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Design and development of a lathe spindle with adjustable stiffness

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Abstract

The autonomous operation of machining systems is set to transform the future of manufacturing. However, most current machine tools lack the ability to adjust key structural characteristics, such as stiffness and damping, limiting their adaptability to varying machining conditions. To overcome this limitation, a lathe spindle with adjustable stiffness has been developed. This adaptability is achieved by controlling the preload of the front bearing node, enabling modifications to the spindle's static and dynamic properties. The primary objective of this advancement is to enable autonomous machining systems to adjust spindle stiffness, thereby compensating for workpiece deflection and mitigating self-excited vibrations during turning operations. This paper presents the spindle design concept, along with finite element analysis of the adjustable stiffness bearing node to support the design rationale, and experimental results from prototype testing, emphasizing both static and dynamic stiffness characteristics.

machine tool; bearings; preload; static stiffness; dynamic stiffness; finite element modeling; autonomous machining

1. Introduction

Autonomous manufacturing is a prominent topic in both scientific research and industrial applications [1]. It involves the development of self-regulating, fully automated production systems that require minimal human intervention. A key characteristic of such systems is their ability to promptly respond to disturbances and autonomously recover, thereby ensuring uninterrupted and efficient production.

The core components of an autonomous production system include machine tools equipped with sensors that monitor the condition of structural elements, cutting tools, and the machining process [2]. In addition, these machines are equipped with control systems that incorporate algorithms to process sensor data, enabling the detection and mitigation of process disturbances, such as the onset of self-excited vibrations [3,4].

The conventional approach to addressing such disturbances involves adjusting process parameters – such as depth of cut, spindle speed, and feed rate – to align with the inherent static and dynamic properties of the machine tool during machining. However, this method often results in suboptimal machining performance.

Increasingly, both in scientific literature and industrial practice, solutions have emerged that focus on modifying the static and dynamic properties of the machine tool itself, rather than adjusting process parameters. This is typically achieved by varying the preload of the spindle bearing nodes, as preload significantly affects the functional properties of the system. Increasing the preload increases stiffness, reduces rolling noise, and improves the precision of shaft guidance. Furthermore, it can extend bearing life by minimizing the slippage of rolling elements. However, excessive preload increases frictional moment, leading to heat generation and a reduction in bearing lifespan [5].

Spindle bearings can be preloaded using two primary methods. The first, known as fixed position preload, is applied by securing the relative position of the bearings, ensuring that the inner and outer rings remain fixed in position during rotation. The second method, constant pressure preload, involves applying axial force to the inner and outer rings using a spring, allowing the outer ring to adjust its position during rotation while maintaining a constant preload force.

Fixed position preload is more suitable for rough machining operations, but it is not recommended for high-speed applications due to excessive temperature rise. In contrast, constant pressure preload is better suited for high-speed operations with light cutting loads, as it mitigates the drop in stiffness observed with increasing rotational speeds. However, due to the wide range of spindle operating speeds and cutting conditions, neither fixed position preload nor constant pressure preload can fully accommodate varying machining requirements. Consequently, an adjustable preload system – providing high preload at low spindle speeds and low preload at high speeds – is widely considered by researchers to be a viable solution.

Adjustable preload systems in machine tool spindles are implemented through various mechanisms, including piezoelectric actuators [6–9], centrifugal preload devices [10–14], electromagnetic actuators [15,16], and shape memory alloys [17]. However, the most adopted solutions utilize hydraulic systems. Jiang and Mao [18] proposed a motorized high-speed spindle supported by two pairs of angular contact ball bearings, preloaded via a hydraulic chamber. The preload level was controlled by adjusting the hydraulic pressure, which caused axial displacement of the outer rings of the rear bearings. This, in turn, caused displacement of the spindle shaft, on which the inner rings of the front bearings were mounted. Consequently, the relative displacement between the inner and outer rings of the front bearings resulted in a change in preload. Based on this design, a method for analyzing variable preload as a function of spindle rotational speed in high-speed machining applications was developed. The study revealed that, compared with the conventional constant pressure preload method, the variable preload approach reduced temperature rise in the bearings at high rotational speeds due to a lower applied preload, thereby contributing to extended bearing service life. Additionally, the variable preload system significantly enhanced the spindle's dynamic stiffness at low rotational speeds due to the application of a higher preload.

A similar solution was presented by Cao and Altintas [19] with the distinction that the belt-driven spindle was analyzed. The study showed that the bearing's static stiffness varied more significantly at lower preload levels compared to higher preload levels as spindle speed increased. In terms of dynamic properties, it was demonstrated that preload had minimal influence on the spindle's first mode shape, while shifting the higher mode (fourth mode) to a higher frequency. These two modes were identified as the most dominant at the spindle tip and had the greatest influence on machining stability.

