedish Agency for Innovation System (Vinnova) and SKF Nova, respectively. The purchase of the vibrometer is financed by the Kempe-foundations. The authors wish to thanks Lars Frisk, for optimizing the polishing procedure of the dummy tool, and also Pär Marklund, for surface roughness measurements.

7 REFERENCES


Paper IV
Milling machine spindle analysis using FEM and non-contact spindle excitation and response measurement

Matti Rantatalo*, Jan-Olov Aidanpääb, Bo Göranssonc, Peter Norman
d

a Luleå University of Technology, Division of sound and vibration, SE- 971 87 Luleå Sweden
b Luleå University of Technology, Division of Computer Aided Design , SE- 971 87 Luleå Sweden
c SKF Nova, HK1-6, 415 50 Göteborg, Sweden
d Luleå University of Technology, Division of Manufacturing Systems Engineering, SE- 971 87 Luleå Sweden
*Corresponding author. Tel: +46 (0)920-49 21 24, fax: +46 (0)920-4910 30, e-mail: matti.rantatalo@ltu.se

ABSTRACT
In this paper a method for analysing lateral vibrations in a milling machine spindle is presented including finite element modelling (FEM), magnetic excitation and inductive displacement measurements of the spindle response. The measurements can be conducted repeatedly without compromising safety procedures regarding human interaction with rotating high speed spindles. The measurements were analysed and compared with the FEM simulations which incorporated a spindle speed sensitive bearing stiffness, a separate mass and stiffness radius and a stiffness radius sensitive shear deformation factor. The effect of the gyroscopic moment and the speed dependent bearing stiffness on the system dynamics were studied for different spindle speeds. Simulated mode shapes were experimentally verified by a scanning laser Doppler Vibrometer (LDV). With increased spindle speed, a substantial change of the eigenfrequencies of the bearing related eigenmodes was detected both in the simulations and in the measurements. The centrifugal force that acted on the bearing balls resulted in a softening of the bearing stiffness. This softening was shown to be more influential on the system dynamics than the gyroscopic moment of the rotor. The study performed indicates that predictions of high speed milling stability based on 0 rpm tap-test can be inadequate.

Key words: Machine tool spindle, centrifugal, gyroscopic, non-contact measurement, angular contact ball bearings
1 INTRODUCTION

Turning operations like milling are common in the automotive and aerospace industry where large metal work pieces are reduced to a fraction of its original weight creating complex thin structures. It is important that unwanted behaviours like tool vibrations can be avoided during these operations. Especially self-excited machine tool chatter caused by the waviness of the machined surface. This type of vibration will cause poor surface finish and in some cases material or machine damage. The phenomena is a significant issue and has been addressed and modelled by numerous authors during the past decades e.g. [1-4]. The developed models predict a specific chatter free depth of cut which is governed by the transfer function of the tool tip, assuming a rigid work piece. The chatter free depth of cut is calculated for different spindle speeds which can be plotted as a stability lobe chart. The transfer function is normally measured manually by tap-tests of a non-rotating spindle/holder/tool system where the tool tip is excited by an impulse hammer and the response is measured by a vibration transducer. The assumption in this procedure is that the dynamics of the spindle/holder/tool system is independent of the spin speed. This is however not true for the case of high speed milling operations where the effect of gyroscopic moments and centrifugal forces must be taken into account [5-7].

To analyse the spindle speed dependency, the machine tool must be analysed in a rotating state that spans the whole range of operating spindle speeds. Schmitz et al. [8] presented an experimental method for the prediction of stable cutting regions which reflects the dynamic change that a rotating system undertakes. The method was based on impulse hammer excitation and capacitive probe response measurement of a tool rotating during different spindle speeds. Stability lobes for a discrete number of spindle speeds were calculated and the limit of stable cut corresponding to the actual spindle speed used was picked out to form a spindle speed dependent stability lobe chart. Experimental tests revealed a changing stable limit for stable cut above 16000 rpm due to changing spindle dynamics.

