CIRP Short Technical Presentation

Damping of Spindle Vibration Through Active Magnetic Bearing

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www.starragheckert.com

www.eaat.de

Damping of Spindle Vibration Through Active Magnetic Bearing

Reasons for Development an Adaptive Motor Spindle

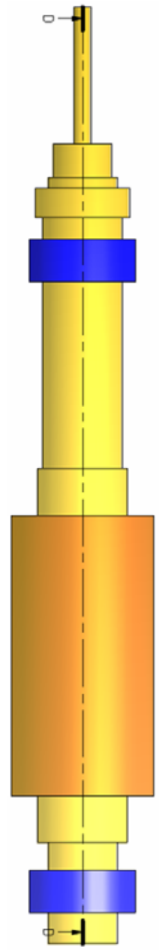
- general trend towards complete machining calls for tool spindles with a measurably greater range of application
- vibrational behaviour (critical frequencies) of tool-spindle-system is one of the limiting aspects for wide range spindles

→ search is for an adaptable universal-application high-frequency motor spindle

→ no need to replace the spindle during different machining operation

→ allowing briefer set-up times with a simple tool change instead of a spindle replace

→ Breaking through the existing limitations with a spindle design that can be statically and dynamically adapted to process efforts.



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2 Approach and Principle

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Approach

Active influencing of the first spindle eigenfrequencies by integrating an additional active magnetic bearing

⇩

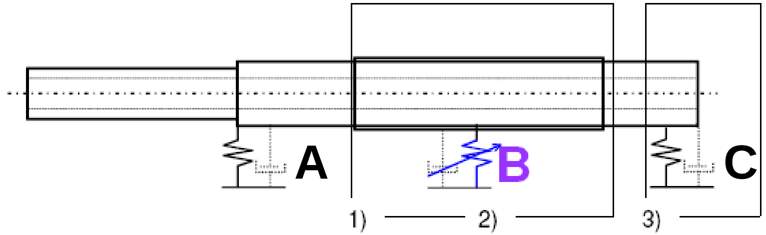
variation of magnetic bearing stiffness and damping

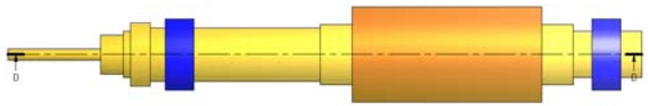
⇩

⇨ shifting and damping of critical frequencies for

Principle

A front rolling bearing
B **active magnetic bearing**
(positive/**negative** stiffness/damping possible)
C rear rolling bearing





Theoretical works were presented 2006 in the PhD-Thesis of Thomas Klaffert “Self-optimizing HSC Motor Spindle” (University of Chemnitz)

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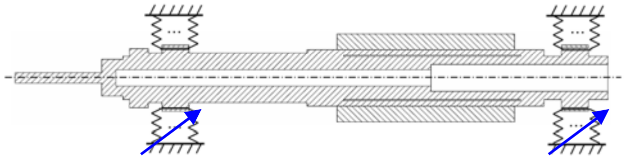
*Markus Wabner

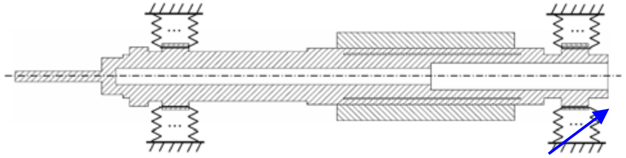
Some Further Adaptive Motor Spindle Structures

I.



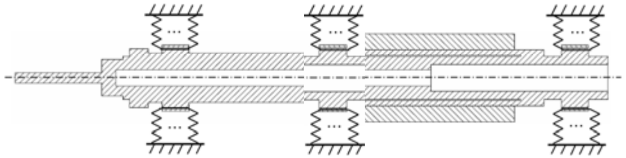
active magnetic bearing — active magnetic bearing

II.



passive roller bearing — active magnetic bearing

III.



passive roller bearing — passive damping bearing — passive roller bearing

Characteristics

I./II.

- active shifting/damping of eigenfrequencies
- limited range of adaptation because basic stiffness is needed
- friction welding of safety bearings in failure case during high revolution speed

III.

