ISSN: 0974-5823 Vol. 7 No. 1 January, 2022

International Journal of Mechanical Engineering

Design and analysis of milling tool spindle from experimental data: An inverse approach

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ABSTRACT

An end milling process's chatter stability analysis is complicated due to a lack of exact information of the spindle's geometrical design, the location of the bearings, and other spindle structural concerns. Self-excited chatter vibrations can only be studied with an exact transfer function in the most flexible area of the spindle tool structure, which an efficient spindle needs design. A new approach for assessing spindle vibration responses using sine sweep tests for a CNC end milling machine tool is the topic of this research. When the bearing span is taken into account as a design variable, the spindle's analytical model may be approximated. There are a number of trial runs until the model and experiment transfer functions are in agreement. To obtain the process stability, the time varying forces are evaluated at the different cutting conditions.

Keywords*:* Self-excited chatter vibrations; Geometrical design; design variable; transfer functions and tool over hang (TO*).*

1. INTRODUCTION

A wide range of industries, including aerospace, automotive, mould and die, and more, still rely on vertical CNC milling for high-speed machining. It is possible to improve process efficiency via optimization, but machine tool chatter is the most important factor in the process's efficacy. Uncertainty in the cutting process produces excessive material removal, low surface area, and nearly guaranteed tool and workpiece damage. An accurate dynamic model of the tool-holder-spindle assembly is needed to choose the elements that minimize chatter and improve surface smoothness. In order to get this dynamic at the tool tip, modal testing may be used, but it requires a large number of tool-holder configurations in a production plant. In the case of micro-end mills, the measurements are time-consuming and sometimes troublesome. Analytical and experimental research on spindle modeling utilizing modal test data have been described in a limited number of papers. An end mill's self-excited vibrations may be inversely studied to determine the transfer function (TF), according to a study by Suzuki et al. [1] developed a transfer function to minimize the percentage of error between the numerical and experimental simulations. To account for the rotor dynamic effects, Gagnol et al. [2] employed a new finite element modeling technique for the novelistic spindle bearing system to characterize the spindle's modal variations. In order to verify the models' correctness, they were first tested. Spindle rotational precision may be quantitatively simulated by Kim et al. [4] using statistical models and software. With increasing the spindle's speed, the spindle's rotational precision might vary dramatically. In addition, bearing preload has a significant influence on performance. To determine the spindle unit's speed-varying dynamics and the accompanying stability lobes, Cao et al. [4] used an alternative technique. In high-speed milling processes, the Nyquist stability criterion is used to assess the stability of the milling dynamics. Finite element modeling including the thermal characteristics for the components of the integrated spindle tool system is provided by Zahedi and Movahhedy [5]. Six-degree-of-freedom Timoshenko beam components were applied to the spindle housing and shaft. When designing a flat end mill geometry, Flutes may be thought of as helicoidal surfaces, and a flute's shank can be considered a rotating surface in three dimensions, according to Tandon and Khan [6]. Finite-element modeling of milling spindle lateral vibrations and the corresponding displacements are measured and it can be used to analyze these vibrations, according to Rantatalo et al. (7). Sarhan and Matsubara employed different sensors at various locations on the spindle tool to obtain the displacements and the corresponding stiffness in the radial direction in order to accurately monitor cutting forces during end milling. Kolar et al.[9] investigated the cutting dynamics of the whole machine tool spindle system, including the machine tool frame. The attached model shows how the spindle and tool system's dynamic characteristics alter in proportion to the machine frame's attributes. [9] For industrial engineers, Cao et al. have updated the FE model so that it can better predict the dynamic response of the spindle tool system. Operating mode analysis was advocated by Zaghbani and Songmene [10] to improve modal characteristics and correct performance metric estimates in various machining scenarios. This was followed by an experiment wherein we used the dynamic process parameters to draw stability lobes. Zivkovic et al.[12] presented a thermo mechanical spindle model and it was relied on the ball bearing contact bearings. Temperature affects the stiffness of a spindle in a non-linear way, and this is tested and validated. Various numerical improvements in the simulation, machining mistakes in curved surfaces, tool indentation effects and vibrations created in inclined and circular surfaces have all been studied extensively. Preload influence on spindle and tool tip frequency responses was presented by Ozturk [13]. An explanation of the spindle's preload-to-spindle-speed relationship was given. Spindle unit modal analysis was performed in the no-run situation in all of the preceding studies. Accordingly, the dynamic modal parameters of the cutting process will change as a result of the different cutting circumstances. The modal properties of machining at different process parameters are estimated in this study. Spindle bearings on the front and back of the spindle unit are evaluated in the study of the complete spindle unit. There are two ways to estimate the modal parameters: first, using Timoshenko beam theory and second, using shear deformation effects. The dynamic state modal parameters are discovered via a series of experiments. Other parts were ordered as follows: (2) Spindle assembly modeling (3) Results and discussion and then (4) Conclusions

