Analysis of Non-Linear Machine Tool Dynamic Behaviour

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ABSTRACT

This paper presents a Finite Element Analysis of a high speed spindle. At high speeds, bearing preload is reduced automatically by a hydraulic mechanism in the analysed machining centre. The finite element model can consider variable bearing preload effect and other factors such as gyroscopic moments, centrifugal forces, and thermal effects which affect the dynamic response. ANSYS and SpindlePro were used to simulate the spindle model. The effect of preload on stiffness, displacement and frequency response function (FRF) was analysed. A simplified thermal model was simulated in SpindlePro, where thermal effects on the bearings preload with back to back arrangement were observed. It was found that the gyroscopic moment is effective at lower damping ratios and stiffness. The variable bearing preload showed the most significant influence on the spindle dynamic response. FRF was measured for different preload scenarios and stability lobes were calculated to analyse the effects of variable preload.

Keywords: Speed; preload; gyroscopic; thermal; stability.

1. Introduction

One major limiting factor for increasing the metal removal rates of machine tools is chatter phenomenon. Chatter occurs due to a self-excitation mechanism between the cutting forces and chip thickness during machining operations [1]. It results in low surface quality by leaving chatter marks on the surface. It also has an adverse effect on cutting tool and machine tool life.

Chatter can be avoided by selection of process parameters according to the stability diagram. In calculation of stability diagrams, the Frequency Response Function (FRF), measured using tap testing at the tool tip, is required. The measurement is made in static conditions. However, it has been shown that dynamics of the structure may change under rotating conditions [2]. In this case, stability diagram's predictions will be erroneous. In order to consider spindle speed dependence of the FRF, measurement of the FRF at rotating conditions can be made, which is quite a challenging task [3]. Alternatively, Finite Element Analysis (FEA) models can be implemented to predict the FRF in rotating conditions. The models must consider several aspects for high speeds that differ from static state. In literature, it is presented that at high speeds, gyroscopic moments, centrifugal forces, variable preload on the bearings and thermal effects play an important role in the dynamics of the system.

In this paper, a spindle of a 5 axis CNC machining centre StarragHeckert ZT-1000, which has maximum spindle speed of 24000 rpm, is analysed. The machine tool has a variable preload mechanism which automatically adjusts the preload according to the spindle speed. In previous work, experimental tests were performed to characterize the dynamic response of the system under different preload settings. Effect of variable preload on dynamic response was analysed. Those experiment results are presented in [3].

A finite element (FE) model of this CNC machining centre is developed to compare against the experimental data in the paper. In the following sections the FE techniques used in ANSYS and SpindlePro [4] for modelling the spindle are presented. In Section 2, the causes of change in FRF are described. The spindle properties are detailed in Section 3. Section 4 and 5 describe the models in ANSYS and SpindlePro respectively. The simulations results and comparisons with experiments are discussed in Section 6. Finally the conclusions are presented in Section 7.
2. The Causes of Change in FRF

2.1 Centrifugal Forces
When spindle speed is increased, bearings exhibit nonlinear behaviour due to changes in stiffness. In literature [5] and [6], it is shown that the resonant frequencies of a high speed spindle system can decrease due to centrifugal forces at high spindle speeds. This is explained as the “bearing softening” effect. In general, the centrifugal forces affect the bearing contact angles. When the spindle is static, the contact angle at inner and outer ring are identical. However, when the spindle is rotating at high speeds, the centrifugal forces modify the contact angles, increasing at the inner ring and decreasing at the outer ring. This change in contact angles modifies the contact loads and consequently the bearing stiffness.

2.2 Gyroscopic Effects
The gyroscopic moment causes resonant frequencies split into two frequencies, one increasing with rotational speed and the other decreasing with rotational speed [7]. This effect is caused by the whirl sense of vibration when the spindle is rotating. However, in [6], Cao noticed that the frequency split by gyroscopic moment was not usually detectable in the experiment measurements. It was observed that increasing or decreasing the damping ratio has a direct influence on this phenomenon. The common damping ratio present at machines (3-5%) makes less evident the FRF split, while lower damping ratios (1%) increase the FRF split. Movahhedy and Mosaddegh [8] performed a study on the effect of gyroscopic moment over stability limits. This analysis shows that gyroscopic moment reduces the stability limits at high spindle speeds.