A motorized spindle for end milling with hydraulic variable preload system, was built by Bossmanns and Tu [20]. The preload in the proposed system consisted of two components: an initial preload provided by a series of springs placed around the circumference of the two front bearings, and an adjustable preload achieved by pressurizing a hydraulic cylinder located between the bearings. The study demonstrated that thermal permissivity – defined as the inverse of thermal contact resistance – decreases significantly at lower preload levels, while showing only a slight increase at higher preload levels, as the contact surfaces become more rigid when they flatten.

An analogous design was presented by Fang *et al*. [21], featuring a spindle equipped with a series of springs to provide initial preload and a built-in hydraulic system to adjust the preload in a controlled manner. The study found that, at low spindle speeds, the bearing temperature increased gradually with rising preload. However, as the rotational speed increased, internal skidding within the bearings intensified at lower preload levels, leading to a rapid temperature rise in the bearing's outer ring. Therefore, at high spindle speeds, an optimal preload that minimizes temperature rise can be achieved, with the optimal preload increasing as spindle speed rises.

Most studies available in the literature focus on the implementation of adjustable preload systems for milling spindles, while this aspect remains largely overlooked in the case of lathe spindles. To address this gap, the present article introduces a variable preload system specifically designed for a lathe spindle. The proposed system utilizes a built-in hydraulic actuator to adjust the preload of the front spindle bearing node. The primary objective of this system is to mitigate self-excited vibrations and compensate for static deflections of the workpiece. However, before these goals can be achieved, it is necessary to determine the spindle's static and dynamic properties at various preload levels, which is documented in this study.

The paper is structured as follows. Section 2 introduces the concept of a lathe spindle with adjustable stiffness. Section 3 presents a finite element model of the system, which was used to evaluate the effectiveness of the proposed concept during the design stage. Section 4 provides the results of experimental tests conducted on the prototype, including static and dynamic stiffness measurements for various preload levels. In Section 5, a detailed discussion of the results is provided. Finally, Section 6 presents the conclusions, summarizing the key findings and contributions of the study.

2. Concept of a lathe spindle with variable stiffness

The concept of a lathe spindle with variable stiffness is illustrated in Figure 1. The system is based on a built-in hydraulic actuator that controls the preload of the front bearing node, thereby enabling adjustments to the spindle's static and dynamic stiffness [22].

The operating principle of the system is as follows. Initial preload is applied during spindle assembly using the fixed position preload method. This is achieved by tightening the lock nut located at the end of the spindle, which moves the inner race of the rear bearing node relative to the spindle shaft. The force is then transmitted through the spacer sleeve, the inner ring of the single bearing in the front bearing node, and finally to the bearing spacer of the front bearing node, which presses against the inner ring of the tandem bearing. As a result, the outer and inner rings of the bearings are aligned, eliminating the stand-out (face/back offset) and securing their position with the lock nut.

To increase the preload of the front bearing node, pressure is applied to the built-in hydraulic system, causing the piston to shift the outer ring of the single bearing. This movement, in turn, causes the inner races of both the single bearing and the tandem bearing to displace, as the force acting on the single bearing is transmitted through the spindle shaft – specifically through the lock nut, which, as previously mentioned, secures the relative position of the spindle shaft and the inner races of the bearings – and subsequently acts on the outer rings of the tandem bearing. Consequently, the preload of the front bearing node increases, leading to a corresponding increase in its stiffness. This method of preload adjustment can be classified as the constant pressure preload method.

It should be noted that, in the analyzed case, the preload distribution among the bearings is not uniform, with the preload applied to the single bearing being twice that applied to each bearing in the tandem arrangement. To better illustrate this phenomenon, Figure 1 includes red arrows indicating the preload (P) distribution.

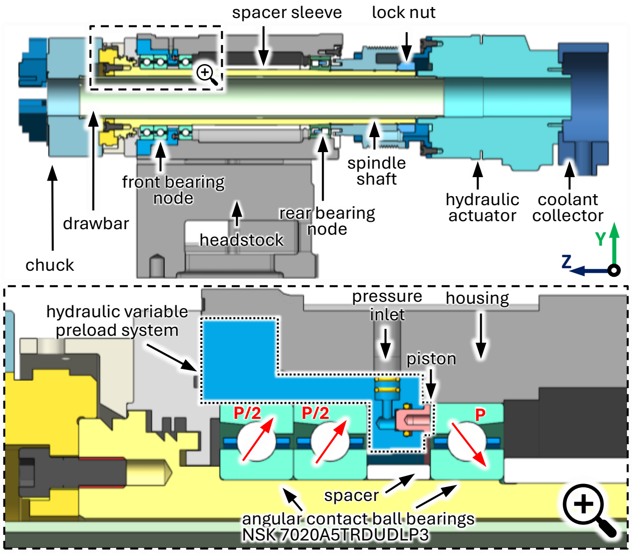

Figure 1. Lathe spindle equipped with variable preload system – general view.