2. MODELING OF SPINDLE ASSEMBLY

High-precision applications need accurate tool dynamics identification. Input forces and output responses are often measured using experimental modal analysis. However, it can only be used if the machine tool is not in use at the time of application. During a cutting operation, the work table sliding locations affect the tool structure's dynamics. Previous studies have indicated that the machine tool's structural dynamics change as a function of the tool's location or spindle speed. Many experiments are carried out due to this. For this reason, data collected during a static condition differs from that collected during a cutting process. The calculation of dynamic spindle structural characteristics with work table cutting motion on spindle dynamics is one of the research's outstanding questions. It is possible to accurately measure the spindle tool response with the angular contact bearings with proper constrained system as indicated in the Figure 1. Timoshenko beam components with shear deformation and rotational inertia effects are used to descrtized all the sections of the integrated spindle tool unit.

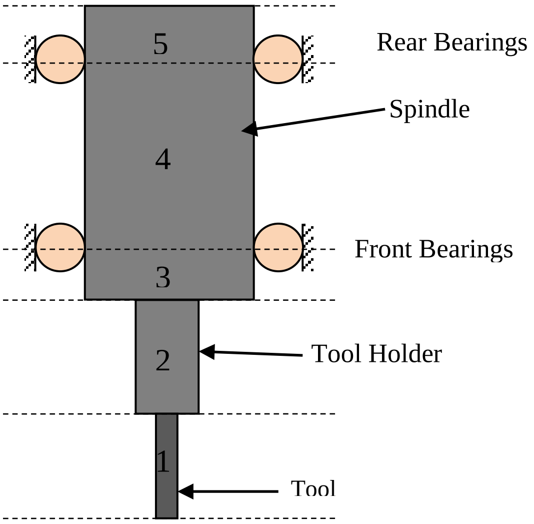


Figure 1. Finite element beam model of the spindle tool unit

In the present analysis, two bending deflections in y and z and one linear deflection(i.e x direction) are included in the total of 24 degrees of freedom for the overall spindle unit. Two angular contact bearings are assumed to support the spindle unit's two nodes. Figure 2 shows the position of the entire spindle tool system with the spring connections as well as the bearing connections.

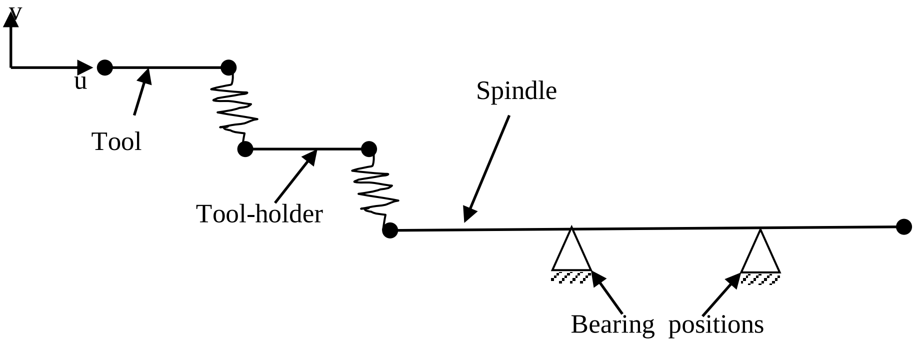


Figure 2. Line model of the spindle-tool unit with spring and bearing connections