2.3 Thermal Effects
The heat generated by friction when bearing rotates at high speeds causes thermal expansion. There is also an increase of heat generated by the motor due to higher input power. As a result the bearing preload is affected, modifying the lubricants and shaft-housing areas as well [9]. The thermal load will change the heat generation at the bearing contact locations, affecting the bearing stiffness and hence the dynamic response [10]. Holkup and Kolář [11] present a thermo-mechanical model where the stiffness differs nearly 300% with the stiffness resulting from the thermo-mechanical simulation after an eight-hour run. It was also observed that the thermal effect is directly dependent on the bearing configuration used.

2.4 Bearing Preload
Several studies about spindle models have been developed to optimize the bearing preload [12-14]. From the model of Altintas and Cao [13], it can be observed that an increase in bearing preload leads to an increase in bearing stiffness and as a result the natural frequency increases. The experiment results, previously presented in [3], show the variable preload effects on the tool tip FRF. The preload variations modify the natural frequencies, stiffness and damping ratios in a different manner for each mode. It is clear that bearing stiffness and natural frequencies are speed dependent. However, due to different spindle configurations, it is not possible to make generalizations about spindle dynamic behaviour and further research of speed effect on FRF and stability of the systems is necessary.

3. System Elements
In Figure 1, a photo of the tool holder and tool, CAD model of the spindle system and bearing arrangement are illustrated. The spindle model analysed in the present work is the Step Tec 6022. Four hybrid bearings (ceramic balls of silicon nitride) with a contact angle of 25° are used to house the spindle. The two front bearings are in tandem arrangement with a SNFA code, VEX 65H1/NS7CE3 T. The other two bearings, at the rear section, are also in tandem arrangement with a SNFA code VEX 60H1/NS7CE3 T. Between the front and the rear bearings a tandem “O” (back to back) arrangement is formed.

The tool holder is a BILZ Thermogrip T2000-100 shrink-fit holder with an HSK-A63 spindle interface. The tool is a SANDVIK R216.32-20025-AP20A H10F with 20 mm diameter and two flutes. The StarragHeckert ZT-1000 has a hydraulic preload system which adjusts the bearing preload automatically according to the spindle speed. This mechanism is described in [3].
Figure 1. Spindle system: Tool holder and tool, CAD model and bearing arrangement

4. ANSYS Model

Using ANSYS APDL, a static model was built, consisting of a solid shaft rigidly connected to the tool and tool-holder and supported by the bearings. The spindle is modelled as a stationary reference frame where the general dynamic equation includes the rotational effects as follows [15]:

\[
[M][\ddot{u}] + ([G] + [C])\dot{[u]} + ([B] + [K])u = [F]
\]

Where:

- \([M]\) = Structural mass matrix
- \([G]\) = Gyroscopic effect originated from the rotational angular velocity applied to the structure
- \([C]\) = Damping matrix
- \([B]\) = Rotating damping matrix
- \([K]\) = Stiffness matrix
- \([F]\) = External force vector

The shaft, tool and tool holder are modelled using PIPE16 elements. The shaft is supported by COMBI214 elements, which simulate the bearings (Figure 2). PIPE16 is a six degree element based in Timoshenko beam theory. It is a tension-compression, torsion and bending element. COMBI214 is a 2D spring-damper element with tension-compression capabilities defined by stiffness, \(K_{11}, K_{22}, K_{12}\) and \(K_{21}\), and damping constants, \(C_{11}, C_{22}, C_{12}\) and \(C_{21}\).

Figure 2. FE model in ANSYS of the spindle system

4.1 Bearing Stiffness

Several tests were performed varying the bearings stiffness values. In Table 1, different stiffness values used in the tests are shown, where \(P_r\) is the preload and \(R_a\) is the axial stiffness. Relation between preload and axial stiffness is provided by SNFA General Catalogue [16].

