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OPTIMUM DESIGN OF SPINDLE-BEARING SYSTEMS

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Abstract: *In this paper we proposed several optimization models for spindle-bearing systems. 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 affect the machining productivity and quality of the work pieces. 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 previous research used only two support bearings, although practical spindles may use more bearings depending on the machining application. In addition, most of them optimize design parameters, such as shaft diameter, bearing span, and bearing preload, to minimize the static deflection.
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 several optimization models for spindle-bearing systems.
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], [10]. Spindle, holder, and tool are modeled as multi-segment beams by using Timoshenko beam theory. In the case of the dynamic analysis four degrees of freedom (DOF) per node

are considered: two displacements and two slopes. The linearized bearing are commonly modeled as four spring coefficients and four damping coefficients.
Based on the modal analysis we propose an "external" (passive) optimization model for spindle-bearing systems.

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 and Euler–Bernoulli beam model is used. The results are compared.
The model consists of a spindle treated as a continuous elastic shaft supported on isotropic or anisotropic elastic bearings. Consider that the dynamic equilibrium configuration of the spindle-bearing system the undeformed shaft is along the *x*- direction of an inertial *x, y, z* coordinate system. In the study of the lateral motion of the spindle, the displacement of any point is defined by two translations (v, w) and two rotations (φ_y, φ_z). In the following, only axisymmetric spindles are considered. The model could use one of the following two beam finite element types [5]:

- Beam C^1 finite element type based on Euler-Bernoulli beam model;
- Beam C^1 finite element type based on Timoshenko beam model;

The beam finite element has two nodes. In the case of the dynamic analysis four degrees of freedom (DOF) per node are considered: two displacements and two slopes measured in two perpendicular planes containing the beam. We do a comparative study of the two proposed models and on its basis we adopt the optimal model of the goal. Timoshenko beam model is finally adopted as the beam might be short and therefore the effect of the shear force must be considered. The gyroscopic effect and damping in bearings may be taken into account. The liniarized bearing are commonly modeled as four spring coefficients and four damping coefficients.

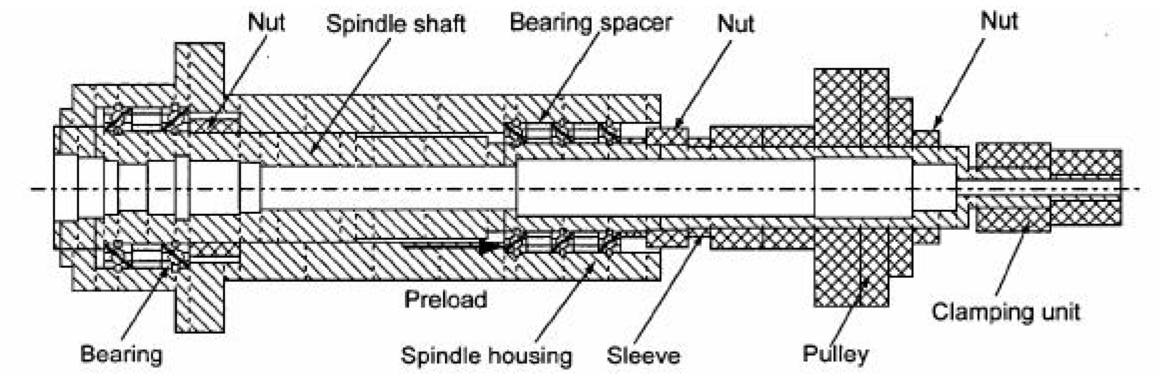

Figure 1: Spindle system

The equation of an anisotropic spindle-bearing systems which consists of a flexible nonuniform shaft and anisotropic bearings may be written as [4], [5]

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

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

$$\boldsymbol{M} = \begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix}_{N\times N}, \quad \boldsymbol{C} = \begin{bmatrix} c_{yy} & c_{yz} \\ c_{zy} & c_{zz} \end{bmatrix}_{N\times N},$$

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

$$\boldsymbol{F} = \begin{Bmatrix} \boldsymbol{f}_y(t) \\ \boldsymbol{f}_z(t) \end{Bmatrix}_{N \times 1} , \quad \boldsymbol{q}(t) = \begin{Bmatrix} \boldsymbol{q}_y(t) \\ \boldsymbol{q}_z(t) \end{Bmatrix}_{N \times 1}$$

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

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

By matrix operational transform the receptance becomes

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

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 analysis, we propose an external (passive) 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 [10]. 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

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

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. 3 shows the design variables for the motorized spindle with five bearings. The main spindle specifications of SH-403 are shown in [7].

The proposed system is demonstrated against a commercially existing machine tool (Mori Seiki SH-403) as shown in Fig. 2 for comparison. The spindle has a motorized transmission with oil–air type lubrication with four bearings at the front and one at the rear. 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].

