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# DEVELOPMENT OF A DESIGN ENVIRONMENT FOR HIGH-SPEED AIR-BEARING SPINDLES IN MACHINE TOOL APPLICATIONS

**CATEGORY**: Fluid Film Bearings

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## INTRODUCTION

A spindle is a motor-driven shaft that either positions and transmits power to a cutting tool or holds a workpiece[1]. The electro-spindle does not rely on an external motor and transmission system to provide torque and power, but is a self-contained motorized spindle system. This allows the spindle to rotate at higher speeds as a compact unit, and eliminates additional limitations coming from the transmission system. Air Bearings have the advantages of very low error motions due to fluid-film-averaging effect, frictionless high-performance positioning, cleanliness and minimal maintenance. Thus electro-spindle equipped with air bearings is of our interest in this study.

This paper describes the development of a design environment for high-speed spindles with air bearings particularly in machine tool applications. It proposes an overall optimization scheme using two-level approach with genetic algorithm for the design of air bearings and spindle shaft. Designs of air bearings are completed in the first level optimization and then it starts the second level optimization task on the spindle shaft by maximizing the first resonance frequency using the shaft's rigid-body mode analysis to obtain the optimum dimensions. Finally it generates the preliminary dimensions of the air-bearing spindle set. The whole design processes have been integrated into an interactive design environment using the graphical user interface.

## DESIGN OPTIMIZATION OF AIR-BEARING SPINDLES

Typical configuration of an air-bearing spindle is shown in Figure 1. Figure 2 and 3 give the configurations of collar thrust bearing and journal bearing. The design of the spindle shaft's dimensions is crucial to the whole spindle set design. Not only the tool interface and the drawbar mechanism are in direct contact with the shaft contours, but also almost all of the spindle's components are directly or indirectly attached to the spindle shaft. Once the dimensions of the shaft are determined, the dimensions of the whole spindle set can almost be finalized [2]. For example, the shaft diameter at front journal bearing is related to the nominal diameter of HSK tool-holder taper, and this is the lower bound of the diameter of the front journal bearing. Similarly the shaft diameter at rear journal bearing is bounded by the drawbar diameter and the inner diameter of rotor. Figure 4 presents the dimensions of a spindle shaft and a cutting tool assembly. In given spindle specifications, the required axial and radial load capacity of the spindle can be transfer into the load constraints in designing the dimensions of the thrust and journal air bearings. Referring to eq-1, we propose an optimization scheme to maximize the static stiffness ($K_S$) with respect to minimum load requirement from spindle specification provided by spindle users. Additional constraints of damping ratio($\zeta$ of 0.2 or higher) is added to guarantee good damping effect. The general optimization problems of air bearings thus can be written in a general form as follows:

$$\left.\begin{aligned} &Maximize: Static\ Stiffness \quad K_S \\ &Subject\ to: \quad (1) \quad W \geq W_0 \\ &\qquad\qquad\quad (2) \quad \zeta \geq \zeta_0 \end{aligned}\right\} \qquad \text{(eq-1)}$$

The optimization problem of spindle shaft is set to maximize the first resonance frequency in radial vibration mode in analyzing rigid-body dynamics of the spindle shaft. The optimization problem of the spindle shaft thus can be formulated as: *Maximize*: $fn_1$ (eq-2)

