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PASSIVE CONTROL OF SPINDLE-BEARING SYSTEMS

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Abstract: *In this paper we proposed a method for passive control of spindle-bearing systems by using optimization tehqniques . The goal is to find out the position of the bearings, the diameters of the spindle (different diameters for several segments of the spindle) in order to maximize dynamic stiffness (minimize receptance), i.e. the diminishing of the vibrations. Some constraints are imposed: the distances between bearings, different diameters for several segments of the spindle, etc. The method is very useful for the design engineers from the very beginning of the design, offering to the designer the optimal values of the parameters.*
Keywords: *spindle, bearing, finite element, optimization, dynamic stiffness.*

1. INTRODUCTION

One of the most important parts of machine tool is the spindle-bearing system. The structural properties of the spindle depend on the dimensions of the shaft, bearings, tool holder, and the design configuration of the spindle systems. For HMS (high speed machining), the spindle design must be carefully decided by designers. The bearing arrangement, the preload for the bearings, the tool holder, tool interface technologies are important issues for high speed spindles [6], [9], [10]. For design optimization of spindles, Yang [1] conducted static stiffness to optimize a bearing span using two bearings, and described the methods used to solve the multi-bearing spans' optimization method. Taylor et al. [2] developed a program which optimizes the spindle shaft diameters to minimize the static deflection with a constrained shaft mass. Wang and Chang [3] simulated a spindle-bearing system with a finite element model and compared it to the experimental results. They concluded that the optimum bearing spacing for static stiffness does not guarantee an optimum system dynamic stiffness of the spindle. Hagiu and Gafiteanu [4] demonstrated a system in which the bearing preload of the grinding machine is optimized. The machining performance can be raised by improving dynamic stiffness of spindle-bearing system [5]. The dynamic performance of the spindle system are strongly influenced by design parameter such as: distance between bearing, diameter of the different portion of a spindle, bearing preload, bearing spacing etc. In most papers this influence is studied by varying the parameters and analyzing of its effect on the system. In this paper we proposed a method for passive control of spindle-bearing systems by using optimization tehqniques. The goal is to find out the position of the bearings, the diameters of the spindle (different diameters for several segments of the spindle) in order to maximize dynamic stiffness (minimize receptance), i.e. the diminishing of the vibrations. caused by cutting forces, shaft unbalance etc.. Some constraints are imposed: the distances between bearings, different diameters for several segments of the spindle, etc. The method is very useful for the design engineers from the very beginning of the design, offering to the designer the optimal values of the parameters. To solve the problem we have combined the finite element method with optimization methods. Therefore, the code computer optimization program in MATLAB is obtained by the coupling of the FEM with the non-linear optimization methods with constraints [5].

2. MATHEMATICAL MODEL

2.1. Finite element model of spindle-bearing systems

The most commonly model for analyzing a spindle systems is shown in Figure 1. In this model are the included tool, tool-holder, spindle shaft, and bearings. In this study, all components of the spindle–holder–tool assembly are modeled as multi-segment beams. Timoshenko beam model is used [5]. In the following, only axisymmetric spindles are considered. The equation of an anisotropic spindle-bearing systems which consists of a flexible nonuniform shaft and anisotropic bearings may be written as [4], [5]

$$\boldsymbol{M}\,\ddot{\boldsymbol{q}}+\left(\boldsymbol{C}+\Omega\boldsymbol{G}\right)\dot{\boldsymbol{q}}+\boldsymbol{K}\,\boldsymbol{q}=\boldsymbol{F} \tag{1}$$

where q is the global displacement vector, whose upper half contains the nodal displacements in the y-x plane, while the lower half contains those in z-y plane, and where the positive definite matrix M is mass (inertia) matrix, the skew symmetric matrix G is gyroscopic matrix, and the nonsymmetric matrices C and K are called the damping and the stiffness matrices, respectively. The matrices of M, C, G, K, q, and F consist of element matrices given as