An alternative method to analyse a spindle bearing system is by modelling. Wang and Chang [9] presents a spindle modelling method based on FEM. The model did however not include rotation and therefore no centrifugal forces and gyroscopic moments. In 1976 Nelson and McVaugh [10] presented a FEM formulation of a rotor bearing system based on the Euler Bernoulli beam theory where the effect of gyroscopic moments and centrifugal forces was included. Zorzi and Nelson [11] later on added internal damping and in 1980 Nelson [12] presented another formulation based on the Timoshenko beam theory which included the shear deformation effects. Xiong et al. [6] presented a way of combining this FEM representation and the milling cutting force model formulated by Altintas [2]. The model, which only consisted of the rotor, predicted that the gyroscopic moment would not affect the stability regions in milling but increases the real part of the eigenvalues and so forth reducing the axial depth of cut. The model also predicted a change of the spindle resonance frequencies of about ± 10 Hz. Chi-Wei Lin et al. [13] integrated a thermo-mechanical-model to the Timoshenko FEM description. Numerical and practical experiments verified an increase in bearing stiffness with increasing bearing preload. The work also predicted a softening of the spindle shaft with increasing spindle speed. It was shown that the softening of the bearing radial stiffness due to speed could be compensated for by the thermally-induced preload. Cao and Altintas [7] presented a general method for the modelling of a spindle bearing system including the axial coordinate and a corresponding spindle speed and preload dependent five degree of freedom bearing stiffness matrix. In the spindle model a rotor related centrifugal force was modelled by subtracting a term $(\Omega^2 \times \text{multiplied by a radial version)$.
of the translational mass matrix) from the stiffness matrix. Simulations for different spindle speeds were performed but only verified for a non-rotating spindle. Simulations predicted that the centrifugal force of the rotor would influence the eigenfrequencies more than the gyroscopic moment of the rotor. This result was not verified experimentally.

This paper describes a method for analysing lateral machine tool spindle vibrations based on a finite element model (FEM) and a contact-less dynamic spindle testing equipment (CDST). The aim of this work is to study the effect of the gyroscopic moment and the speed dependent bearing stiffness on the system dynamics. The study was performed on a 5-axis Liechti Turbomill ST1200 with an Fischer spindle (MFWS-2305/24/8) with an integrated motor capable of speeds up to 24000 rpm and a pair of 25º angular contact ball bearings. The spindle spring preload was achieved experimentally and a speed dependent bearing stiffness was calculated. The FEM elements were based on separate stiffness and mass radius. Simulations were performed with and without the speed sensitive bearing stiffness, together with a stiffness radius sensitive shear deformation factor. Tap tests with accelerometers and a scanning laser Doppler vibrometer were used to verify the CDST measurements and the FEM simulation.

2 CDST MEASUREMENTS

The CDST measured the frequency response function (FRF) of the tool tip by exciting the rotor with electromagnets while inductive displacements sensors registered the rotor position in the x and y direction. The use of electromagnets and non-contact displacement sensors is common in the field of active magnetic bearings (AMB). An AMB uses a control system in order to keep the rotor in place by adjusting the coil current in the magnets according to the measured rotor position and a desired location. In the segment of machine tools AMB’s are mainly used as rotor support bearings but other applications have been developed during the past years. Auchet et al. [14] developed a method for indirect cutting force measurement by analysing the command voltage of magnetic bearing in a milling machine spindle supported by active magnetic bearings. Knospe [15] investigated the potential of active chatter suppression by the use of AMB, and Chen and Knospe [16] presented an approach to estimate the cutting dynamics by both exciting the system and increasing the damping of the lathe tool using an AMB.
Fig 1. A: Photo of the setup with the dummy tool in an elevated state. B: See-through sketch of the CDST with dummy tool in place. +dx and +dy denotes the displacement sensors. C: Electromagnet setup with dummy tool (DT).

Fig 1 shows a photo and a sketch of the experimental setup used in the study. The spindle/holder/tool system dynamic was measured at the tool tip in the x- and y-direction separately by the use of a CDST. The excitation of the rotor was carried out by electromagnets which were fed by frequency step vice sine sweep coil current, thereby introducing a magnetic force $F_m(t)$ which acted on the rotor. The rotor consisted of a specially manufactured dummy tool with a laminated rotor part in order to reduce the energy losses due to eddy current effects. In each direction two electromagnets (e.g. x-direction v1 and v3) on opposite sides of the rotor were working out of phase with each other while attracting the rotor to form the excitation.

The force $F_m(t)$ applied to the rotor in the x-direction (analogous in the y-direction) is expressed in terms of the stator coil current and the instantaneous air gap as [17]:

$$F_{sw} = C_{sw} \left( \frac{I_{sw}}{(d + x_{w0} + x_m)} \right)^2 - \left( \frac{I_{sb}}{(d + x_{a0} + x_m)} \right)^2, \quad (1)$$

where $C_{sw}$ is a calibration factor, $I_{sw}$ is the measured current of top quadrant (v1), $I_{sb}$ is the measured current of bottom quadrant (v3). $d$ is the effective gap of 150 µm between the magnets and the rotor, $x_{w0}$ is the magnetic centre offset and $x_m$ is the instantaneous displacement measured by the displacement sensors. The coil current $I_{sw}$ and $I_{sb}$ is a superposition of the excitation current, a bias current and a compensation current for static loads, which are zero in this case.
The instantaneous displacement of the rotor in the x and y direction was measured by two displacement sensors each which were coupled together for each coordinate with opposite signs facing the rotor with 180 degrees apart. This arrangement enables the cancellation of any changes in displacement due to thermal expansion of the rotor diameter.

