Innovations in Air Bearings and Their Techniques

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***Abstract*: For micro machining high-speed aerostatic spindles are needed to achieve the required cutting speeds at high rotational accuracy. In this paper, a study of aerostatic bearings for the use in compact high-speed spindles for micro machining applications is introduced. The perturbation and the finite element method are used to determine the static and dynamic characteristics of the air bearings considering the translation and tilt motion of the rotor. These characteristics are then used to build a rotor dynamic model of the entire spindle, providing critical rotational speeds. With the help of the model a spindle was designed which could reach speeds up to 466,000 *rpm* at a high rotational accuracy. In this content there will be a clear and expanded explanation about the Air Bearing which is used in the small machines and that can have good accuracy of Values. In this explanation of there is a clear study of the Air Bearing and its types and the different bearings. The questions such as why do we use air instead of oil and there uses will be verified and why and when the Air Bearing should and can be used are also discussed in the upcoming slides of the content and the content also discusses about why are we supposed to use the Air Bearing when compared with the Ball Bearing and how it varies from Ball Bearing in the measuring and analyzing environment. In these presentations there will a discussion about the types of Air Bearing which was separated due to their load carrying capacity namely i)Aerostatic Bearing and ii)Aerodynamic Bearing . When there is a talk about the two types of Ball Bearing , the Aerostatic Bearing is a external pressurized as it has external supply of air is fed under pressure between the two surfaces being kept apart and it is a continuous flow system where pressurized gas from the source flow through out. Generated by the relative motion by the two surfaces being kept apart. There are two classifications in the Aerodynamic Bearing those are the Foil Bearing and the Spiral groove bearing. After which there will be a tabulation of the advantage and disadvantage of the use of the Air Bearing.**

Key words: Air breathing engine, static pressure, combustion efficiency, compact air bearing spindle, stability analysis of the engine

I. INTRODUCTION

Due to the rising need of miniaturization of components and functionalization of component surfaces, the accuracy and economic aspects of manufacturing processes need to be improved. Industrial ultra-precision (UP) machine tools for micro machining exceed the average component size which normally is less than 100 x 100 *mm*2 by several orders of magnitude . The filigree micro structures on the component surface are fabricated by micro tools, e.g. micro end mill so micro pencil grinding tools, with diameter ranges from 3 *µm* to 3 *mm*. To improve the handling of UP machine tools in regard to installation space and dynamic properties of the axis, machine tools need to be adapted to the work piece and micro tool size. In order to achieve this downsizing and to improve performance of machine tools, small and lightweight parts need to be developed. The main spindle is the machine tools major component defines many characteristics of the machining process. Because of the small dimensions of the tools, rotational speeds offered by conventional machining spindles are not sufficient to reach require cutting speeds. Spindles for micromachining can be classified in electric motor driven and air turbine driven. The main advantage of air turbine driven spindles is their easy handling, as they require less installation space and no power electronics compared to the electric driven ones. The rotational speed and torque of an air turbine driven spindle is adjusted by the air mass flow through the turbine. In addition to better handling the components of air turbines can be manufactured at comparably low cost. Spindles for micro machining have been analyzed in several studies. Rotational speeds of 422,400 *rpm* have been achieved by an air turbine driven spindle designed by SUNG et al. The micro tools were implemented into the spindle by a shape memory alloy based clamping system with a radial run out of 3.2 *µm*. An air bearing spindle, reaching rotational speed of approx. 500,000 *rpm* has been developed by JAHANMIR et al. The spindles tool is shrink fitted onto the rotor, which is supported by aerodynamic foil bearings. This constructive design allows for a low run-out error of 2.5 *µm*. Further analysis of machining tests showed, that the stiffness of foil air bearings is too low to handle the feed normal force initiated by the micro end mill. A spindle designed by LI et al. reached 240,000 *rpm* with a tool run-out of 2.79 *µm*. This spindle was equipped with a specially designed reducing the radial tool run-out [11]. It decouples the turbine shaft and the tool at high speeds. In this case the tool is supported by aerostatic bearings. The Printed Circuit Boards (PCB) industry is a major customer of commercially available air bearing spindles. These spindles are provided e.g. from ABL, West wind or Aerolas and reach rotational speeds up to 300,000 *rpm*, tool run-outs are in the lower two-digit micro meter range. While the rotational speed of most spindles reach sufficient cutting speed for micro machining, their rotational accuracy needs to be improved. Spindles reaching satisfying values of radial run out in micron level achieve less than 200,000 *rpm*. Additionally, these spindles require high installation space, making them inadequate to be implemented in desktop sized machine tools. This study deals with the development of air bearings for compact high-speed spindles. Therefore, different designs of aerostatic bearings were studied with Finite Element Analysis (FEA). In addition, the characteristic properties of the air bearings were implemented in a rotor dynamic model of the spindle, calculating critical rotational speeds. With the help of this simulation procedure spindles that achieve 450,000 *rpm* and above at high rotational accuracy could be designed.