3. Finite element modeling

To evaluate how changes in preload affect the spindle's static stiffness, a finite element model of the adjustable preload system was developed using the Midas 2022 R1 preprocessor (Midas Information Technology Co. Ltd., Seongnam, Korea) [23]. To reduce computation time, only a section of the axisymmetric mechanism was modeled [24]. The width of the modeled section was determined based on Hertzian contact theory [25], which resulted in an assumed width of 0.4 mm. A structured mesh was generated using eight-node cubic isoparametric CHEXA elements and six-node five-walled isoparametric CPENTA elements. These elements were characterized by linear shape functions and three translational degrees of freedom at each node. The final model consisted of 13,988 elements and 22,767 degrees of freedom.

The finite element analysis of the system was conducted in three stages. In the first stage, the assembly of the bearing node was simulated by imposing an initial preload. Since only part of the spindle was modeled, the preload was applied as a relative prescribed displacement between the spacer sleeve (u_1) and the spindle shaft (u_2), omitting other, previously described, components that transfer the force generated by tightening the lock nut. As a result, the preload gap ($2\times f = 2\times b$) was eliminated, and the assembly preload was imposed.

The second stage involved securing the position using the lock nut. In the model, this was achieved by fixing the relative position between the spacer sleeve and the spindle shaft through a welded contact, which replaced the bidirectional sliding contact used in the first stage.

The final stage modeled the preload control process performed by the built-in hydraulic system. This was accomplished by prescribing a displacement on the outer ring of the single bearing (u_3), corresponding to a specific hydraulic pressure. Schematic representation of the analysis stages, including boundary conditions and contact definitions, is shown in Figure 2.

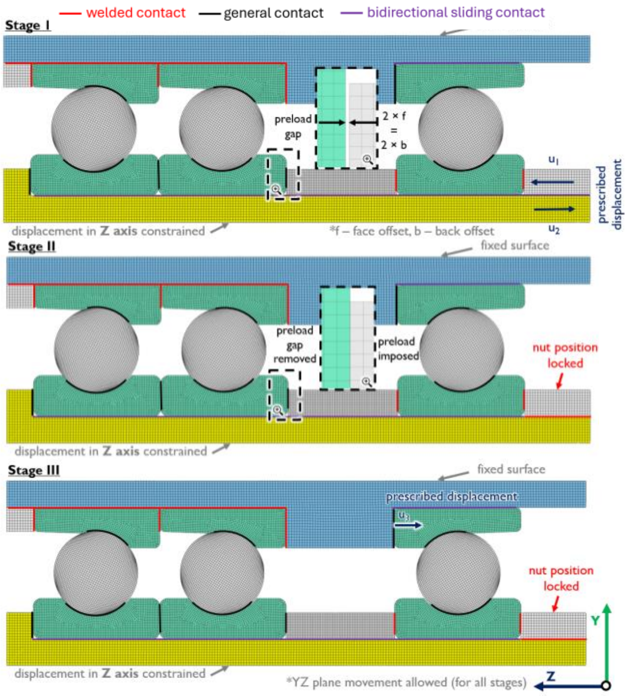

Figure 2. Finite element model of variable preload system.

Subsequently, three scenarios were analyzed using a nonlinear static analysis solver (SOL 106): (i) low preload, with $u_3 = 0$ which corresponds to a pressure equal to 0 bar, (ii) medium preload with u_3 = 25 µm corresponding to a pressure of 25 bar, and (iii) high preload with u_3 = 50 µm which corresponding to a pressure of 45 bar. Exemplary results for medium preload scenario are presented in Figure 3.

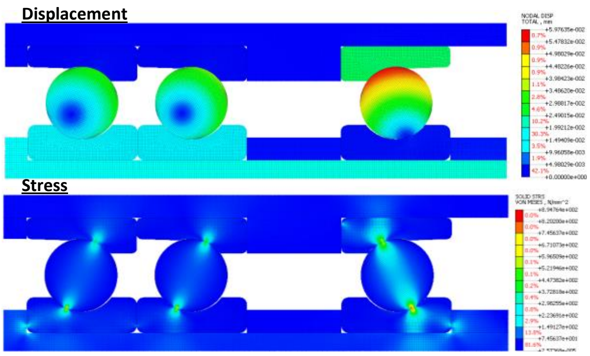

Figure 3. Finite element analysis results – medium preload.

Based on the finite element analysis results, the axial stiffness of a spindle (K_z) was found to be 424 N/µm, 588 N/µm and 682 N/µm for low, medium, and high preload conditions, respectively. These results demonstrate the feasibility of the proposed concept.