The displacement response and the exciting force for each frequency component in the interval 400-2000 Hz were measured and a spectrum estimation (H1) of the transfer function was calculated. This procedure was repeated for all speeds in the interval [0:2000:24000] rpm and for each radial direction x and y. A reference accelerometer was mounted on the CDST housing to ensure that the assumption of a rigid CDST construction and a firm machine table mounting would hold. The CDST were verified experimentally by tap tests of the mounted dummy tool at 0 rpm. Except for the third mode in the y direction the two different measurement methods resulted in similar frequency response functions see Fig 2. The mode shapes for 0 rpm were analysed by a scanning LDV which performed a line scan along the z-axis of the visible part of the dummy-too, holder and rotor.

3 SPINDLE MODELLING

A finite element model described by [12] is used to simulate the mode shapes and the eigenfrequencies of the rotor bearing system. The simulations are used to analyse and identify any detected gyroscopic or centrifugally induced speed dependency in the frequency response measured by the CDST. Fig 3 illustrates the finite element model and the element division of the rotor bearing system plotted along the x-z plane. Each FEM element of length \( l \) consists of two parts with an inner and outer radius see Fig 4.
The hollow shaft of the rotor with its inner stiffness radius $r_i$ and the outer stiffness radius $R_i$ governed the stiffness properties of the spindle. The physical outer mass radius $R_m$ which included the shaft, motor package, inner rings of the bearings and other additional parts governed the mass together with an assumed physical inner mass radius $r_m$. The assumed inner mass radius $r_m$ inside the spindle shaft models the spring-package and the drawbar inside the spindle used for connecting the tool holder to the spindle. The connection between the spindle, holder and the dummy tool was not specially modelled. Damping, gravity and the centrifugal effects of the rotor described by [7] was not included. The axial load during free run was considered neglectable and therefore excluded in the study.

The radial displacement and rotation along the radial coordinates of a single element is expressed in the generalised coordinates $q = \{x_i, y_i, \phi_{x,i}, \phi_{y,i}, x_{y,i}, y_{x,i}, \phi_{x,y,i}, \phi_{y,x,i}\}$ where $i$ is the node number. The area moment of inertia which is used in the forming of the stiffness matrix was based on the stiffness radiuses $I = (R_i^4 - r_i^4) \pi / 4$. The polar moment of inertia $J_p = (R_i^4 - r_i^4) / 4$ and the diametral moment of inertia $J_d = (R_m^4 - r_m^4) / 2$ were based on the mass radius. The shear deformation is expressed as $\Phi = (12EI)/(\pi G J^2)$, where $E$ is the modulus of elasticity, $G$ is the shear modulus and $\kappa$ is the shear deformation factor. The shear deformation factor is normally determined experimentally and for a solid circular shape a usual value is approximately 0.9. However, the spindle studied in this paper consists of a hollow circular shaft with a variable stiffness radius. In 2001 Hutchins [18] proposed an analytical expression for the shear deformation coefficients of a hollow circular shaft expressed as

$$\kappa = \frac{6 \left( R_i^4 + r_i^4 \right)(1 + \nu)^2}{7r_i^4 + 34r_i^4 R_i^2 + 7R_i^4 + \nu(12r_i^4 + 48r_i^2 R_i^2 + 12R_i^4) + \nu^2(4r_i^4 + 16r_i^2 R_i^2 + 4R_i^4)}$$  \hspace{1cm} (2)

which has been used in this work.
The homogenous equation of motion for the finite element assembly used in this model is expressed as:

\[
[M] \ddot{\mathbf{q}} + \Omega[G] \dot{\mathbf{q}} + [K] \mathbf{q} = \mathbf{0}
\]

(3)

where \([M]\), \([G]\), and \([K]\) are the system matrixes of a shaft element see Appendix. The assembled second order homogenous differential equation was transformed into a first order differential equation, using the state vector notation described in [19]. The equation of motion could then be rewritten as:

\[
\begin{bmatrix}
\Omega[G] & [M] \\
-\Omega[M] & 0
\end{bmatrix}
\begin{bmatrix}
\dot{\mathbf{q}} \\
\dot{\mathbf{q}}
\end{bmatrix}
+
\begin{bmatrix}
[K] & [0] \\
[0] & [M]
\end{bmatrix}
\begin{bmatrix}
\mathbf{q} \\
\mathbf{q}
\end{bmatrix}
= \mathbf{0},
\]