II.SPINDLE AND AIR BEARING DESIGN

Air bearing spindles were developed by the Institute for Manufacturing Technology and Production Systems (FBK) in the past [12]. For all spindles, cemented carbide tool shanks Were adapted as the spindles rotor. Thus, the tools themselves are a part of the spindles bearing system, no further chucking for the micro tool is necessary. Figure 1 exemplary shows a model in half section of such a spindle. The simulation procedure for the design of such spindles will be presented in the following.

Fig:1 DESIGN OF A COMPACT AIR BEARING SPINDLE

III.BEARING CHARACTERISTICS

Figure shows the geometric features of the air bearing analyzed in the present paper. All bearing are inlet compensated with inherent feedholes. The available space for the rotors bearing is limited to 18 *mm* in length. In order to achieve a better tilting performance, by two single-rowed bearing points.The bearing length *L*, supply pressure *ps*, diameter of feedholes *df* and bearing gap *c* have been varied to receive information about the influence of the named parameters on the stability of a rotor at defined working conditions. The different bearing combinations can be seen in table 1.

combination	z	D	c	d_f	p_s	L
1-3	6	3.175	4	75	8	3, 6, 8
4-6					16	3, 6, 8
7-9				100	8	3, 6, 8
10-12					16	3, 6, 8
13-15				200	8	3, 6, 8
16-18					16	3, 6, 8
19-21			6	75	8	3, 6, 8
22-24					16	3, 6, 8
25-27				100	8	3, 6, 8
28-30					16	3, 6, 8
31-33				200	8	3, 6, 8
34-36					16	3, 6, 8
37-39			8	75	8	3, 6, 8
40-42					16	3, 6, 8
43-45				100	8	3, 6, 8
46-48					16	3, 6, 8

IV. BEARING GEOMETRY COMBINATION

4.1. Static characteristics

Figure 4 shows the calculated static pressure profile for bearing combination 8 with the length of 6 *mm* and a supply pressure of 8 *bar* at 100,000 *rpm*. With the static pressure profile, the friction losses of the bearing, the load capacity, as well as the mass flow rate can be calculated . Static pressure profile of the journal bearing at a rotational speed of 100,000 *rpm* and eccentricity of 0.01 for bearing combination 8.

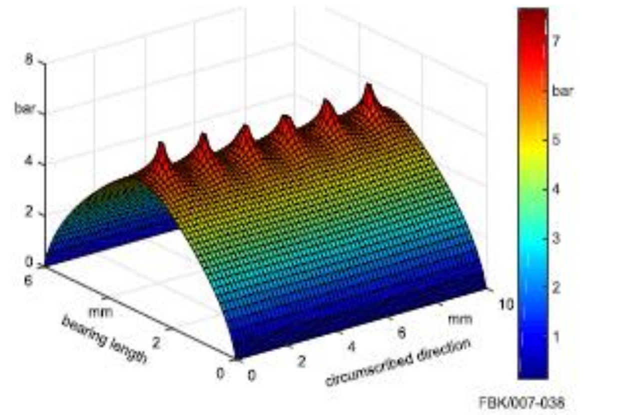
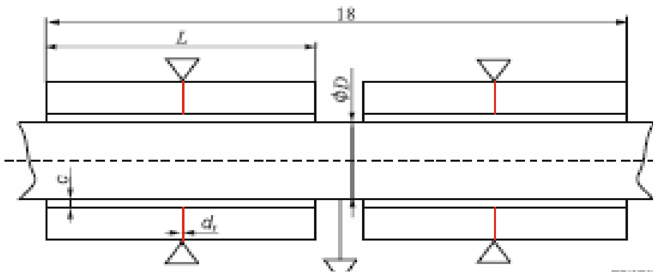