- damping eigenfrequencies
- no adaptation possible

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3 Actual Works

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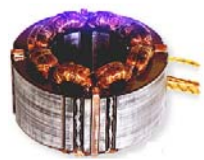
Overview: Design of an Evaluation Model

HSC spindle of StarragHeckert AG Rorschacherberg



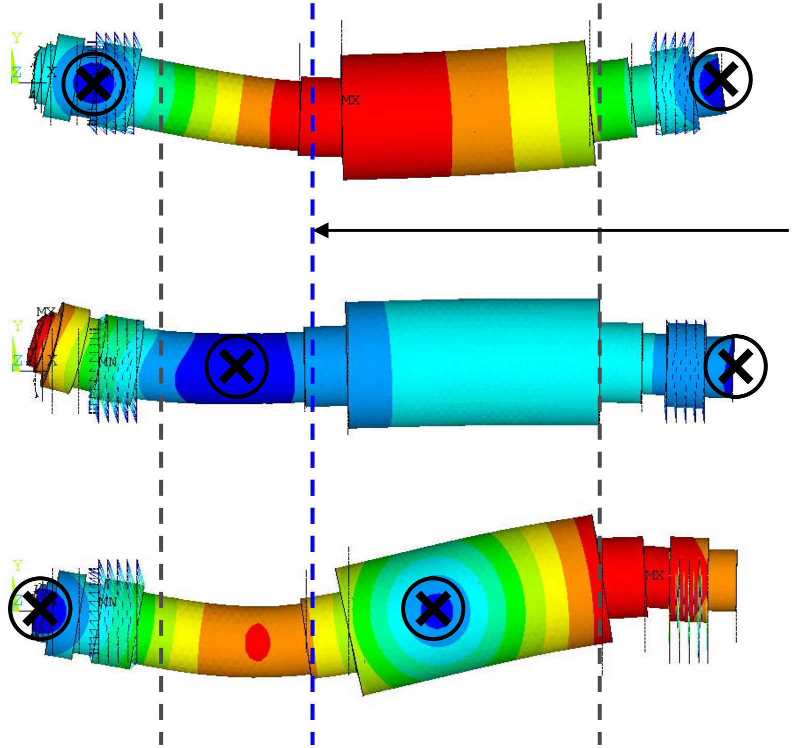
Active magnetic bearing of EAAT GmbH Chemnitz

- modifying of a industrial applicated standard spindle and prediction of its static and dynamic behaviour with an additional active magnetic bearing
- replacement of the conventional asynchronous motor by a smaller synchronous motor for getting enough cross-section for a magnetic bearing
- dimensioning of the active magnetic bearing
- manufacturing and assembly of the spindle components

Optimal Placement of Magnetic Bearing

Bending Modes (FEM)