Also, as with the spindle shaft, the tool holder is attached to the tool. The following is a description of the unit's finite element model:

$$[M_b]\{\ddot{q}\}+\left[[C_b]-\Omega[G_b]\right]\{\dot{q}\}+\left([K_b]-\Omega^2[M_{cb}]\right)\{q\}=F(t) \tag{1}$$

Here, $[M_b]$, $[K_b]$ and q$[M_{cb}]$ are the combined mass, viscous damping, and stiffness $[K_b]$ matrixes for the beam element, is the rotation speed, while $[G_b]$ represents the gyroscopic matrix and the term 2$[M_{cb}]$q denotes the softening impact of spring forces. There are a number of factors that affect angular contact bearings' stiffness. Bearings with radial stiffness (Static Radial Stiffness) may be assessed using several empirical equations, such as the ball diameter (D_b), the axial preload (F_a), the contact angle of static angular-contact ball bearing, and the number of balls (N_b).

$$k_{xx}= k_{yy}=1.77236\times10^{7}\times (N_b^{2}.D_b)^{1/3}\frac{\cos^2\theta}{\sin^{1/3}\theta}F_a^{1/3} \quad \text{N/m} \tag{2}$$

2.2 Cutting dynamics

Using a well-known mechanical model for milling, chatter is generated by the cutting force. Figure 3 shows various instances of milling cutters by considering a two degree of freedom model with a zero helix angle and N_t is the number of teeth. The work piece is treated as the rigid in nature and the cutting tool is considered as the flexible in nature. Vibration is caused by the whole amount of cutting power.

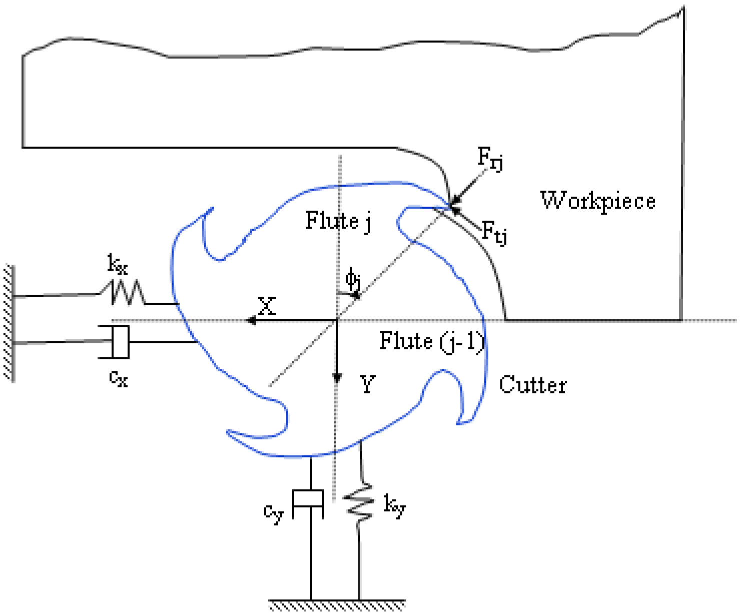


Figure 3. Two-degree of freedom milling model

The radial (n) thickness of the chip is used to represent the variable chip thickness

$$h(\phi_j)=\left(f_t \sin(\phi_j)+n_{j-1}-n_j\right)g(\phi_j) \tag{3}$$

The Eigen value equation for chatter frequency ω_c, static cutting factors (K_t, K_r), radial immersion angles (ϕ_s and ϕ_e) and the frequency response function of the structure are worked out using the formula. By means of,

$$\kappa=\frac{\Lambda_{Im}}{\Lambda_{Re}}=\frac{\sin\omega_c\tau}{1-\cos\omega_c\tau} \tag{4}$$