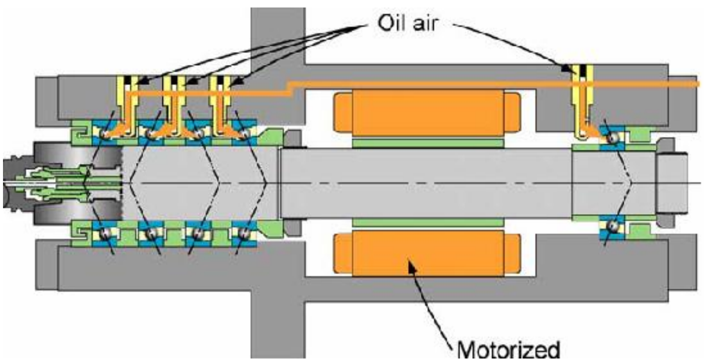

Figure 2: Mori Seiki SH-403 spindle system

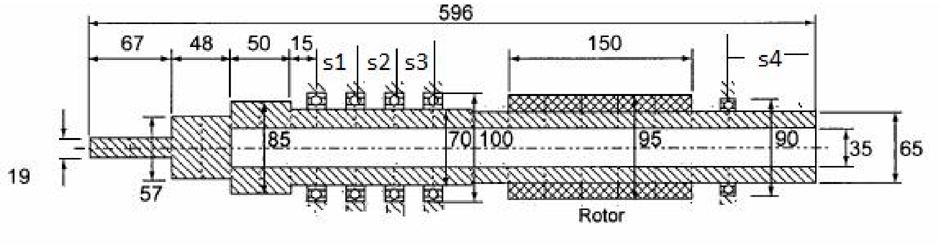

Figure 3: Design variables for the motorized spindle with five bearings

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

Optimization results:

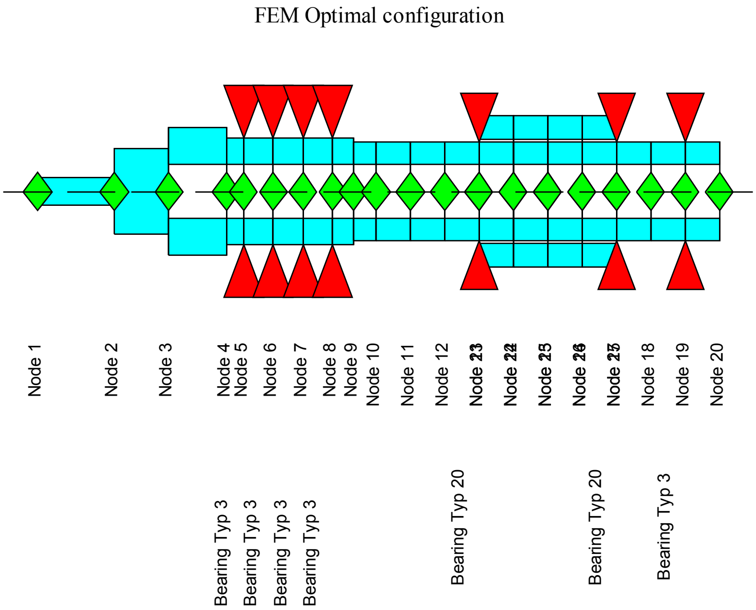

Figure 4: Optimal configuration spindle-holder-tool system

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];

Natural frequencies and response for optimal configuration:

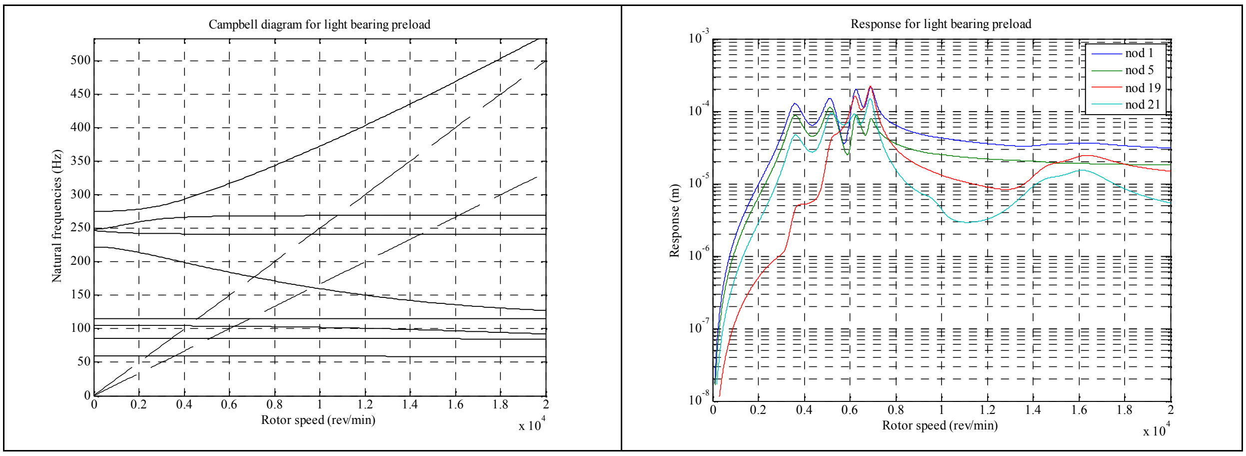

Figure 5: The Campbell diagram and response for bearing leigth preload

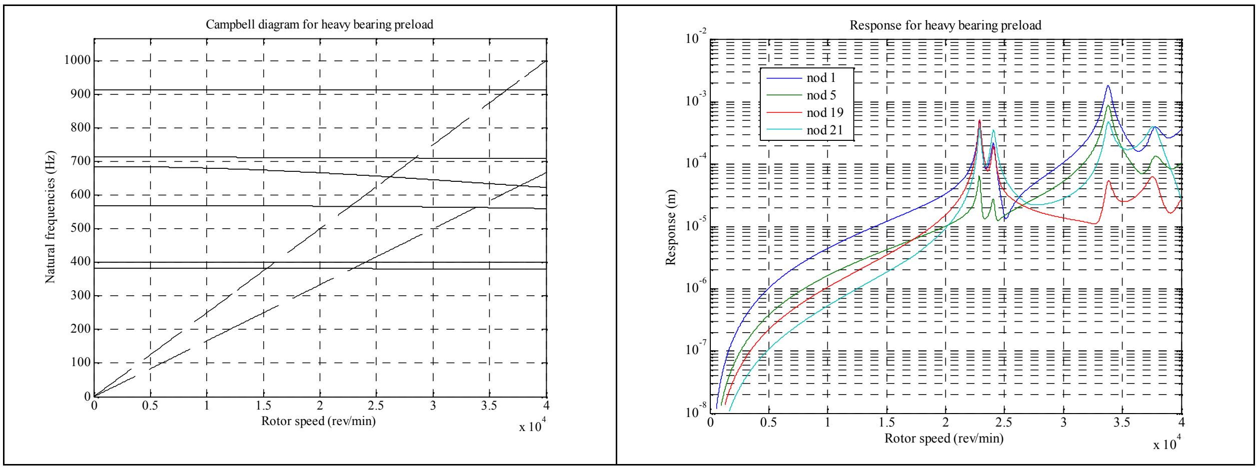

Figure 6: The Campbell diagram and response for bearing heavy preload

3. CONCLUSION

The static and dynamic behavior of machine tools is influenced significantly by the design of the spindle and its bearings. The distance between the bearings and bearing preload has considerable influence on the stiffness of the spindle. It is often important to consider the dynamic behavior of a spindle before establishing an optimum bearing span. In this paper we propose an external (passive) 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). The spindle has a motorized transmission with oil–air type lubrication with four bearings at the front and one at the rear. 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].

REFERENCES

[1] S. Yang, S., A study of the static stiffness of machine tool spindles, International Journal of Machine Tool Design & Research 21 (1), 23–40, 1981.

[2] Taylor, S., Khoo, D. B. T., Walton, Microcomputer optimization of machine tool spindle stiffnesses, International Journal of Machine Tools and Manufacture 30 (1), 151–159,1990.

[3] Wang, W. R., Chang, C. N., Dynamic analysis and design of a machine tool spindle-bearing system, Journal of Vibration and Acoustics, Transactions of the ASME 116 (3), 280–285, 1994.

[4] Hagiu, G., Gafiteanu, M. D., Preload optimization: a high efficiency design solution for grinding machines main spindles, ASTM Special Technical Publication, No. 1247, 1995.

[5] Nicoara, D., Optimizarea sistemelor mecanice. Aplicatii la sistemele rotor-lagare, Editura Universitatii Transilvania din Brasov, 2003.

[6] Y. Altintas, Manufacturing automation-metal cutting mechanics, Machine Tool Vibrations, and CNC Design, Cambridge University Press, Cambridge, 2000.

[7] SH-403, MORI SEIKI CO., LTD.

[8] Vanderplaats, S.V., Numerical Optimization Tehniques for Engineering Designe: with Applications, McGraw-Hil, New York, 1984.

[9] Maeda, O., Cao, Y., Altintas Y., Expert spindle design system, International Journal of Machine Tools & Manufacture 45, 537–548, 2010.

[10] Ertὺ rk , A., et al., Analytical modeling of spindle–tool dynamics on machine tools using Timoshenko beam model and receptance coupling for the prediction of tool point FRF, International Journal of Machine Tools & Manufacture 46, 1901–1912, 2006.