(4)

where

\[
\{\dot{\mathbf{q}}\} = \begin{bmatrix}
\dot{\mathbf{q}}_1 \\
\dot{\mathbf{q}}_2 \\
\vdots \\
\dot{\mathbf{q}}_n
\end{bmatrix}
\]

(5)

is the state vector. Solving the obtained first order homogenous differential equation gives complex eigenvectors with corresponding complex eigenvalues of the displacement and the velocity of each node and its generalized coordinates. The front and the back bearing stiffness were added into the complete rotor stiffness matrixes at corresponding nodal coordinates marked by triangles in Fig 3.
3.1 Preload measurement

To calculate the bearing stiffness the preload force of the bearings has to be known. However when dealing with a real milling machine spindle the bearing preload is normally not a known parameter and must therefore be measured. On line measurements of the bearing load would be preferable and a method for this has been presented by Chen and Chen [20]. For most spindles in operation, this facility is however not included and the preload must therefore be measured or provided by the designer of the spindle. The spindle used in this study was designed using spring loaded bearings as shown in Fig 5. The front and the back bearings are hybrid angle-contact bearings placed back to back. The preload was measured by pulling the spindle towards –z while the axial displacement was measured using a dial indicator with a resolution of 1μm and a total measuring range of +/- 50μm. The force used to pull the spindle was measured by a static force sensor with a range from 0 to 5kN with a resolution of about 20N. The static force sensor was made out of strain gauges and connected to a strain gauge amplifier and an oscilloscope. Data was collected manually according to predefined steps. The estimated spring preload was equal to the force needed to unload the front bearing. The force needed to unload the front bearing was estimated to 1450N.

![Fig 5. Principal spindle drawing and preload measurement setup.](image)

3.2 Bearing stiffness calculations

Using the result from the preload measurement the bearing stiffness within the speed interval [0:2000:24000] rpm were calculated. The bearing stiffness calculations were performed by SKF and an in house developed software “Bearing Beacon” based on the theory described in [21] and [22] by de Mul et al.

![Fig 6. Contact angles together with the Hertzian and centrifugal forces which acts on one ball.](image)
4 RESULTS

4.1 Bearing stiffness

With increasing speed the load conditions between the ball and the rings in the bearing changes due to the centrifugal forces $F_c$ which act on the balls. See Fig 6. The centrifugal force induced by the rotating balls forces the rings to separate axially. A new equilibrium state is reached where the contact forces $Q_i$ and $Q_e$ will balance the new force condition which includes both external forces and the centrifugal force $F_c$. When the ball reaches this new asymmetric position it will act like a spring when the external force changes. This spring is in serial with the normal Hertz contact springs resulting in a decreased bearing stiffness for high spindle speeds. Table 1 shows the stiffness matrix of the back bearing and front bearing calculated for 0 rpm.

<table>
<thead>
<tr>
<th></th>
<th>x [1/m]</th>
<th>y [1/m]</th>
<th>yz [1/rad]</th>
<th>zx [1/rad]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Back</td>
<td>Fx [N]</td>
<td>3.36e8</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>Fy [N]</td>
<td>0.00</td>
<td>3.36e8</td>
<td>6.10e6</td>
</tr>
<tr>
<td></td>
<td>Myz [Nm]</td>
<td>0.00</td>
<td>6.10e6</td>
<td>1.15e5</td>
</tr>
<tr>
<td></td>
<td>Mzx [Nm]</td>
<td>-6.10e6</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td>Front</td>
<td>Fx [N]</td>
<td>3.38e8</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>Fy [N]</td>
<td>0.00</td>
<td>3.38e8</td>
<td>6.76e6</td>
</tr>
<tr>
<td></td>
<td>Myz [Nm]</td>
<td>0.00</td>
<td>6.76e6</td>
<td>1.39e5</td>
</tr>
<tr>
<td></td>
<td>Mzx [Nm]</td>
<td>-6.76e6</td>
<td>0.00</td>
<td>0.00</td>
</tr>
</tbody>
</table>

The bearing stiffness for different spindle speeds related to the stiffness at 0 rpm are plotted in Fig 7. The plot shows that the stiffness in the radial direction of the back bearing is decreasing to a level of 62% of its original (0 rpm) value when the speed increases to 24000 rpm. The corresponding value of the front bearing is 38%. The angular stiffness shows approximately the same amount of reduction for the same speed range. The reduction of the front bearing stiffness tends to diminish more than the back bearing above 18000 rpm, giving it a characteristic shape (hereinafter denoted “s-shaped”).
Fig 7. Changes in bearing stiffness with increasing spindle speed. 100% represents the bearing stiffness at 0 rpm. The corresponding radial, angular and coupled stiffness in the other directions (see white cells in Table 1) shows an equal behaviour and are therefore not presented in the plot.