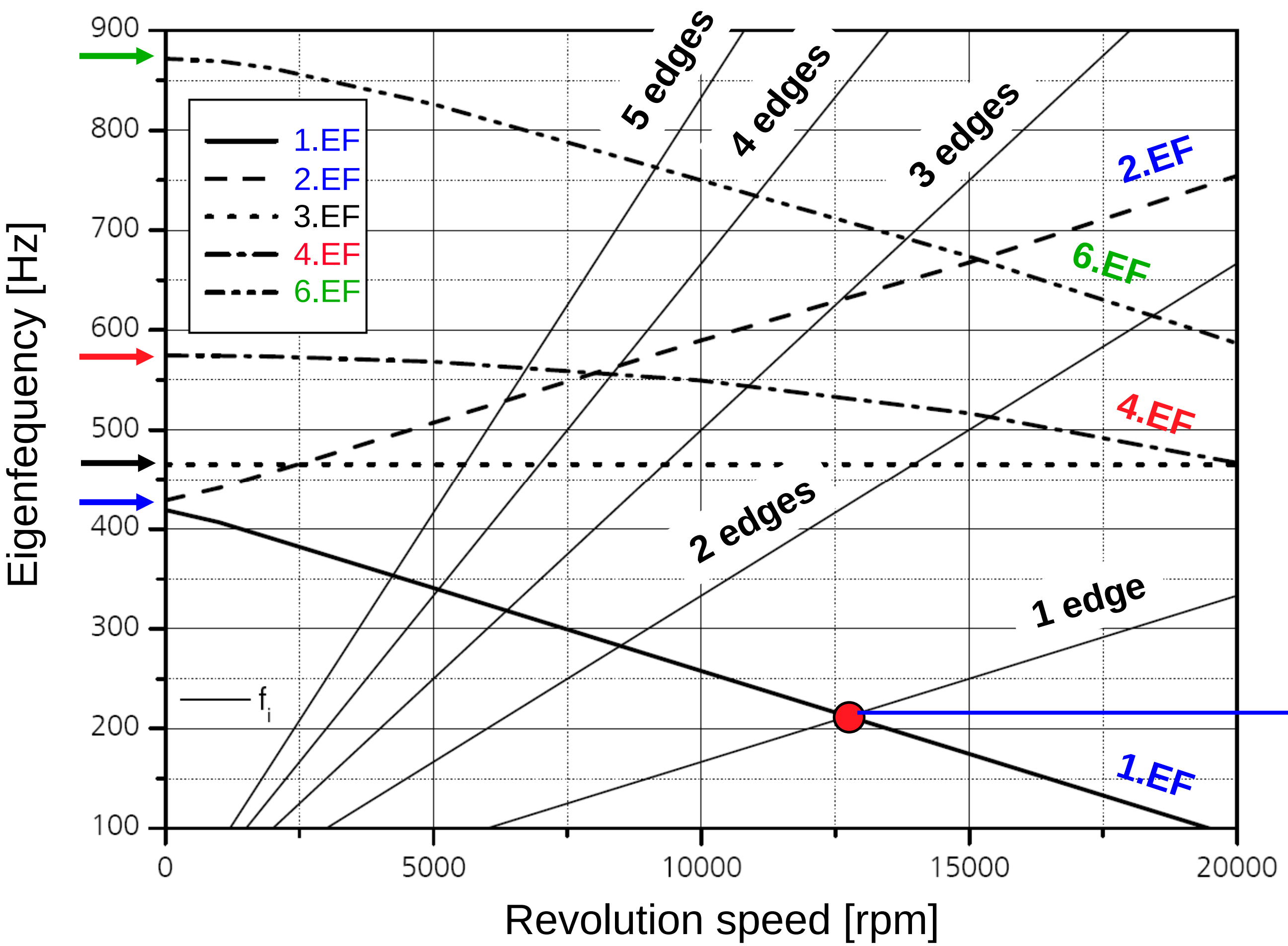
Number	Frequency (Rev.=0/min)	Eigenform
1/2	420 Hz	
4/5	580 Hz	
6/7	880 Hz	



Selected plane for magnetic bearing between nodal points of relevant bending modes

⊗ nodal point

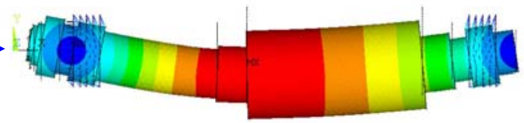
- - · possible planes for magnetic bearing



Dependency between eigenfrequency and revolution speed in rotor dynamics

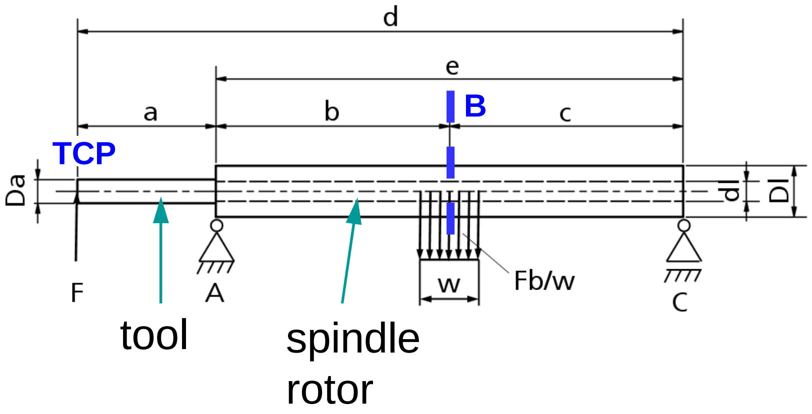
-damped model
-coriolis forces
-gyroscopic effects