$$\boldsymbol{M} = \begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix}_{N\times N} , \boldsymbol{C} = \begin{bmatrix} c_{yy} & c_{yz} \\ c_{zy} & c_{zz} \end{bmatrix}_{N\times N} , \boldsymbol{G} = \begin{bmatrix} \boldsymbol{0} & \boldsymbol{g} \\ -\boldsymbol{g} & \boldsymbol{0} \end{bmatrix}_{N\times N} , \boldsymbol{K} = \begin{bmatrix} \boldsymbol{k}_{yy} & \boldsymbol{k}_{yz} \\ \boldsymbol{k}_{zy} & \boldsymbol{k}_{zz} \end{bmatrix}_{N\times N} , \tag{2}$$

$$\boldsymbol{F} = \begin{Bmatrix} \boldsymbol{f}_y(t) \\ \boldsymbol{f}_z(t) \end{Bmatrix}_{N\times 1} , \quad \boldsymbol{q}(t) = \begin{Bmatrix} \boldsymbol{q}_y(t) \\ \boldsymbol{q}_z(t) \end{Bmatrix}_{N\times 1}$$, where $N = 4n$, n is the number of nodes.

2.2 Receptance and dynamic stiffness

The equation of motion (1) can be rewritten in state space form as

$$\boldsymbol{A}\dot{\boldsymbol{X}} + \boldsymbol{B}\boldsymbol{X} = \boldsymbol{Q} \tag{3}$$

where

$$\boldsymbol{A} = \begin{bmatrix} C+\Omega G & M \\ M & 0 \end{bmatrix}_{2N\times 2N} , \boldsymbol{B} = \begin{bmatrix} K & 0 \\ 0 & -M \end{bmatrix}_{2N\times 2N} , \boldsymbol{Q} = \begin{Bmatrix} F \\ 0 \end{Bmatrix}_{2N\times 1} , \boldsymbol{X} = \begin{Bmatrix} q \\ \dot{q} \end{Bmatrix}_{2N\times 1}$$

The $2N \times 2N$ matrices A and B are real but in general indefinite, nonsymmetric. The resulting system of equations (3) gives nonself-adjoint eigenvalue problem. In the case of the synchronous excitation

$$\boldsymbol{F} = \boldsymbol{A}_F\, e^{j\Omega t} , \quad \boldsymbol{q} = \boldsymbol{A}_q\, e^{j\Omega t} \tag{4}$$

transforming Eq. (3) into frequency domain, we obtain

$$\boldsymbol{A}_X = \boldsymbol{R}_d\, \boldsymbol{A}_Q , \quad \boldsymbol{A}_X = \begin{Bmatrix} \boldsymbol{A}_q \\ j\Omega \boldsymbol{A}_q \end{Bmatrix} , \boldsymbol{A}_Q = \begin{Bmatrix} \boldsymbol{A}_q \\ \boldsymbol{0} \end{Bmatrix} \tag{5}$$

where the matrix $\boldsymbol{R}_d$ is receptance matrix

$$\boldsymbol{R}_d = (j\Omega \boldsymbol{A} + \boldsymbol{B})^{-1} , \ (j = \sqrt{-1}) \tag{6}$$

By matrix operational transform the receptance becomes

$$\boldsymbol{R}_d(\Omega) = \boldsymbol{U}(j\Omega \boldsymbol{a} + \boldsymbol{b})^{-1}\boldsymbol{V}^{\boldsymbol{T}} \tag{7}$$

where $\boldsymbol{U} = [\boldsymbol{u}_1\, \boldsymbol{u}_2 \,.....\, \boldsymbol{u}_{2N}]$, $\boldsymbol{V} = [\boldsymbol{v}_1\, \boldsymbol{v}_2 \,.....\, \boldsymbol{v}_{2N}]$ are the $2N \times 2N$ matrices of right and left eigenvectors.
Next, let us introduce the dynamic stiffness matrix $\boldsymbol{K}_d$, defined as the inverse of receptance matrix

$$\boldsymbol{K}_d = \boldsymbol{R}_d^{-1}(\Omega) \tag{8}$$

From the Eq. (5) and (7) we obtain

$$\boldsymbol{A}_q = \sum_{r=1}^{2N} \frac{\boldsymbol{u}_r^* \boldsymbol{v}_r^{*T}}{j\Omega a_r + b_r} \boldsymbol{A}_F \ , \boldsymbol{v}_r^T \boldsymbol{A} \boldsymbol{u}_r = a_r \ , \boldsymbol{v}_r^T \boldsymbol{B} \boldsymbol{u}_r = b_r \tag{9}$$

where $\boldsymbol{u}_r^*$ and $\boldsymbol{v}_r^*$ are the upper halves of the corresponding modal vectors.