4.2 Gyroscopic and centrifugal effects

In a rotor bearing system the modes often appears as a mix of rotor related flexural modes and rigid body modes governed by the bearing properties [17]. The modes will furthermore be effected by the gyroscopic moment of the rotor and split up into two mode shapes [23]. The eigenfrequencies of these forward and a backward modes will be influenced by the driving frequency $\Omega$. This can be seen in Fig 8 where the simulated eigenfrequencies with speed independent bearing stiffness are plotted. The simulation shows an increasing difference with increasing speed between the two eigenfrequencies and their value at 0 rpm. The deviation at 24000 rpm compared to 0 rpm is $\pm 9$ Hz, $\pm 3$ Hz, $\pm 15$ Hz and $\pm 23$ Hz for modes 1-4 respectively, see Table 2. The gyroscopic effect of the rotor can also be seen in the eigenfrequencies of the simulation with speed dependent bearing stiffness, see Fig 9. In this simulation the influence of the characteristic shape of the decreasing bearing stiffness can be seen added to the gyroscopic effect resulting in a deviation at 24000 rpm of 204 and 190 Hz, 202 and 192 Hz, 16 and 15 Hz and finally 100 and 55 Hz for the first four backward and forward modes.
Fig 8. Eigenfrequencies of the four first modes with the effect of the gyroscopic moment of the rotor.

Fig 9. Eigenfrequencies of the four first modes with the effect of the speed dependent bearing stiffness and the gyroscopic moment of the rotor.
4.3 Mode shape analysis

Analysis of the simulated mode shapes, see Fig 10, reveals that the first mode shape starts at 0 rpm as a mode governed by the back bearing with its node close to the front bearing position. When the speed reaches 24000 rpm the mode has transformed into a mode governed by the front bearing stiffness. According to the simulation the transformation is performed in the interval 12000-16000 rpm where the two first modes intersect. The first mode transforms its appearance by sliding its front bearing node position towards the back bearing position. The second mode shape goes through a similar transformation by starting as a front bearing mode at 0 rpm, going through a cylindrical mode shape governed by both bearings around 14000 rpm and finally end up as a mode shape governed by the back bearing at 24000 rpm. The transformation area can be seen in Fig 9 as a disturbance of the shape of the changing eigenfrequency. Mode 3 and 4 are primarily flexural modes and are not changing its appearance in the same extent. The LDV measurement confirmed the simulated mode shapes at the visible part of the spindle Fig 10.

Fig 10. Mode shape analysis of the four first eigenmodes. Where no complex conjugate (dashed shapes) is present in the simulation the mode shape is real valued. Drawings at top illustrate the positions of the mode shapes. LDV measured mode shapes are displayed to the left.
Fig 11. CDST measurement frequency response function Hxx.

Fig 12. CDST measurement frequency response function Hyy
4.4 CDST measurements

The frequency response functions $H_{xx}$ and $H_{yy}$ of the spindle/holder/dummy-tool measured by the CDST are shown in Fig 11 and Fig 12. The two measurements made in the $x$ and $y$-directions show a decreasing s-shaped pattern for the eigenfrequencies of the second mode (at 752 Hz) and the fourth mode (at app. 1500 Hz). Only the second front bearing related rigid rotor mode can be seen in the measurement for all spindle speeds. The first back bearing related mode is only vaguely seen in the $y$-direction at 664 Hz. The same mode is detected in the tap test at 661 Hz see Fig 2. The third eigenmode at app. 900 Hz shows a relatively constant eigenfrequency even thought the spindle speed increases. A small change of the eigenfrequency (0.9-2.2%) of this mode (forward and backward) is detected in the simulation and in the CDST measurements. The simulated separation of the modes into a backward and a forward mode can not be detected in the measurements. A possible intersection area where the two rigid rotor bearing modes intersects can be seen in the CDST measurement at 12000 rpm and above as a change of magnitude. Furthermore the amplitude of this second mode increases with increasing spindle speed, especially above 12000 rpm. The CDST measurement also shows a higher frequency for the fourth mode when excited in the $y$ direction compared to the $x$ direction.

Table 2. Eigenfrequencies of CDST measurement and FEM simulation for 0 and 24000 rpm. FEM 1 is simulation with speed dependent bearing stiffness and FEM 2 is without. (-) Not detected.