Example:
1. Eigenfrequency



RS = 12,700 rpm

Estimation of Needed Static Magnetic Force



Standard spindle parameters
- interspace e=600 mm
- rotor outer diameter Dl=65 mm
- ball bearing A and C stiffness about 1 kN/µm
- process force F up to 5 kN

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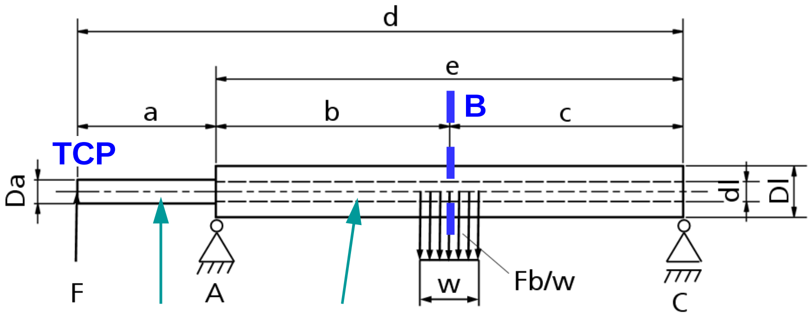
- influence analysis by using an analytic model of the base spindle and idealized magnetic bearing
- consideration of
  - main spindle dimensions
  - tool dimensions
  - ball bearing stiffness
  - process force range
  - magnetic bearing width w and force Fb
- calculating the bending line
- compensation of deflection by magnetic force

Magnitude of magnetic force has to be in order of the process forces to avoid deflections at **B** under quasi-static loads at **TCP**.

Estimation of Needed Dynamic Magnetic Force



Standard spindle parameters
- interspace e=600 mm
- rotor outer diameter Dl=65 mm
- ball bearing A and C stiffness about 1 kN/µm
- process force F up to 5 kN

- influence analysis by using an FEM model of the base spindle and idealized magnetic bearing
- additional consideration of coriolis forces and damping
- gyroscopic effects
- calculation of quotient of deflection on magnetic bearing place **B** between
  - excitation on magnetic bearing place **B**
  - excitation on **TCP**

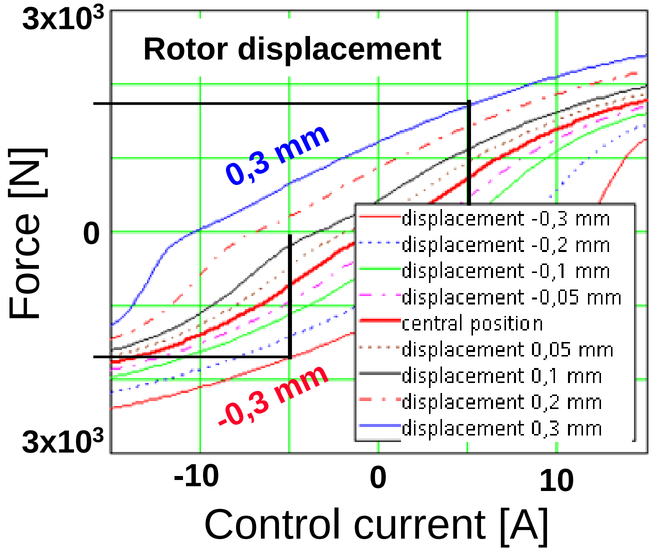
⇒ Magnitude of magnetic force has to be 2.5 times bigger than process forces to avoid deflections at **B** under dynamic loads at **TCP** in case of resonance.