<table>
<thead>
<tr>
<th>SNFA</th>
<th>Light preload</th>
<th>Medium preload</th>
<th>Heavy preload</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(P_r) (N)</td>
<td>(R_a) (N/µm)</td>
<td>(P_r) (N)</td>
</tr>
<tr>
<td>VEX 60</td>
<td>150</td>
<td>142</td>
<td>440</td>
</tr>
<tr>
<td>VEX 65</td>
<td>170</td>
<td>152</td>
<td>520</td>
</tr>
</tbody>
</table>

The radial stiffness \(R_r\) is dependent on contact angle and preload. Radial stiffness for an angular contact ball bearing with 25° contact angle can be empirically calculated as follows [16]:

\[
R_r = 2 R_a
\]
4.2 Damping
A sensitivity analysis with different damping values was performed to the spindle model using Rayleigh Damping (proportional damping) and common damping for each mode.

Rayleigh damping is defined by $\alpha$ and $\beta$ constants. The damping matrix $[C]$ is calculated by using these constants to multiply the mass matrix $[M]$ and stiffness matrix $[K]$:

$$[C] = \alpha[M] + \beta[K]$$

(3)

Assuming that $\alpha$ damping may be ignored (for practical structures analysis [15]), for the spindle model, $\beta$ damping constant can be calculated as follows:

$$\beta = 2 \xi_i \omega_i$$

(4)

Where $\omega_i$ is the natural frequency of mode $i$ and $\xi_i$ the modal damping ratio. Experimental modal damping ratios and natural frequencies from the StarragHeckert ZT-1000 machine centre [3] were used to find a suitable damping value for the spindle simulation.

4.3 Boundary Conditions
A full harmonic analysis was performed to obtain the FRF of the spindle. A force of 500 N with 0 phase was applied at the tool tip in Y direction. The analysis covered a frequency range from 0 to 3500 Hz with 10Hz intervals. The traction and torsion were suppressed by restricting displacement and rotation in X direction. The second nodes of the bearings are fully constrained acting like the housing. The dynamic model considers inertia effects using CORIOLIS and OMEGA commands.

5 SpindlePro Model
Figure 3(a) shows the spindle mechanical model in SpindlePro [4]. The bearings were defined by inside and outside diameter, ball diameter, Young’s modulus and Poisson ratio of balls and rings, contact angle, number and density of balls and preload. The preload is applied in the central area (Figure 3(a)). Assuming that preload is distributed evenly among the bearings (by the spacers), the bearing preload of the bigger bearings (VEX 65) from SNFA catalogue is applied, simulating three scenarios: light preload (340 N), medium preload (1040 N) and heavy preload (2080 N).

5.1 Thermal Model
A simplified thermal model was simulated in SpindlePro [4] to observe the thermal effects in a spindle with bearings in “O” arrangement. Figure 3(b) shows the simplified thermal model proposed by Holkup [17]. This model consists of a solid shaft of 100mm diameter, a housing with an inside diameter of 150mm and 200mm outside diameter, and two bearings in “O” arrangement with code number HCB7020E.T.P4S and rigid preload of 340N. In both models, mechanical and thermal, an impact load at node 1 in Y direction of 722 N was applied at 0.0164 seconds.

6 Simulations and Comparisons with Experiments
The simulations results are discussed in this section. Both models, ANSYS and SpindlePro, are compared with experiment results.

6.1 Variable Preload
In Figure 4 the FRF in Y direction is shown for the different preload values. The FRF was calculated at the tool tip according the variation in bearing stiffness with light, medium and heavy preload (Table 1). The
FRF was obtained in ANSYS from the ratio of the nodal displacement \( (U_y) \) and the force introduced to the system \( (F_y) \), both in Y direction:

\[
FRF(\omega) = \frac{U_y}{F_y}
\]  

(5)

Figure 4. FE prediction in ANSYS of tool tip FRF in Y direction at different preload values

The preload pressure seems to have different effect on each mode. An increase in FRF magnitude is observed for the first mode when preload is increased. However, for the second and third modes an opposite behaviour is observed, FRF magnitude decreases when preload is increased. For all modes, the natural frequencies are increased when preload is increased.