4. Experimental characterization of prototype built.

Next, a prototype spindle was constructed, mounted on a dummy bed, and connected to a hydraulic power unit. The built-in hydraulic variable preload system was controlled via a proportional pressure-reducing valve, with control executed through an algorithm implemented in a dSpace MicroLabBox. A detailed description of the control system is available in [26]. The prototype spindle is depicted in Figure 4.

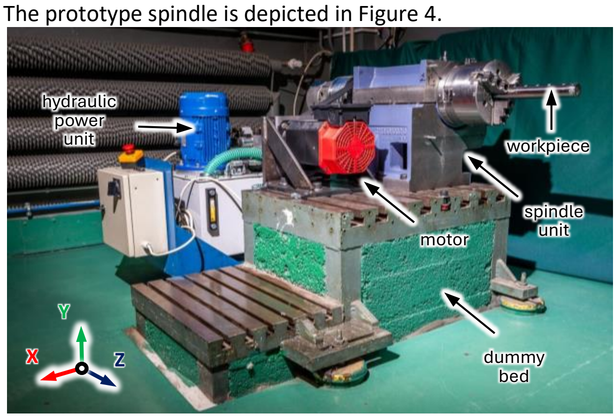

Figure 4. A prototype spindle built.

To evaluate the properties of the prototype, static and dynamic stiffness measurements were performed at different preload levels. The static stiffness measurements were conducted following the procedure described in [27]. The static stiffness values determined for the chuck relative to the headstock at various preload levels are presented in Figure 5.

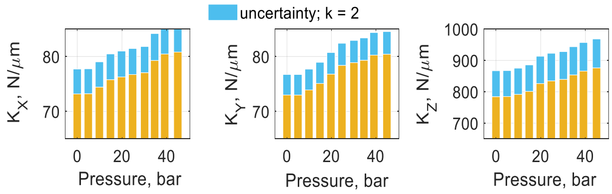

Figure 5. Static stiffness measurement results.

Subsequently, impact tests were performed following the procedure described in [28]. The driving point receptances measured at the chuck are presented in Figure 6.

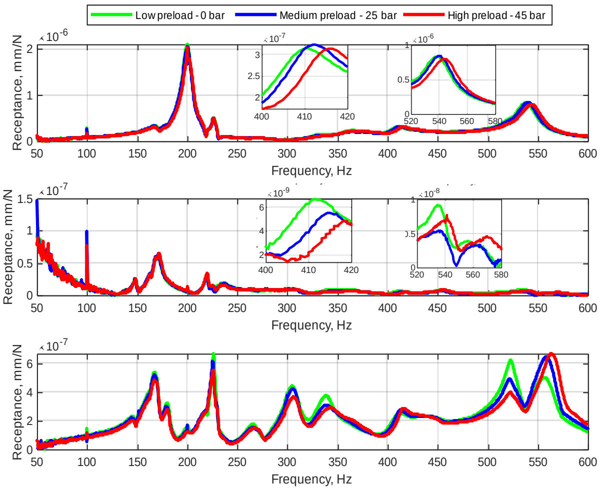

Figure 6. The driving point receptances determined at the chuck for different preload levels.

5. Discussion

The results indicate that the static stiffness of the spindle increases with the preload level. Specifically, increasing the preload from low to medium resulted in a stiffness increase of 4.2 %, 5.2 %, and 5.3 % in the X, Y, and Z directions, respectively. A further increase from medium to high preload led to a stiffness improvement of 5.9 %, 4.7 %, and 6.0 % in the X, Y, and Z directions, respectively.

When analyzing dynamic stiffness, the differences are less straightforward, particularly below 300 Hz. However, above this frequency, the impact of preload on receptance becomes more pronounced. An increase in preload causes a shift in the resonant frequencies toward higher values, with changes of up to 1 % (approximately 7 Hz). Notably, the amplitude of the resonances decreases with higher preload, suggesting a potential increase in system damping. However, further research is required to confirm this observation.

Comparing the static stiffness results obtained from the finite element model with those from experimental tests reveals significant discrepancies. These differences may result from model simplifications. It should be emphasized that the presented model and its corresponding results were intended to provide a quantitative assessment of the effectiveness of the adjustable preload system, rather than precise qualitative predictions.

6. Conclusions

There is an increasing trend toward implementing machining processes with a high degree of autonomy. However, most machine tools are not equipped with systems that allow for adjustments to their structural properties, which limits their ability to adapt to changing machining conditions. To address this issue, this paper presents the concept of a lathe spindle with variable stiffness. The static and dynamic stiffness of the spindle was altered by adjusting the preload of the front bearing node, achieved through an integrated hydraulic system.

The proof of concept was demonstrated through both modeling and experimental tests. However, the extent to which this adjustment can effectively suppress chatter or compensate for the static deflection of the workpiece remains unknown. These aspects define key directions for further research.

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