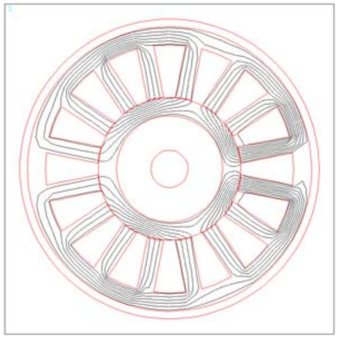
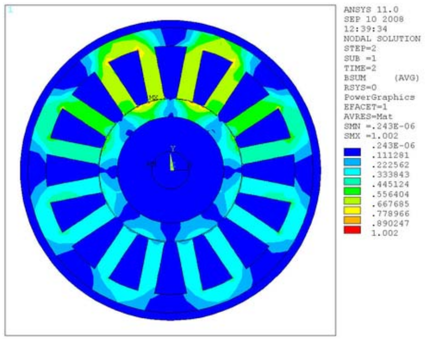
Design of Magnetic Bearing

- dimensioning of the magnetic bearing
- calculating magnetic and electric values
- dimensioning of the measuring system

Finite Element Simulation Model calculation of

- magnetic fields
- magnetic force
- rotor displacement



H-field (field intensity) B-field (flux density)

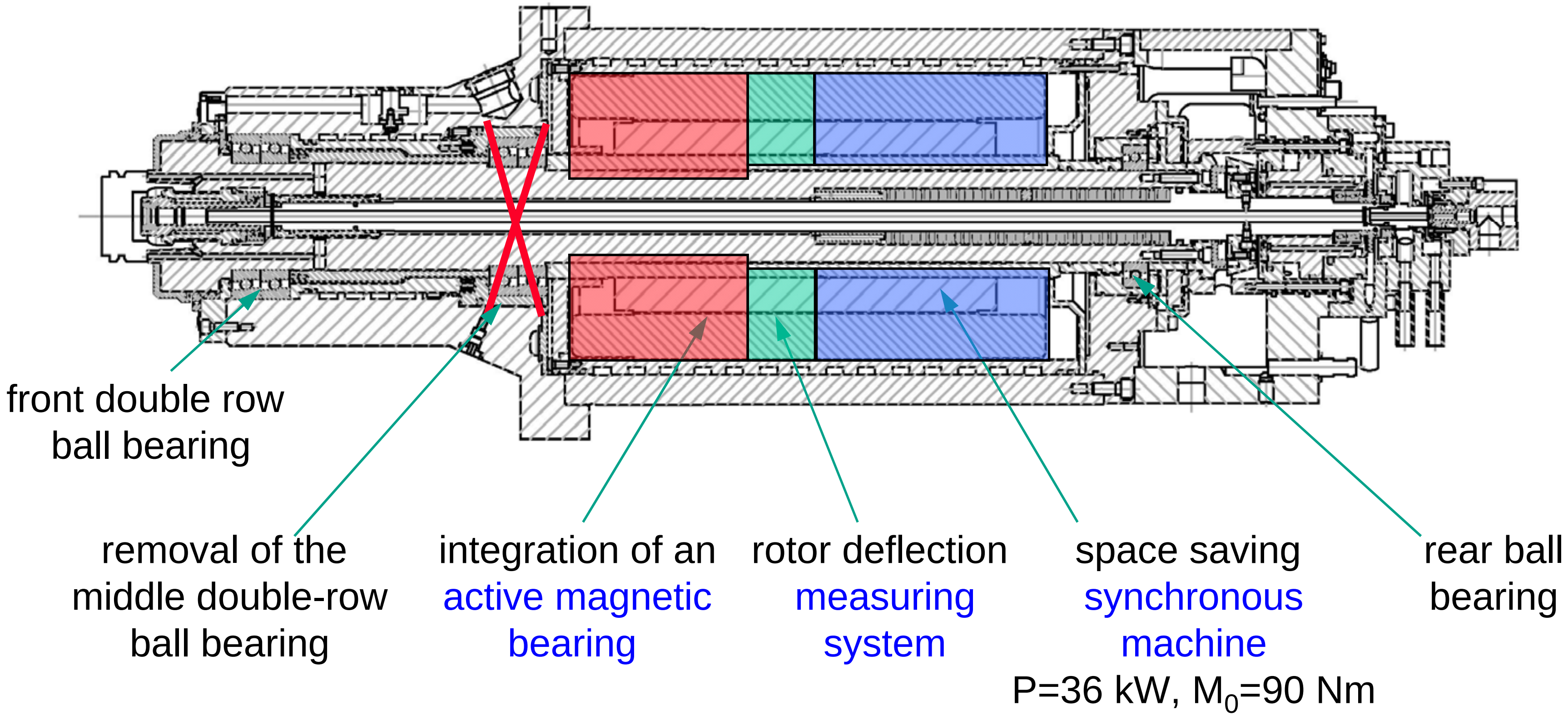
<u>Bearing Parameters</u>

- stator diameter
- rotor diameter DI = 65 mm
- width of magnetic bearing 120 mm
- force up to 1,8 kN