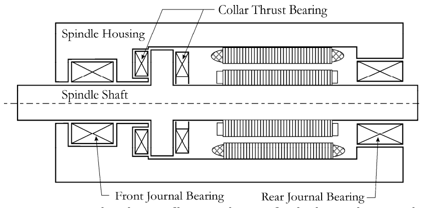


*Figure 1.Typical configuration of air-bearing spindle*

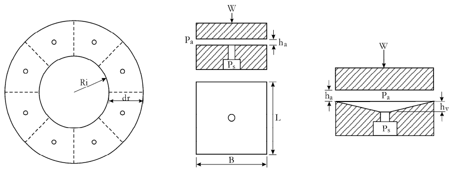

*Figure 2. Configuration of collar thrust bearing*

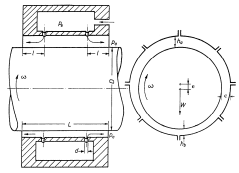

*Figure 3. Configuration of journal bearing*

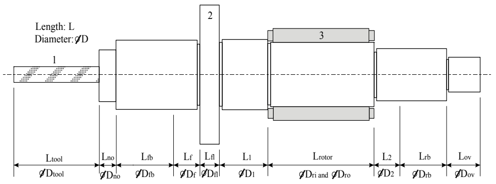

*Figure4. Dimensions of a spindle shaft and cutting tool assembly*

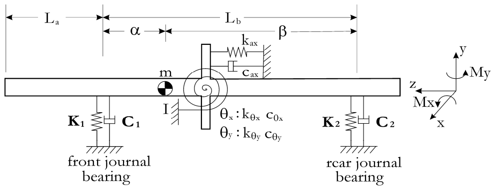


*Figure 5. Rigid-body dynamics of spindle shaft*

The optimization algorithm of genetic algorithm is used here with penalty function chosen as the Modified version of Powell and Skolnick's method [4]. Figure 5 depicts the rigid-body dynamics of the spindle shaft. Analyzing the rigid-body dynamics of the spindle shaft is to calculate the principal resonance frequencies of the spindle shaft under the assumption of infinitely rigid shaft. It gives a good initial indication about the lower vibration modes of the spindle shaft. In particular a quick check on the risk of self-excited vibration due to stiffness cross-coupling or to negative damping is afforded. The main purpose of analyzing the rigid-body dynamics is to calculate the (two radial and one axial) principal resonance frequencies of the of the spindle shaft as a first estimate before carrying out an extended analysis of flexible modes. From dynamics point of view, it is important to understand the dynamic behavior of spindle during the machining process, especially the principal resonance frequencies due to their influence on the machining quality of workpiece. Figure 6 explains the overall optimization scheme of air-bearing spindles.

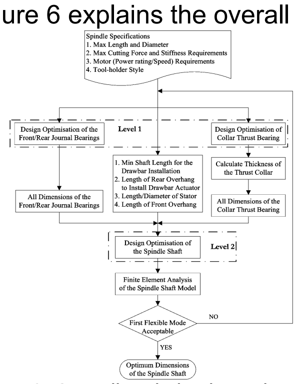

*Figure 6. Overall optimization scheme*

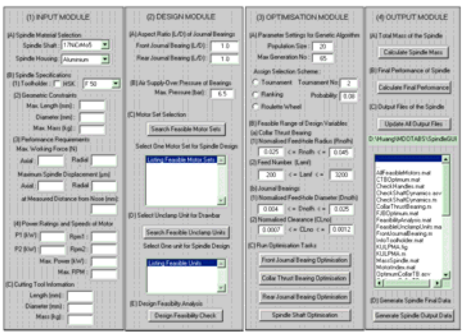

*Figure 7. Graphical user interface of the system*

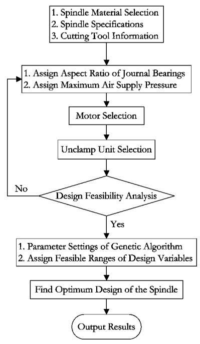


*Figure 8. Design procedure of air-bearing spindle via GUI*

## DESIGN ENVIRONMENT OF AIR-BEARING SPINDLES

The design environment consists of four modules to cover all the design issues of an air-bearing spindle with the particular configuration of our interest. Figure 7 shows an overview of the graphical user interface of the system. The source codes of this design environment are developed in the PC platform using MATLAB software. The four modules are the input module, design module, optimization module and output module. Each module has its unique functionality to accomplish a specific task in the spindle design processes. Figure 8 summarize the design procedure of air-bearing spindle via GUI.