Figure. 3 1STATIC PRESSURE PROFILE OF THE JOURNAL BEARING AT A ROTATIONAL SPEED OF 10,000 rpm AND ECCENTRICITY OF 0.01 FOR BEARING COMBINATION



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4.2.Dynamic characteristics

To predict the dynamic behavior of the air bearing, the dynamic stiffness and damping coefficient have to be calculated first.

The dynamic stiffness and damping characteristics of the air bearings are calculated with the linear perturbation method. A detailed description of this method is presented by HAKWOON et. al. The coefficients *kxx* and *cxx* represent the direct stiffness and damping behavior of the supporting fluid film, while the two other coefficients *kxy* and *cxy* describe the cross-coupled behavior. Generally, the dynamic stiffness of an air bearing reaches an asymptotic value at high perturbation frequencies (rotational speeds), while the damping of the bearings tends to zero.

V. STABILITY ANALYSIS

In this paper, a rotor dynamic model, based on the Finite Element Method (FEM) performed with Ansys® was used for the analysis of the stability. The procedure to receive information about the stability of a rotor at specific working conditions can be seen in figure 6. The parameters of the bearing combination and working conditions in combination with FEM provide values for *kxx*, *kxy*, *cxx* and *cxy*. This calculation, in conjunction with the rotor geometry, is the input for a stability analysis. This analysis results in detailed information about the particular stability of the rotor at present working conditions. Based on the speed-dependent values of *kxx*, *kxy*, *cxx*, *cxy* and the related Eigen frequencies, Campbell diagrams reveal critical speeds. Critical speeds of the spindle are rotary frequencies where the perturbation unstable operation conditions. For the FEM a stationary reference frame of the rotor system is described by the following differential equation.

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

[M] represents the mass matrix created for each finite element. [C] represents the damping, [K] the stiffness and [G] the gyroscopic term. The gyroscopic term considers how the rotational speed of the rotor directly affects the natural frequencies of the rotor vibration and the mode shape and its corresponding orbit of the system. The system of equations is solved as an eigenvalue problem. The corresponding eigenvalues represent the natural frequencies of the system, the eigenvectors describe the mode shape and the resulting orbit of the rotor system. Eigenvalues contain a real part and an imaginary part. The real part η indicates the stability of the system, as $\eta < 0$ stands for a stable solution. The imaginary part of the eigenvalues describes the damped Eigen frequency ωd of the system. A mode is considered forward whirling if it is rotating in the same direction as the rotor rotation and a backward whirling if it is rotating in the opposite direction. Modes typically come in pairs, with the forward whirling modes increasing in frequency with rotor speed, while the backward whirling modes decrease in frequency with rotor speed. Figure 7 shows the Rotor with air turbine and the predefined FE elements for the springs and dampers used for the rotor dynamic analysis. The bearing consists of four Voith-Kelvin-elements for the direct *ii* and cross-coupled *ij* stiffness *k* and damping properties *c* shows the stability map of different simulated operating conditions and bearing combinations according to the introduced rotor dynamic model. The design parameters of the simulated bearing combinations are stated in table 1. The combinations in figure 8 are sorted by increasing bearing gap *c*. The stability map presents the real part of the eigenvalues of the different combinations at different rotational speeds. As the real parts of the eigenvalues are an indicator for stable or unstable oscillation, an area for stable spindle design can be deduced. The color scaling enables conclusions about the influence of bearing gap *c* on the stability of the rotor geometry at specific working conditions. The combinations with smaller bearing gap point to smaller real parts of eigenvalues, meaning more stable oscillations. With increasing bearing gap, more working conditions lie in the unstable area, which is marked in red. The stability map also reveals that the rotational speed at otherwise constant conditions only has a minor influence on the real parts of the Eigen frequencies. For stable combinations, the real part stays in the green region for rising rotational speeds. The stability map also reveals that higher bearing lengths *L* at otherwise constant conditions highly contribute to stable conditions. In the examined range, the supply pressure *ps* only has a minor influence on the stability, still, it can be stated that higher pressures contribute to a higher stability. The feed hole diameter *df* also only has a small influence, while smaller diameters contribute to a more stable behavior. In addition to the illustration of stable bearing combinations, critical speeds of the particular combination can be estimated.