6.2 Gyroscopic Effect

A damped modal analysis was simulated with speeds of 1000, 10000, 20000, 50000, 100000 and 150000 rpm. The FRF in Y direction with light preload and 7% of damping is plotted in Figure 5(a). Two scenarios are presented, at 1000 and 100000 rpm. In this figure, no important changes due to gyroscopic moment can be observed even at very high speed (100000 rpm). Another test was performed using a damping ratio of 3%. Similar situation where no significant variations due to gyroscopic moment was observed. The FRF in Y direction using a lower stiffness, detailed in Table 2, is illustrated at Figure 5(b), a damping ratio of 3% and speeds of 1000, 50000 and 100000 were used. The frequency split is observed for the three last modes. In Figure 5(c), the FRF with 3% and 5% of damping ratio are compared at 100000 rpm. The gyroscopic effect is clearly diminished at higher damping ratios.

Table 2. Lower bearing stiffness values

<table>
<thead>
<tr>
<th></th>
<th>( Ra ) (N/( \mu )m)</th>
<th>( Rr ) (N/( \mu )m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frontal bearings</td>
<td>15.2</td>
<td>30.4</td>
</tr>
<tr>
<td>Rear bearings</td>
<td>14.2</td>
<td>28.4</td>
</tr>
</tbody>
</table>

Figure 5. FRF in Y direction at different speeds and damping ratios: (a) 7% of damping ratio; (b) 3% of damping ratio; (c) 5% and 3% of damping ratio at 100,000 rpm

It was seen that the bearing stiffness and damping ratio has a great influence on gyroscopic effect for this particular spindle. At higher stiffness and higher damping ratios the gyroscopic effect is diminished. Damping ratios observed in experiments from 2 to 8.7% [3], and bearing stiffness of StarragHeckert machine causes the gyroscopic effect to be practically vanished. A slightly variation at the higher mode
was observed at very high speeds, but at the spindle speed operational range (0-24000rpm) there is no variation observed due to gyroscopic effect in the FRFs.

6.3 Thermal Effects
Thermal effects for the simplified case are illustrated in Figure 6. This simplified model in SpindlePro couples mechanical and thermal effects considering operational parameters specified by duration, speed and power. Figure 6(a) shows the bearing stiffness at 2000rpm after 3600sec running the spindle. A stiffness value of 2.9*10^7 N/m is found when considering thermal effects. This stiffness value is increased to 7.6*10^7 N/m when thermal effects are not considered. After the transient response of the system due to the rigid preload applied, the impact load causes the perturbations observed around 0.0164 seconds. At Figure 6(b), the effect of the gradual warm up at 1000, 2000 and 3000 seconds on real part of the FRF is illustrated together with the case without thermal effects. It can be seen that the natural frequencies are reduced gradually with spindle warming up.

![Figure 6. Thermal effects in simplified thermal model: (a) bearing stiffness in axial direction; (b) FRF at spindle nose without thermal effects and at different running times with thermal effects](image)

The thermal effects in this spindle with bearings in “O” arrangement tend to reduce the pressure instead of increase it. A reduction in stiffness and natural frequencies was observed when considering thermal effects. However, the hydraulic preload mechanism of the StarragHeckert machine tool seems to compensate the preload variations by thermal effects as has been observed in similar machine tools with hydraulic preload systems [14]. Nevertheless, more detailed analysis needs to be done specially for the thermal model. Housing information and the complete model with the four bearing arrangement need to be considered for this analysis.

6.4 Experimental Tests, ANSYS and SpindlePro Models Comparison
The tool tip FRFs from ANSYS, SpindlePro and experiment measurements by Ozturk et al. [3] are provided in Figure 7. FRFs with medium preload from ANSYS and SpindlePro are presented. The medium preload scenario (35 bar) from experiment measurements is shown. The modal parameters from Figure 7 are extracted and illustrated in Figure 8 for comparison purposes.