The analytical stability limit for the predicted frequency responses are arrived as follows:

$$b_{lim}=-\frac{2\pi}{N_t K_t}\Lambda_{Re}(1+\kappa^2) \tag{5}$$

The following tooth passages have a phase change of $\varepsilon=\pi-2\psi$. Next, the tooth passing times are stated as $\tau=\frac{1}{\omega_c}(\varepsilon+j2\pi)$, here j represents integer and is the lobe number for each tooth. All that's left to do is plot the stability lobe between the spindle speeds and the graph.

$$\Omega=\frac{60}{N_t\tau}\ (\text{rpm}) \tag{6}$$

3. RESULTS AND DISCUSSIONS

The experimental modal analysis is verified using a full-order finite element technique. For this experiment, the settings listed in Table-1 were employed.

Table 1 Parameters of the full-order finite element model

	Element of the spindle				
Parameter	**E1**	**E2**	**E3**	**E4**	**E5**
Length(mm)	65	51	111	90	47
diameter(mm)	12	40	75	75	75
E (Pa)	2.35×10^{11}	2.1×10^{11}	2.1×10^{11}	2.1×10^{11}	2.1×10^{11}

The stability barrier is tested using a CNC milling machine with the same spindle. These machines are equipped with three-axis motion with spindles that can spin at up to 4,000rpm. A HSS end mill cutter cutting tool with four edges and with a diameter of 12 mm fits into the tool holder. Aluminum alloys are milled completely. Figure 4 shows the list of components for predicting the vibration response using an accelerometer with a charge amplifier and oscilloscope.

FIGURE 4 Experimental set-up employed

While the transmission's amplitude stays constant, the frequency of the signal is modulated. At each input frequency level, the oscilloscope captures corresponding accelerometer responses from the vibration shaker on the cutting tool. It is possible to see the responses' amplitudes for each frequency in Fig. 5. Natural frequencies in this range are 1910 Hz and 2550 Hz, as you can see.

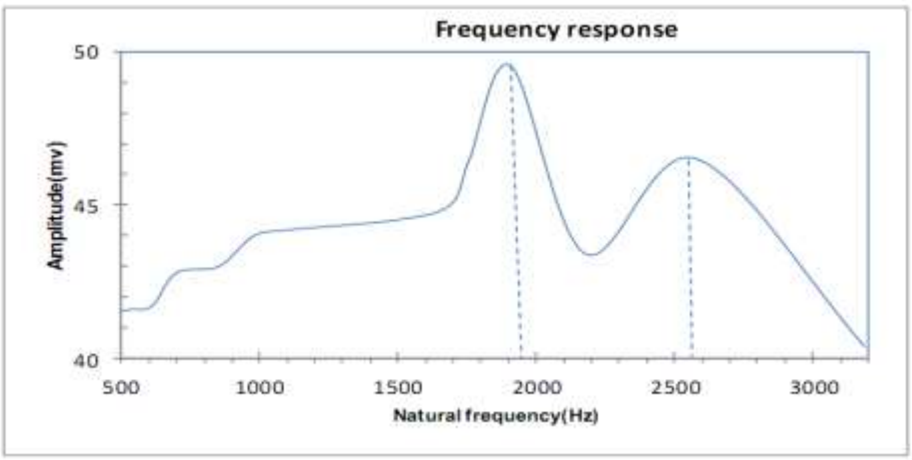
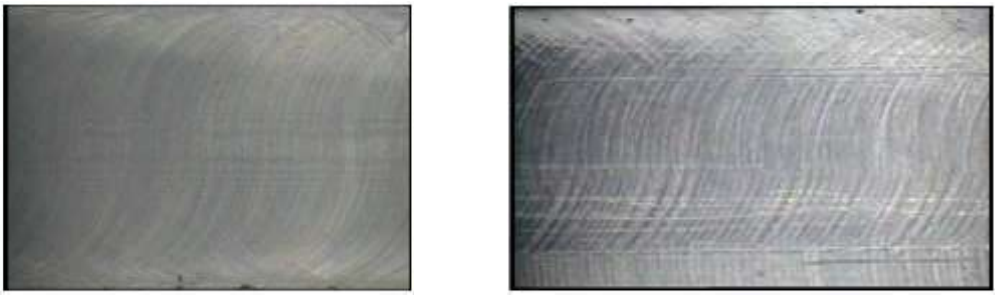