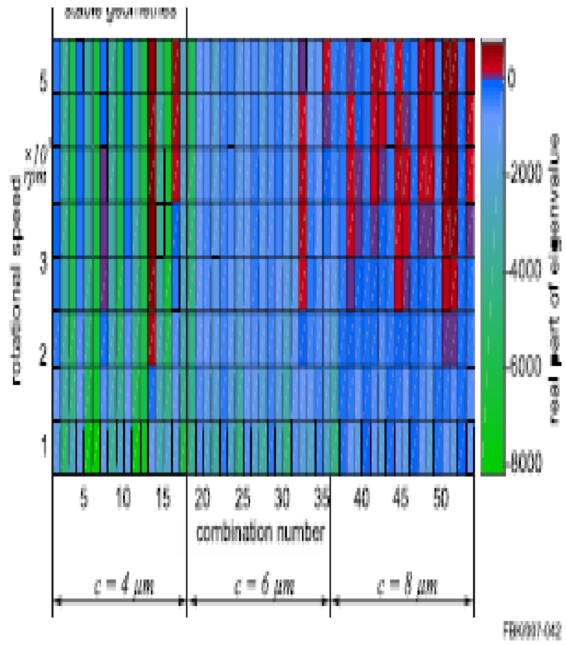

Fig: STABILITY MAP OF DIFFERENT BEARING COMBINATIONS,RESULTS ASSOCIATED TO A GRID POINT.

Figure shows exemplary the Campbell diagram of combination, with its first five Eigen frequencies. The squares highlight the resonance frequencies as the consequent excitement frequency reaches the related Eigen frequency of the rotor. Working conditions near these critical frequencies can lead to vibrations causing contact between rotor and stator of the air bearing accompanied by critical damages.

Based on the information about resonance frequencies, critical rotational speeds of the rotor can be deduced. .

EIGEN FREQUENCY	CRITICAL ROTATION SPEED/rpm
1	~131,400
2	~180.000
3	~221.400
4	~445.200

CRITICAL ROTATION SPEED FOR THE COMPARITION WITH TH E FREQUENCY

Figure shows the corresponding calculated mode shapes and orbits for the rotor in the Campbell diagram. The first two modes (a) and (b) are conical modes and the second (c) and third (d) are cylindrical modes. Respectively forward and then backward whirls. The fifth mode (e) is the first bending mode of the tool shank, which is very high because of the high Young's modulus of the cemented carbide.

VI. FUTURE USES

In this paper, a simulation procedure to evaluate the design of aerostatic bearings for the use in compact high-speed spindles for micro machining applications was presented. The Finite Element Method was used to solve the time-dependent Reynolds Equation, providing information about dynamic stiffness and damping characteristics of the bearings for different design parameters (feedhole diameter, bearing gap, supply pressure, bearing length). The calculated characteristics of the bearings were used to build a rotor dynamic model of the spindle. With this model the spindles Eigen frequencies and mode shapes can be calculated. Thus critical speeds could be deduced. Based on the parameter study, following design guidelines can be formulated:

• The air bearing gap has the biggest influence on the Dynamic stiffness and therefore the resulting
Eigen frequency of the rotor. A smaller air bearing gap Results in a higher stiffness and higher Eigen frequencies.

• A higher length of the bearing also significantly contributes to more stable conditions.

• The feedhole diameter and the supply pressure do not have a high influence on the stability. Still, it can be seen that smaller feedhole diameters and higher supply pressures contribute to a higher stability. The simulation procedure can reduce time and cost intensive manufacturing of air bearings. Future work will include the manufacturing of a beneficial combination as revealed by the simulation and the application of the bearing/spindle for micro machining.

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