![Figure 7. Tool tip FRF in ANSYS, SpindlePro and experiments by Ozturk et al. [3]](image)

In general, the FE models have a reasonable match with the experiments. The natural frequencies and bearing stiffness are very similar for the first two modes. SpindlePro has better frequency and stiffness prediction for the third mode than ANSYS model, where it is almost vanished. The difference between
FE models and experiments could be due to different damping ratios simulated. Damping ratio values are mode dependent in experiment results. From the sensitivity analysis in ANSYS, a better match was observed when common damping was applied (7%). However, the 7% damping ratio is too high for the second and third modes compared to the experimental ones and reduces notably the FRF magnitude. In addition to the damping ratio variations, the bearing definition is different in ANSYS (by stiffness) and SpindlePro (by preload) models. This discrepancy modifies the spindle stiffness and natural frequencies and a further study is necessary to evaluate the different bearing preload scenarios. Similar behaviour is observed for the light and heavy preload scenarios. Despite the limitations in damping ratio and bearings definitions, FE models present reasonable agreement with experimental measurements and are helpful to simulate spindle behaviour, providing a better understanding of speed effects for machine tools with variable preload mechanisms.

![Figure 8. Modal parameters comparison with medium preload: (a) natural frequencies; (b) stiffness; (c) damping ratio](image)

### 6.5 Stability Lobes
Stability lobes were calculated using CutPro™ software for an example process of a down milling operation (Figure 9 (a)). The radial depth of cut used was 10mm, feed-rate of 0.1 mm/tooth and clockwise spindle direction. Carbide tool with 20 mm in diameter, two flutes with uniform pitch and helix angle of 30° was simulated. The cutting material was an AL-6061-T6. As the bearing preload is the parameter that showed the greatest influence on the FRF for this particular spindle, stability lobes were calculated using FRF with light, medium and heavy preload. Figure 9 (b) shows the magnitude variations in FRF for each mode according to the different preload values.

![Figure 9. (a) Stability lobes; (b) ANSYS prediction of tool tip FRF magnitude at different preload values](image)

The first mode in the FRF is the most flexible and highest magnitude is predicted with the Heavy Preload case. This does not agree with the measurements presented in [3] possibly because of the differences in damping ratio and bearing definition previously stated which causes an error in magnitude of FE prediction. Because of this mismatch, the absolute stability limits for the heavy preload case is the minimum compared to other preload cases (Figure 9(a)).

### 7 Conclusions
A FE spindle model was simulated in ANSYS and SpindlePro. Spindle speed effects were considered such as variable preload and gyroscopic moment. These effects were analysed separately on spindle stiffness, natural frequencies and damping ratios. FE models have a reasonable agreement with the experiments in natural frequency predictions; however, further improvements are necessary for stiffness and damping predictions to calculate the magnitudes with a better accuracy. It can be concluded that for the practical scenario that was considered, the high damping ratio and bearing stiffness of the spindle
reduce the effects caused by gyroscopic moment. To demonstrate this, tool tip FRFs were presented where gyroscopic effect was clearly observed for a case with a lower stiffness and lower damping ratio. The variable preload of the StarragHeckert ZT-1000 machine centre seems to compensate preload variations due to thermal effects. Consequently, the speed effect on stability lobes is observed by using FRFs with preload variations, where stability region is modified with preload variations. Furthermore, a simplified thermal model was simulated in SpindlePro. This model shows that the bearing stiffness is reduced due to thermal effects for bearings with “O” arrangement; therefore natural frequencies are reduced as well. However further analysis is required to perform an accurate thermal analysis. Housing information and the complete model with the four bearing arrangement need to be considered. Finally an update of the structural and thermal model is necessary to predict the dynamic response of the spindle considering thermal effects.

8 Acknowledgments

To StarragHeckert AG for supplying the relation between the preload and the spindle speed on the StarragHeckert ZT-1000 machining centre and collaboration of Tomáš Holkup, Murat Kilic and Hongrui Cao in the solution of thermal model in SpindlePro.

9 References