## DESIGN CASE STUDIES AND DISCUSSION

Here we present two design cases of air-bearing spindles to explain the computer-aided design procedures of the system. These two spindles are designed for (i) woodworking industry for routing applications and for (ii) high-speed light metal milling applications. Table 1 is the spindle specifications required by spindle user. Table 2 is the preliminary selection of motor and unclamp unit based on the spindle requirement. Table 3 is the optimum dimensions of spindle shaft. The final optimum design of woodworking spindle is shown in Figure 9. For woodworking spindle, the FRF of rigid spindle shaft with tool mass of 0.5kg and 1.5kg is presented in Figure 10. In this figure, the first, and most prominent, natural frequency, which corresponds to a tilt mode, is in the range 515 to 552 Hz depending on the tool mass (1.5kg and 0.5kg) and the damping. Figure 11 shows, with different damping levels of journal bearings, the resulting FRF response of the rigid shaft with a big cutting tool. The value of the damping of the front and rear journal bearing is indicated on the panels of the figure by $C_{fb}$ and $C_{rb}$ respectively. The two eighefrequencies of the spindle shaft at tool tip in the Figure 11(1) are 515 Hz and 719 Hz. For Figure 11(4), the two eigenfrequencies are 532 Hz and 773 Hz. From this figure, we see that as the magnitude of damping of the front and rear journal bearings decreases, the two resonance peaks of the rigid shaft become sharper, and their frequencies are the same at each location, as expected. Table 5 shows the variation of the radial flexible mode for the spindle design case 1 with small cutting tool. As can be seen, the general trend of increasing both the front and rear journal bearings' dimensions will shift the flexible mode into a higher frequency range, owing to the increasing of the shaft bending stiffness in comparison with its inertia. On the contrary, decreasing the dimensions of the journal diameters will result in a lower flexible mode of the spindle shaft.

| | woodworking spindle | | milling spindle | | |
|---|---|---|---|---|---|
| | axial | radial | axial | radial | remarks |
| max working force (N) | 200 | 400 | 300 | 300 | |
| distance from nose (mm) | 100 | | 100 | | can vary depending on cutting tool |
| max displacements (μm) | 50 | 50 | 10 | 30 | |
| resulting stiffness (N/μm) | 4 | 8 | 30 | 10 | higher values desirable |
| operating speeds | 0 – 32000 rpm | | 0 – 45000 rpm | | in both senses |
| power | 13 kW @11000-22000rpm | | 12 kW @12000-45000 rpm | | |
| max. tool mass | 1.5 kg up to 11000 rpm | | 2 kg up to 9000 rpm | | |
| min. tool mass | 0.5 kg @ 22000 rpm | | 0.5 kg @ 36000 rpm | | |
| max spindle mass | 30 kg | | 35 kg | | |
| max spindle length | 500 mm | | 480 mm | | |
| spindle diameter | 160 mm (customised shape) | | 140 mm h4 (cartridge type) | | |

*Table 1. Spindle specifications given by users*

| | Woodworking Spindle | Milling Spindle |
|---|---|---|
| 1. Motor Set Type | E und A (12/09-4) | E und A (10.6/10-4) |
| No of Poles | 4 | 4 |
| Max Power (kW) | 13 | 12 |
| Max RPM | 32000 | 45000 |
| Stator Length (mm) | 157 | 158 |
| Stator Outer Diameter Dso (mm) | 120 | 100 |
| Stator Inner Diameter Dsi (mm) | 70 | 65 |
| Rotor Length Lrotor (mm) | 120 | 116 |
| Rotor Min Inner Diameter Dri_min | 25 | 28 |
| Rotor Max Inner Diameter Dri_max | 45 | 46 |
| 2. Unclamp Unit Type | Ott-Jakob(LE 150P) | Ott-Jakob(LE 150P) |
| Pull Force (N) | 6800 | 6800 |
| Release Pressure Max (bar) | 8 | 8 |

*Table 2. Preliminary selection of motor and unclamp unit*

| | Woodworking Spindle | | Milling Spindle | |
|---|---|---|---|---|
| | Axial | Radial | Axial | Radial |
| Max. working force (N) | 1577 | 714 | 1520 | 462 |
| Final stiffness (N/μm) | 47.3 | 21.6 | 45.1 | 17.7 |
| Total Spindle Mass | 26.3 kg | | 30.6 kg | |
| Total Spindle Length | 426 mm | | 409 mm | |
| Spindle Diameter | 160 mm (customised shape) | | 140 mm h4 (cartridge type) | |

