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Investigation of Adaptive Spindle System with Active Electromagnetic Bearing

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Abstract

Available motor spindles and tool chucks generate uncontrolled process deviations at several eigenfrequencies leading to geometrical and surface distortions. A demand for highest precision motor spindles, tool chucks and cutting tools arises for advanced cutting processes, free from tool deflection and vibrations. This paper targets a new design of a non-typical Adaptive Spindle System (AS) with an additional electromagnetic bearing based on mechatronics and adaptive control methods for advanced cutting technologies and proposals drafting on AIS development. The static and dynamic performance determination of the AS at speeds of up to 15,000 rpm has been performed on a test bench with actuator stimulated forces and displacement sensors. The analysis of time-domain and amplitude-frequency characteristics confirmed the demand in adaptive closed-loop control methods for compensating tool deflections and vibrations at eigenfrequencies.

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Keywords: motor spindle, magnet bearing, tool deflection, tool vibrations, adaptive control

Nomenclature

AC	adaptive control
AS	adaptive spindle system
MB	electromagnet bearing
TCP	tool center point

1. Cutting process demand

A demand for highest precision motor spindles, tool chucks and cutting tools arises for advanced cutting processes, which should be free from tool deflection and vibrations.

Dynamically balanced rotors, tool chucks and cutting tools are prerequisites for precision manufacturing. Monitoring for cutting process forces and displacements between the TCP and work piece requires additional instrumentation for work piece clamping and spindle. The state of the art analysis in machine tool main spindle units does not include the following adaptive spindle system. [1]

Answering the technological demand, an innovative Adaptive Spindle System (AS) has been designed with an additional electromagnetic bearing for a radial force < 2 kN. Double ball bearings support the rotor on both sides, while the electromagnetic bearing in the center is shifting the rotor for tool deflection and vibration compensation. [2], [3], [4]

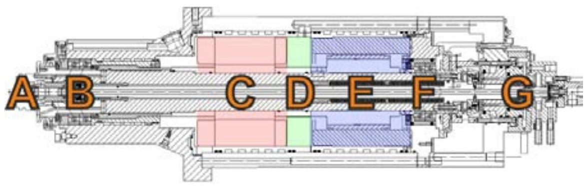

Fig. 1. Motor spindle structure with active magnet bearing

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The motor spindle rotor scheme in Figure 1 shows:

A	tool chuck HSK 63
B	double ball bearing
C	electromagnetic bearing (red)
D	rotor position sensors (green)
E	synchronous spindle motor (blue)
F	double ball bearing
G	tool clamping mechanism

The new design model was already analyzed for static and dynamic compliance and results in calculated values for stiffness and eigenfrequencies. The experimental performance determination of the AS prototype has been performed on a test bench. (nominal 82Nm / 4,000 rpm; maximal 34.5kW / 15,000 rpm; water cooling 20°C ±1°C).

2. Tool deflection

The tool deflection (TCP) is a function of the cutting force and electromagnetic bearing force. Experimental tests of the motor spindle on a test bench with different static TCP/MB forces result in TCP/MB displacements, as shown in Figure 2.

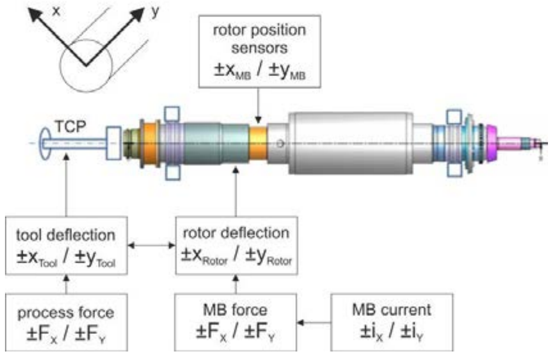

Fig. 2. Spindle rotor and tool bar

Methodology: The MB current has been set from minus to plus 15 A in the x-axis direction and the resulting TCP displacement in x and y direction has been measured. A negligible correlation between TCP displacements in x and y axes has been observed, see Fig. 3.

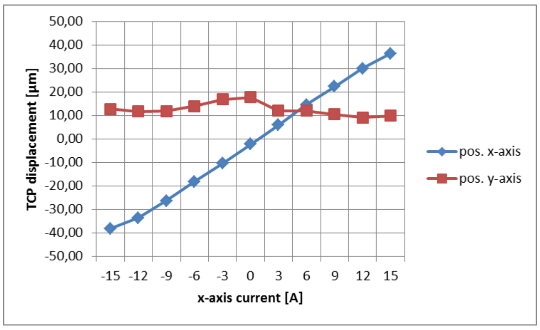

Fig. 3. TCP displacement as a function of x-axis current

The rotor position sensor consists of four pairs LED photosensors in (x; y) with ±3V interface. A displacement of 1 μm of the rotor at the rotor position sensor leads to a tool deflection of 1.13 μm at the TCP when no process force is applied. The active electromagnetic bearing is able to generate a magnetic force, compensating a radial process force of 1 kN and tool deflection of 75μm.

3. Tool vibrations

The tool vibration as a function of the rotor vibration, caused by rotor misbalance and electromagnetic bearing force, has been analyzed at the test bench. Methodology: The tool bar has been radially stressed by an impact hammer in x-axis direction and monitored by an acceleration sensor. The dynamic reaction of the tool bar and motor spindle has been measured in time and frequency domain.

3.1. Dynamic compliance test in time domain

The spindle rotor was stimulated with an impact hammer in negative x (or y) direction at the TCP. An acceleration sensor was placed at the top of the TCP and the TCP force sensor was not in contact.

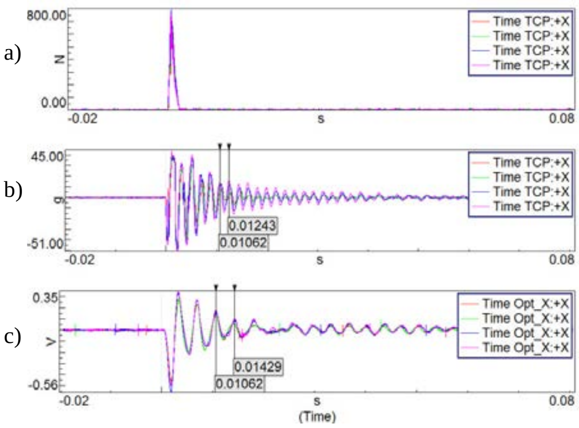

Fig. 4. TCP force control and displacement measurements

Diagram a) in Figure 4 shows the pulse resulting from an 800 N impact force generated by an impact hammer at the TCP. Diagram b) shows the resulting acceleration at the TCP. Diagram c) displays the voltage at the rotor position sensor (1 V is equivalent to 40 μm position shift). The time delay between TCP force and rotor position sensor is 0.7 ms. The dominant frequency for the TCP is 550 Hz, measured in the time domain, and for the MB it is 270 Hz for the stand still rotor. The TCP eigenfrequency of 550 Hz is equivalent to a 3 teeth tool at spindle speed 11,000 rpm or 5 teeth tool at 6,600 rpm.

3.2. Dynamic compliance test in frequency domain

The measurements were made with a data acquisition system. Figure 5 shows rotor stiffness amplitudes at TCP