<table>
<thead>
<tr>
<th>Mode 1 [Hz]</th>
<th>FEM 1</th>
<th>FEM 2</th>
<th>CDST</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Backward</td>
<td>Forward</td>
<td>Backward</td>
</tr>
<tr>
<td>0 rpm</td>
<td>711</td>
<td>711</td>
<td>711</td>
</tr>
<tr>
<td>24000 rpm</td>
<td>507</td>
<td>521</td>
<td>702</td>
</tr>
<tr>
<td>Frequency Change %</td>
<td>-40.2</td>
<td>-36.5</td>
<td>-1.3</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Mode 2 [Hz]</th>
<th>FEM 1</th>
<th>FEM 2</th>
<th>CDST</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Backward</td>
<td>Forward</td>
<td>Backward</td>
</tr>
<tr>
<td>0 rpm</td>
<td>783</td>
<td>783</td>
<td>783</td>
</tr>
<tr>
<td>24000 rpm</td>
<td>581</td>
<td>591</td>
<td>780</td>
</tr>
<tr>
<td>Frequency Change %</td>
<td>-34.8</td>
<td>-32.5</td>
<td>-0.4</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Mode 3 [Hz]</th>
<th>FEM 1</th>
<th>FEM 2</th>
<th>CDST</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Backward</td>
<td>Forward</td>
<td>Backward</td>
</tr>
<tr>
<td>0 rpm</td>
<td>912</td>
<td>912</td>
<td>912</td>
</tr>
<tr>
<td>24000 rpm</td>
<td>896</td>
<td>927</td>
<td>897</td>
</tr>
<tr>
<td>Frequency Change %</td>
<td>-1.8</td>
<td>+1.6</td>
<td>-1.7</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Mode 4 [Hz]</th>
<th>FEM 1</th>
<th>FEM 2</th>
<th>CDST</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Backward</td>
<td>Forward</td>
<td>Backward</td>
</tr>
<tr>
<td>0 rpm</td>
<td>1578</td>
<td>1578</td>
<td>1578</td>
</tr>
<tr>
<td>24000 rpm</td>
<td>1478</td>
<td>1523</td>
<td>1555</td>
</tr>
<tr>
<td>Frequency Change %</td>
<td>-6.8</td>
<td>-3.6</td>
<td>-1.5</td>
</tr>
</tbody>
</table>
5 DISCUSSION AND CONCLUDING REMARKS

In this paper, a method for analysing lateral vibrations in a milling machine spindle has been presented including a contact less spindle dynamic measurement equipment substantiated with FEM simulation. The FEM formulation was based on reference [12] and was extended by including a separate mass and stiffness radius together with a stiffness radius dependent shear deformation factor. The frequency response functions in the radial directions for speeds in the interval [0:2000:24000] rpm were measured by the CDST without compromising safety regulations regarding human interaction with high speed rotating spindles. The method was applied to a Liechti Turbomill ST1200 equipped with a Fischer spindle capable of 24000 rpm. The machine tool was designed with an integrated motorised spindle supported by a pair of 25º angular contact ball bearings. In order to calculate the bearing stiffness the spindle spring preload was retrieved experimentally.

5.1 Gyroscopic and centrifugal effects

The bearing stiffness was found to be sensitive to the centrifugal force acting on the bearing balls. This effect resulted in a substantial decrease in bearing stiffness (38-62%) and hence bearing related eigenfrequencies with up to 40 %. This effect was confirmed by the CDST measurements. According to the simulations, the centrifugal effect in the bearings had a more significant effect on the eigenfrequencies of bearing related modes than the gyroscopic moment of the rotor. E.g. mode 2 shows a 33-35 % frequency change with centrifugal effect present compared to only 0.4% when not. One could expect that this softening could be neutralised for high loads forcing the balls back to their nominal positions. Another possibility of reducing the axial separation force of the rings would be by using angular contact bearings with a smaller angle (e.g. 15º). The softening of the rotor with increasing spindle speed and hence the reduction of the eigenfrequencies which was reported by previous authors could not be detected. This is evident when analyzing the third simulated and measured mode, (flexural) at app. 900 Hz, where the eigenfrequency only seems to be affected by the gyroscopic moment.

5.2 Mode analysis

The separation of the modes into a backward and a forward mode could not be detected in the measurements most likely due to damping which could smears the signal energy from the two modes into a single peak. The measured eigenmode at 752 Hz for 0 rpm is assumed to be governed by the front bearing. This assumption is based on the mode shape analysis where the front bearing mode normally would result in a larger displacement at the tool tip than the back bearing mode. Individual damping and phase conditions of these two modes could reduce and smear the amplitude peak of the first mode. According to the simulation of the second mode, the shape of the frequency change would begin by following the “s-shape” of the front bearing stiffness, and then jump over to follow the shape of the back bearing stiffness. The same but in the opposite order would then also apply to the first eigenmode. This tendency can be seen in the simulation see Fig 9. Due to this, a correct reading and identification of the eigenfrequencies of these two modes above the intersection area is difficult. Hence the CDST frequencies identified and listed in Table 2 of the second mode at 24000 rpm could be influenced by each mode individually or by both modes together. The relatively constant eigenfrequency of the eigenmode at 912 Hz is consistent with the simulation which also indicated that the third eigenmode was a flexural mode with little influence of the radial bearing stiffness. This was also confirmed by the
mode shape simulation see Fig 10. The characteristic s-shaped pattern could also be seen in the CDST measurement of the fourth mode. This indicates that the mode in some way is related to the front bearing. This indication was confirmed by the mode shape analysis which showed that this mode was a mix of a front bearing mode and a flexural mode.