*Table 4. Spindle performance of final design*

| Spindle Dimensions (in mm) | Woodworking Spindle | Milling Spindle |
|---|---|---|
| Nose Length, $L_{no}$ | 20 | 13 |
| Length of Front Journal Bearing, $L_{fb}$ | 76 | 60 |
| Length of Front Section, $L_f$ | 30 | 30 |
| Length of Thrust Collar Flange, $L_{fl}$ | 7 | 9 |
| Length of Section 1, $L_1$ | 51 | 70 |
| Rotor Length, $L_{rotor}$ | 120 | 116 |
| Length of Section 2, $L_2$ | 52 | 45 |
| Length of Rear Journal Bearing, $L_{rb}$ | 42 | 43 |
| Length of Rear Overhang, $L_{ov}$ | 28 | 23 |
| Nose Diameter $D_{no}$ | 63 | 40 |
| Diameter of Front Journal Bearing, $D_{fb}$ | 76 | 60 |
| Diameter of Front Section, $D_f$ | 68 | 54 |
| Diameter of Thrust Collar Flange, $D_{fl}$ | 126 | 113 |
| Diameter of Section 1, $D_1$ | 68 | 54 |
| Inner Diameter of Rotor, $D_{ri}$ | 42 | 43 |
| Diameter of Section 2, $D_2$ | 42 | 43 |
| Diameter of Rear Journal Bearing, $D_{rb}$ | 42 | 43 |
| Rear Overhang Diameter, $D_{ov}$ | 33 | 33 |
| Total Length of Spindle, $L_{total}$ | 426 | 409 |
| Gap Thrust Collar to Housing, $Gap_{FH}$ | 3.5 | 3.5 |
| Outer Diameter of Spindle $D_O$ | 160 | 140 |
| Diameter of Hollow Shaft, $D_{ho}$ | 20 | 20 |
| Housing Thickness $T_H$ | 13.5 | 10 |

*Table 3. Optimum shaft dimensions*

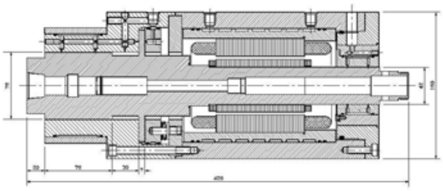

*Figure 9. Final optimum design of spindle case 1*

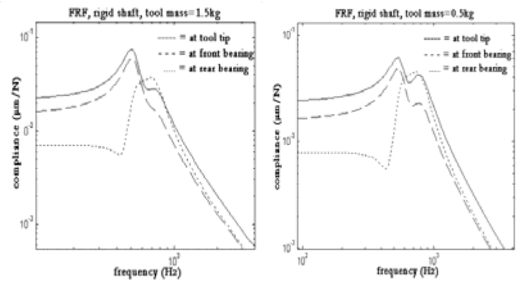


*Figure 10. FRF of the rigid shaft system for spindle case 1*

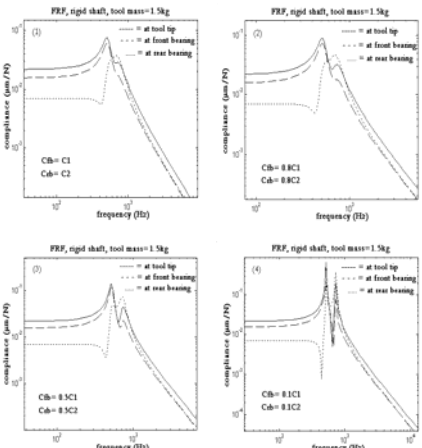


*Figure 11. Effect of journal bearings' damping magnitude on FRF of rigid shaft*

| | Condition of the Spindle Design Case 1 | | |
|---|---|---|---|
| Changed Shaft\ Dimensions, mm | Original Spindle | New Spindle1 | New Spindle 2 |
| $L_{fb}$ | 76 | 80 | 76 |
| $D_{fb}$ | 76 | 80 | 70 |
| $D_f$ | 68 | 71 | 63 |
| $L_{fl}$ | 7 | 12 | 8 |
| $D_{fl}$ | 126 | 126 | 125 |
| $D_1$ | 68 | 71 | 63 |
| $D_{ri}$ | 42 | 46 | 39 |
| $D_2$ | 42 | 46 | 39 |
| $L_{rb}$ | 42 | 45 | 42 |
| $D_{rb}$ | 42 | 45 | 39 |
| $L_{total}$ | 426 | 438 | 427 |
| Rigid-Body Mode Frequency | 552 Hz | 536 Hz | 582 Hz |
| Flexible Mode Frequency | 432 Hz | 450 Hz | 407 Hz |
| Remark | Other shaft dimensions refer to Table 5-3 of the spindle 1 | Spindle with increased journal diameters | Spindle with decreased journal diameters |

*Table 5. Effects of journal diameters on the spindle's first radial mode*

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## KEYWORDS

Air Bearings, Optimisation, Genetic Algorithm, Rigid-Body Mode