Figure 5. Tool tip frequency response of the spindle-tool unit

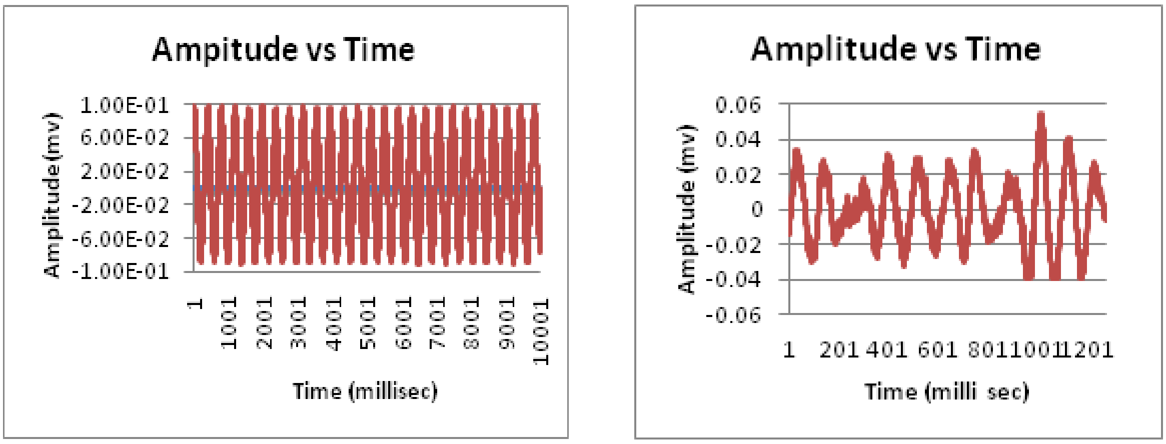
Stability lobe diagrams based on analytical predictions are used to perform experiments at different speeds and depths of cut. Charge amplifier and acclerometer are linked to the non-rotating section of the spindle to measure the cutting tool's vibration response during milling. Using an optical microscope with a magnification factor of 10X is used to acquire the images of the machining areas. Figure. 6 shows the chatter marks on the workpiece material with a cutting speed of 2200rpm and a depth of cut is taken as 0.07mm. The machining reaches the chatter prone zones as indicated in Figure. 6 if cutting surpasses the limit of 0.07mm as indicated in Figure 6(b).



(a) Axial depth of cut 0.07mm (b) Axial depth of cut 0.09mm

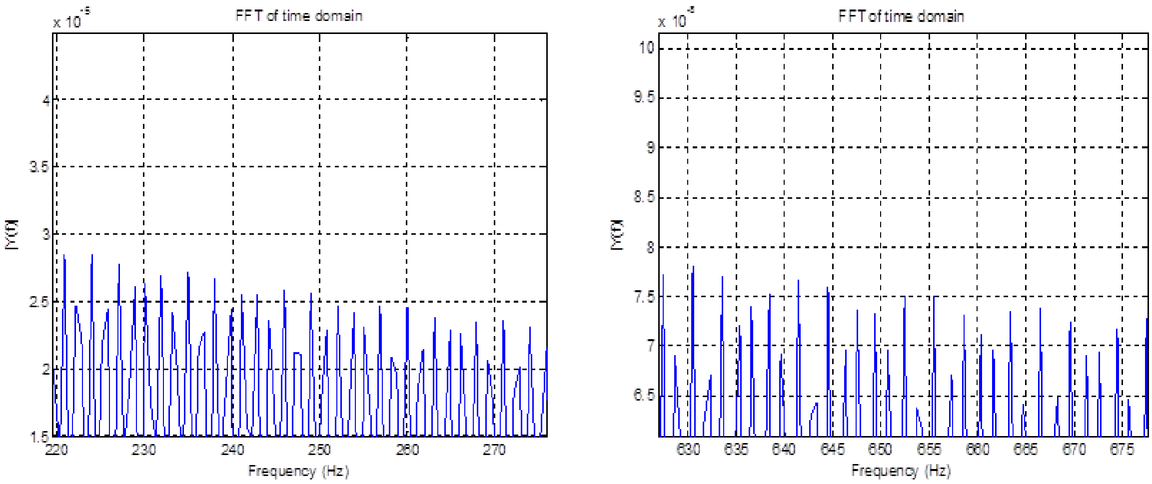
Figure 6. Optical microscopic images of machining areas