5.3 Accuracy and validation

The CDST measurements were compared with traditional tap-tests at 0 rpm. The tap-tests showed e.g. slightly lower eigenfrequency for the third mode in the y-direction. This could be due to the differences in dummy tool positions for the two measurement cases. Despite this small difference the tendency shown in the CDST measurements is not compromised and the change of spindle dynamic can clearly be seen. The different eigenfrequencies noted between the two radial directions indicates a non-symmetrical mounting of the spindle. This asymmetry was not considered in the model. A deviation of approximately 0-10% between the simulated and measured eigenfrequencies at 0 rpm could be seen in the result. A possible reason for the deviation could be the stiff connection between dummy-tool, holder and spindle in the model. The chosen value of the inner mass radius of the spindle and the absence of a modeled housing could also be a source of this deviation.

Further studies will include the conversion of CDST measurements from a dummy tool setup to a real cutting tool setup. The study performed indicates that predictions of high speed milling stability based on 0 rpm tap-test can be difficult due to speed dependent system dynamic.

6 ACKNOWLEDGEMENT

The study was financed by The Swedish Agency for Innovation Systems (Vinnova). SKF Nova is acknowledged for their technical support and for their provision of their developed CDST equipment. The Kempe Foundations is acknowledged for financing the LDV system.
7 NOMENCLATURE

- $F_m(t)$: Magnetic force
- $x_m, y_m$: Instantaneous displacement
- $x_{m0}, y_{m0}$: Magnetic centre offset
- $A_x$: Reference acceleration
- $C_{sm}$: Calibration factor CDST
- $I_{v1}, I_{w1}$: Coil current quadrant $v_1$ and $w_1$
- $I_{v3}, I_{w3}$: Coil current quadrant $v_3$ and $w_3$
- $d$: Effective gap between rotor and magnets
- $\{q\}$: Generalised coordinates
- $R_e$: Outer stiffness radius
- $r_i$: Inner stiffness radius
- $R_m$: Outer mass radius
- $r_m$: Inner mass radius
- $[M_T]$: Element translational mass matrix
- $[M_R]$: Element rotational mass matrix
- $[G]$: Element gyroscopic mass matrix
- $[K]$: Element stiffness matrix
- $\{h\}$: Stat vector
- $\mu$: Element mass per unit length
- $I$: Area moment of inertia of a hollow shaft
- $J_p$: Polar moment of inertia
- $J_d$: Diametral moment of inertia
- $\Phi$: Shear deformation
- $E$: Modulus of elasticity
- $G_s$: Shear modulus
- $\kappa$: Shear deformation factor
- $l$: Element length
- $\nu$: Poisson’s number
- $F_c$: Centrifugal force
- $Q_e$: Contact force outer ring
- $Q_i$: Contact force inner ring
- $\alpha_e$: Contact angle outer ring
- $\alpha_i$: Contact angle inner ring
8.1 System matrices

Translational mass matrix: 
\[
[M_T] = [M_o] + \Phi [M_1] + \Phi^2 [M_2]
\]

Rotational mass matrix: 
\[
[M_R] = [N_o] + \Phi [N_1] + \Phi^2 [N_2]
\]

Mass matrix: 
\[
[M] = [M_T] + [M_R]
\]

Gyroscopic matrix: 
\[
[G] = [G_o] + \Phi [G_1] + \Phi^2 [G_2]
\]

Stiffness matrix: 
\[
[K] = [K_o] + \Phi [K_1]
\]

where

\[
[M_o] = \frac{\mu l}{420(1 + \Phi)^2} \begin{bmatrix}
156 & 0 & -22l & 4l^2 & \text{Sym.} \\
0 & 156 & 0 & 4l^2 & \\
0 & 22l & 54 & 0 & 13l & 156 \\
0 & 0 & 54 & -13l & 0 & 0 & 156 \\
0 & 13l & -3l^2 & 0 & 0 & 22l & 4l^2 \\
-13l & 0 & 0 & -3l^2 & -22l & 0 & 0 & 4l^2
\end{bmatrix}
\]