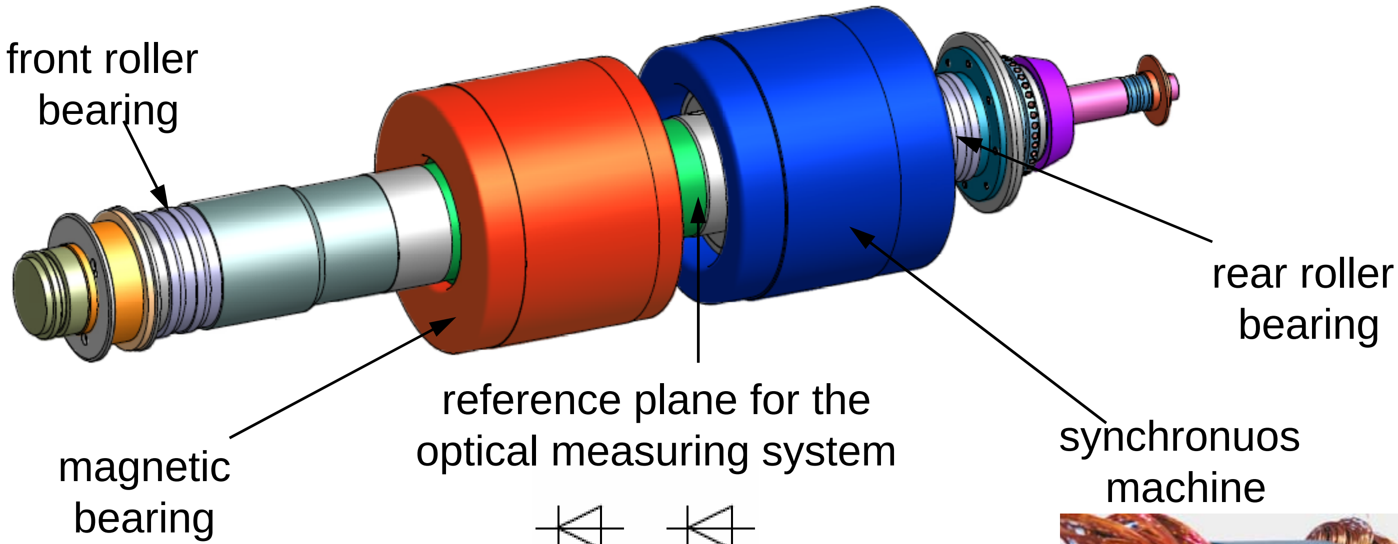
System Concept of the Evaluation Model

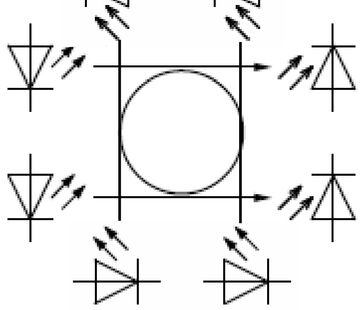
bases on StarragHeckert 262-19-15 asynchronous motor spindle

P=19 kW, M_0=165 Nm, n=15.000 min^{-1}



Design of Spindle Rotor





www.ate-system.de

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4 Summary and Outlook

Summary

- control the dynamic stiffness of a motor spindle by using an active magnetic bearing
- substitution of one ball bearing by a magnetic bearing including a deflection measuring system
- saving cross-section by using a synchronous machine

Outlook

- all components are manufactured and will be assembled soon
- subsequent tests
  - to verify simulation results
  - to optimize mechatronic simulation model
  - to develop optimal control strategy
- optimized mechanical and electrical design of motor spindle

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