The oscilloscope is used to capture the vibrations of the cutting tool in the time domain. Figures (7) and (8) illustrate the amplitudes and FFT graphs for axial depths of 0.07mm and 0.09mm, respectively. Increasing the axail depth of cut results in an increase in tool vibration, which causes chatter marks on the workpiece as a result of this vibration.



(a) Axial depth of cut 0.07mm (b) Axial depth of cut 0.09mm

Figure 7. Tool vibrations at different depths of cut



(a) Axial depth of cut 0.07mm (b) Axial depth of cut 0.09mm

Figure 8. FFT plots at different depths of cut

Analytical stability lobe diagrams are interpolated by considering the tool vibration levels and optical microscopic images of the machining areas. Figure. 9 shows that the theoretically anticipated lobes are giving correct bounds for experimental results.

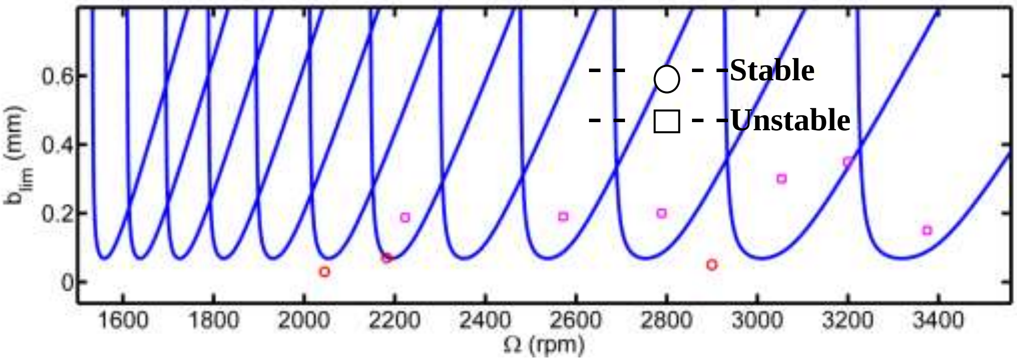
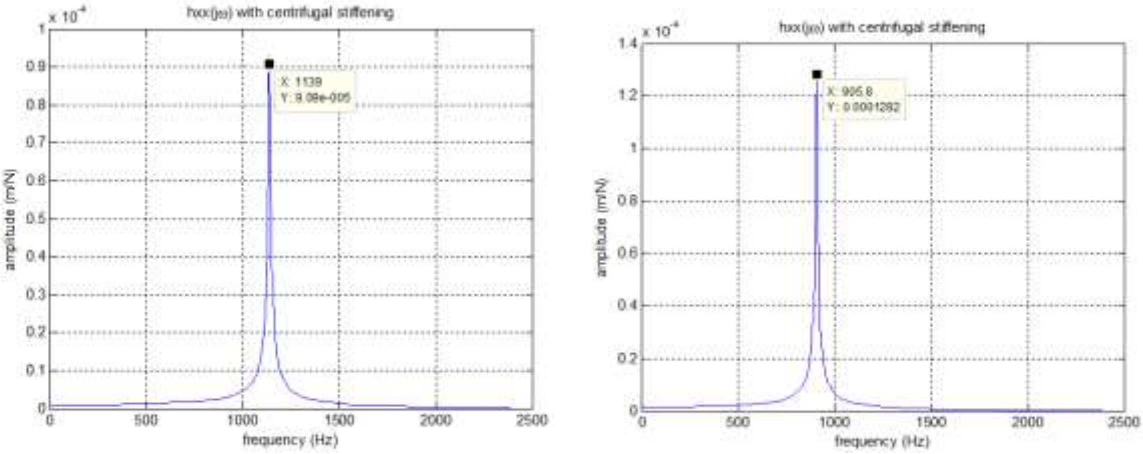