\[
[M_1] = \frac{\mu l}{420(1 + \Phi)^2} \begin{bmatrix}
294 & 0 & -38.5l & 7l^2 & \text{Sym.} \\
0 & 294 & 0 & 7l^2 & \\
0 & -38.5l & 38.5l & 0 & 0 & 31.5l & 294 \\
0 & 0 & 126 & -31.5l & 0 & 0 & 294 \\
0 & 31.5l & -7l^2 & 0 & 0 & 38.5l & 7l^2 \\
-31.5l & 0 & 0 & -7l^2 & -38.5l & 0 & 0 & 7l^2
\end{bmatrix}
\]

\[
[M_2] = \frac{\mu l}{420(1 + \Phi)^2} \begin{bmatrix}
140 & 0 & -17.5l & 3.5l^2 & \text{Sym.} \\
0 & 140 & 0 & 3.5l^2 & \\
0 & -17.5l & 17.5l & 0 & 0 & 140 \\
0 & 0 & 70 & 17.5l & 0 & 0 & 140 \\
0 & 17.5l & -3.5l^2 & 0 & 0 & 17.5l & 3.5l^2 \\
-17.5l & 0 & 0 & -3.5l^2 & -17.5l & 0 & 0 & 3.5l^2
\end{bmatrix}
\]
\[
[N_0] = \frac{j_x}{30(l + \Phi)^2} \\
\begin{bmatrix}
36 & 0 & 36 \\
0 & -3l & 4l^2 \\
3l & 0 & 0 & 4l^2 \\
-36 & 0 & 0 & -3l & 36 \\
0 & -36 & 3l & 0 & 0 & 36 \\
0 & -3l & -l^2 & 0 & 0 & 3l & 4l^2 \\
3l & 0 & 0 & -l^2 & -3l^2 & 0 & 0 & 4l^2 \\
\end{bmatrix}
\]

\[
[N_1] = \frac{j_x}{30(l + \Phi)^2} \\
\begin{bmatrix}
0 & 0 & 15l & 5l^2 \\
-15l & 0 & 0 & 5l^2 \\
0 & 0 & 0 & 15l & 0 \\
0 & 0 & -15l & 0 & 0 & 0 \\
0 & 15l & -5l^2 & 0 & 0 & -15l & 5l^2 \\
-15l & 0 & 0 & -5l^2 & 15l^2 & 0 & 0 & 5l^2 \\
\end{bmatrix}
\]

\[
[N_2] = \frac{j_x}{30(l + \Phi)^2} \\
\begin{bmatrix}
0 & 0 & 10l^2 \\
0 & 0 & 0 & 10l^2 \\
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 5l^2 & 0 & 0 & 0 & 10l^2 \\
0 & 0 & 0 & 5l^2 & 0 & 0 & 0 & 10l^2 \\
\end{bmatrix}
\]

\[
[G_0] = \frac{j_x}{30(l + \Phi)^2} \\
\begin{bmatrix}
0 & 36 \\
-3l & 0 & 0 & Skew \ sym. \\
0 & -3l & 4l^2 & 0 \\
0 & 36 & -3l & 0 & 0 \\
-36 & 0 & 0 & -3l & 36 & 0 \\
-3l & 0 & 0 & l^2 & 3l & 0 \\
0 & -3l & -l^2 & 0 & 0 & 3l & 4l^2 & 0 \\
\end{bmatrix}
\]
\[ [G_1] = \frac{J_p}{30l(1+\Phi)^2} \begin{bmatrix} 0 & 0 & 0 & Skew \ sym. \\ 0 & 15l & 5l^2 & 0 \\ 0 & 0 & 15l & 0 \\ 0 & 0 & 0 & 15l \\ 0 & 15l & 0 & 5l^2 \\ 0 & 0 & 5l^2 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \]

\[ [G_2] = \frac{J_p}{30l(1+\Phi)^2} \begin{bmatrix} 0 & 0 & 0 & Skew \ sym. \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 5l^2 & 0 \\ 0 & 0 & 5l^2 & 0 \end{bmatrix} \]

\[ [K_3] = \frac{EI}{l^3} \begin{bmatrix} 12 & 0 & 0 & Sym. \\ 0 & 12 & 0 & 4l^2 \\ 0 & 0 & 6l & 4l^2 \\ 0 & 0 & 0 & 6l \\ -12 & 0 & 0 & 12 \\ 6l & 0 & 0 & 12 \\ 0 & -6l & 6l & 0 \\ 0 & 2l^2 & 0 & 6l \end{bmatrix} \]

\[ [K_4] = \frac{EI}{l^5} \begin{bmatrix} 0 & 0 & 0 & Sym. \\ 0 & 0 & 0 & 1^2 \\ 0 & 0 & 0 & 1^2 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \]
9 REFERENCES