3. OPTIMIZATION

3.1 Objectiv functions and design parameters

In this section, based on the modal analaysis, we propose an external (passive control) optimization model for spindle-bearing systems. The goal being the diminishing the vibrations by the maximizing of the dynamic stiffness, i.e. by minimizing of the receptance. To do this we need to find out the design parameters: the position of the bearings, the diameters of the shaft (different diameters for several segment of the shaft). Therefore, the code computer optimization program in MATLAB is obtained by the coupling of the FEM with the nonlinear optimization methods with constraints [5]. The SQP algorithm is used to optimize the bearing locations. The numerical differentiation and a Newton method are used to calculate the Hessian matrix, and BFGS (Boyden-Fletcher-Goldfarb-Shanno) algorithm is used to update the Hessian matrix. In the case of synchronous excitation the objective function is the receptance for a given rotating speed, or the average receptance for an interval of rotating speeds. The optimization problem obtained is

$$\begin{aligned} &\min_{s_k, d_k} \frac{1}{\Omega_1 - \Omega_2} \int_{\Omega_1}^{\Omega_2} \frac{A_u}{A_F} d\Omega \\ &s_k^i \le s_k \le s_k^s \\ &d_k^i \le d_k \le d_k^s \\ &\Omega \in (\Omega_1, \Omega_2) \\ &V = const. \end{aligned} \tag{10}$$

The design parameters are the distances s_i between the bearings and the diameters d_i of the different portions of the shaft. Assume shaft type Timoshenko with gyroscopic effects included.
In the above equations A_u is the amplitude of the displacement, A_F is the force amplitude, Ω is the rotor spin speed and ω is the whirl speed. The objective function is a measure of dynamic stiffness defined by relation (8). The authors elaborated several computer codes in MATLAB programming language.

3.2 Numerical example. Optimization of bearing locations

The design variables are bearing spans s1, s2, s3 and s4. In the numerical simulations, the same numerical data set, as in the paper [9], has been used, for compare sake. Fig. 1 shows the design variables for the motorized spindle with five bearings. The main spindle specifications of SH-403 are shown in [7].
The maximum spindle speed is 20,000 rpm and the power and torque properties of the spindle motor are set from the data shown in [7].

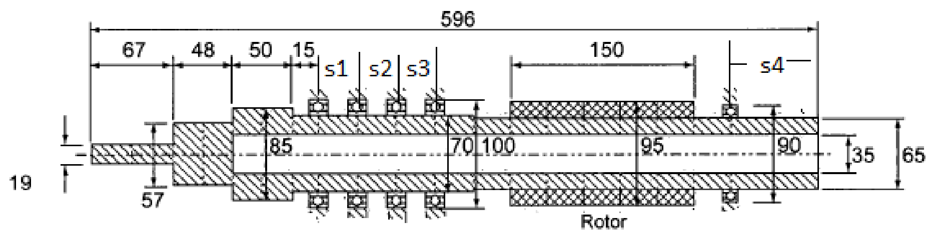

Figure 1: Design variables for the motorized spindle with five bearings

The material parameters: E = 2.07e11; Poisson = 0.3; G = E/(2*(1+Poisson)); rho = 8300;

Optimization results:
Natural frequencies and response for optimal configuration:

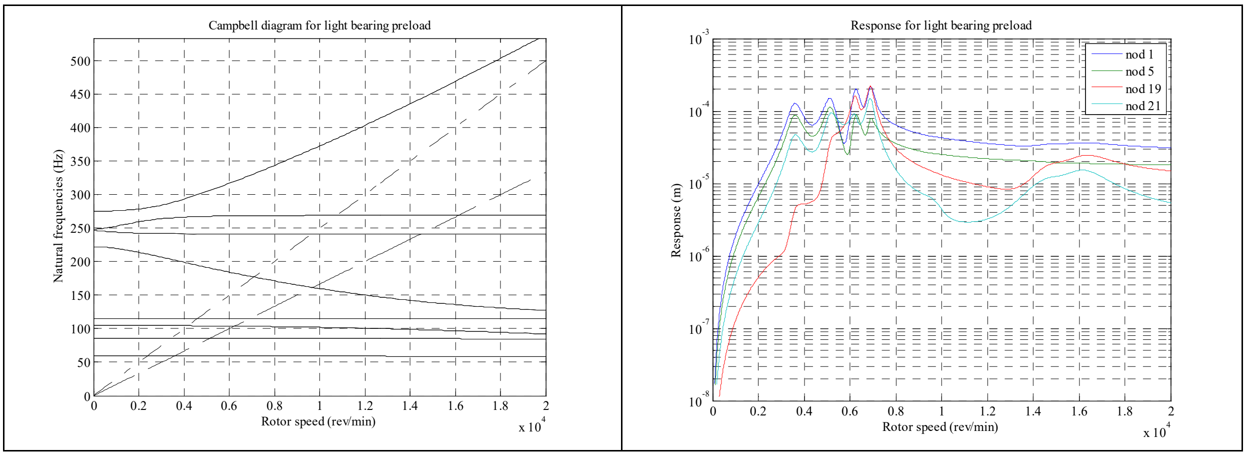

Figure 4: The Campbell diagram and response for bearing leigth preload

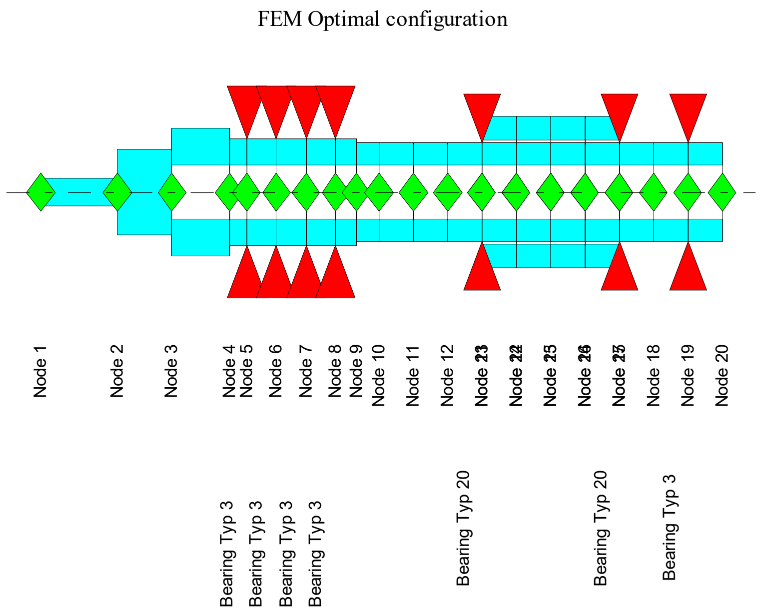

Figure 3: Optimal configuration spindle-holder-tool system

Optimal bearings pozitions: [5 0.180; 6 0.206; 7 0.232; 8 0.258; 19 0.566; 20 0.596; 21 0.386; 23 0.446];

4. CONCLUSIOS

The distance between the bearings and bearing preload has considerable influence on the stiffness of the spindle. In this paper we propose an external (passive control) optimization model for spindle-bearings systems. The goal being the diminishing the vibrations by the maximizing of the dynamic stiffness, i.e. by minimizing of the receptance. The paper proposes a bearing spacing optimization strategy. The spindle is analyzed by a proposed Finite Element Method (FEM) algorithm based on Timoshenko beam elements. Therefore, the code computer optimization program in MATLAB is obtained by the coupling of the FEM with the nonlinear optimization methods with constraints. The proposed system is demonstrated against a commercially existing machine tool (Mori Seiki SH-403

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