Figure 9. Analytically predicted stabilty lobes at different depths of cut

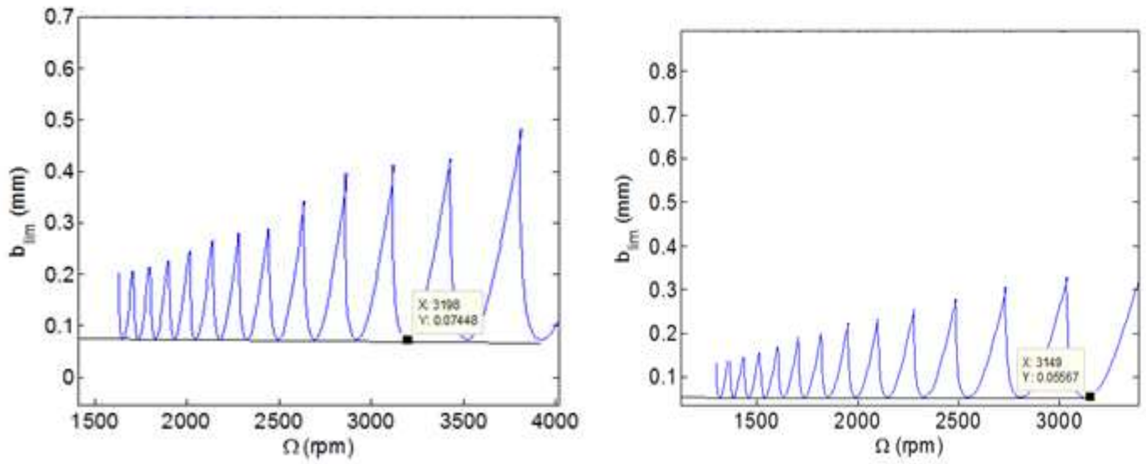
When it comes to determining milling instability, the average axial depth of cut and the dynamic stiffness play a key role in the design of machining process. Many key design factors are associated with these problems, including the size of the spindle shaft and the positions of its housing and bearings, as well as bearing preload, tool overhang, and the cutting tool's helix angle. The stability of slot milling for varying tool overhang lengths is examined in this study. As shown in Figure 10, tool overhang length has a significant impact on system stability. When the cutting tool's overhang length increases, the peak first mode of the natural frequencies shifts to the left side of the amplitude axis.



(a) Tool overhang of 85mm (b) Tool overhang of 95mm

Figure 10. Frequency responses at two overhang lenghts

It is observed that, when the tool overhang length increases there is slight decrement in the average stable depth of cut and it is indicated in the Figure 11. When examining the spindle dynamic system's natural frequency in the first mode has affected the overhang length of the tool and also has the greatest impact on the system's milling dynamics.



(a) Tool overhang of 85mm (b) Tool overhang of 95mm

Figure 10. Stability plots at two different overhang lenghts

4. CONCLUSIONS

Finite element analysis employing Timoshenko beam theory with rotational and shear deformation effects was used to examine a real spindle tool unit in the current study. On the CNC milling centre, a spindle device-tool holder experimental modal analysis is performed to arrive at the tool tip FRF. Trail runs using tool overhang as the design parameter may be used to estimate the modal since isolating the spindle from the housing is difficult. Nonlinear bearing forces as well as cutting forces were taken into account at certain nodes while determining the spindle tool unit's dynamic stability. Both the cutting tests and the numerical simulations, which used the identical applied circumstances, yielded later similar FFT displays.

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