ANALYSIS STATIC AND DYNAMIC BEHAVIOR OF HYDRODYNAMIC SPINDLE

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Abstract: Becoming more stringent demands with regard to quality of ball bearings, as well as complexity of physical and chemical process that occur inside of them, which cannot be explained based on theoretical knowledge, caused intensive development of experimental methods and devices for testing and diagnostic of rolling bearings. Experimental tests are carried out on special for this purpose intended devices. The most important element of device is hydrodynamic spindle from who depend the accuracy of device. The objective of this work is to study the static and dynamic behavior of spindle supported by hydrodynamic bearings, using the FEM method.

Key words: hydrodynamic spindle, hydrodynamic bearings, static analysis, dynamic analysis, CAE

1. INTRODUCTION

Experimental results have a major impact both on the resolving existing problems and the development of new designs and technologies throughout the life of the roller bearings, therefore is very important that results are accuracy [1]. The hydrodynamic spindle system is the most important element of a measuring device since its properties are closely related to the accuracy of the device, because during the test roller bearing is placed on spindle and every imperfection of spindle will reflect on results. The dimension of the spindle as well as the location, the stiffness of the hydrodynamic bearings, loads of tourniquet and driving system affect the spindle behavior. For the hydrodynamic bearings, the journal and the bearing surface are separated by the sliding action with a wedge pressure-generating mechanism to develop a pressure within the bearing [2]. So that stiffness depends on the pressure and thickness of the lubrication film [3-4]. Spindle used in precision devices must have low error motion over a range of operation speeds, small temperature rise and minimum wear. The above requirements can be achieved by appropriate choice material and construction of the spindle and its bearings [5]. Hydrodynamic bearings are characterized by very high accuracy of work, running smoothness with high vibrations dumping, simply technology and low cost of making, economically maintenance and so on.

The purpose of static analysis is validation that the spindle deformations are in allowed limits and finding the force in bearings. On the other hand dynamic analysis is made with aim to analyze the dynamic behavior, for what is used modal analysis to determinate natural frequencies and mode shapes as well as harmonic analysis to determine the response of certain nodes at the effect of the load. Both analyzes are carried out using the finite element method.

2. STATIC AND DYNAMIC ANALYZES

The spindle is mounted in a housing of device via two radial and one axial hydrodynamic bearing for satisfying the high precision rotating of spindle. The Fig 1 shows the 3D model of hydrodynamic spindle system with marked characteristic parts.

Fig. 1. 3D CAD model of hydrodynamic spindle system [6]

Due to simple Axial-symmetric nature of the spindle system, the spindle is represented by one dimensional element type with two nodes BEAM188, which is six degree element, based in Timoshenko beam theory, with circular annular cross-section, which
is suitable for representing the stepped nature of spindle. Stiffness of hydrodynamic bearings is defined by equivalent spring-dumper element type COMBIN14 whereby value of initial stiffness both hydrodynamic bearings are Cr=70 N/\(\mu\)m [7]. The front conical surface of the spindle which is used to accept and positioning the arbor is replaced with cylindrical surface with the dimension of middle diameter of cone. The Fig. 2a shows model described with 16 nodes and 15 elements type BEAM 188 and 2 elements type COMBIN14 (Fig. 2b).

In reality, the spindle has a total length of 324 mm and its outside diameter is variable, on place where is spindle connected with bearings diameter is 60 mm. The spindle is connected with front and rear bearing with the isotropic bearing stiffness designated as nodes 15 and node 16 respectively.

Using ANSYS APDL, a static model was built, consisting of a solid shaft rigidly connected to arbor and belt pulley and supported by the bearings hydrodynamic bearings. The Fig. 3 shows model for analysis, the same model will be used for static and dynamic analysis, only the constraints will be changed.

Material of spindle is 20MnCr5, and his data are specified in Table 1.

### Table 1. Spindle material data

<table>
<thead>
<tr>
<th>Material</th>
<th>20MnCr5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Modulus of elasticity</td>
<td>2.1E+05 MPa</td>
</tr>
<tr>
<td>Density</td>
<td>7.8E-06 kg/mm³</td>
</tr>
<tr>
<td>Tensile strength</td>
<td>800 MPa</td>
</tr>
<tr>
<td>Poisson Ratio</td>
<td>0.3</td>
</tr>
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</table>

#### 2.1 Static analysis

The static analysis is carried out with radial load on node of finite element mesh where is spindle in contact with belt pulley (node 14), and axial load on spindle nose (node 1). Applied forces are the result of the maximal axial force (Fa=200 N), which acting on the test bearing and the radial force (Fr=430 N) by the belt transmissions.

The Fig. 4 shows deformations in Y direction, the biggest value of deformations is on the end of spindle, showed by blue color, what was expected because that is the place of radial force. The maximum of radial displacement occurs at the end of the spindle and is 16.9 \(\mu\)m, while the minimum radial displacement occurs at the spindle nose and is 0.5 \(\mu\)m (Fig. 4). On the place where is spindle connected with a front hydrodynamic bearing deformation is 5.0 \(\mu\)m, and on the rear bearing place is 12.0 \(\mu\)m. Hydrodynamic bearings are designed with clearance of 20 \(\mu\)m, so then mentioned displacements on front and rear bearing will not disturb proper work. Force reactions in bearings are: front and rear bearing Ffb=203N, and Frb=633N respectively.

Material of spindle is 20MnCr5, and his data are specified in Table 1.

\[
C_a = \frac{F_a}{\delta_a} \times \frac{1}{N/\mu m} \\
C_a = \frac{200}{1.8} \times 111; N/\mu m
\]
On the basis of displacement from Figure 4, radial stiffness of the front \(C_{r,f}\) and rear bearing \(C_{r,r}\), under applied the radial load are:

\[
C_{r,f} = \frac{203}{5} = 40.6 \, \text{N/µm}
\]

\[
C_{r,r} = \frac{633}{12} = 52.7 \, \text{N/µm}
\]

Similar stiffness values for the given displacements are shown in the paper [1].

### 2.2 Dynamic analysis

For dynamic analysis spindle is freely supported, because mode frequencies as well as the natural frequencies are depend on the system parameters. In order to find the spindle natural frequencies domain of the spindle was used finite element model represented in Fig. 5. Two nodes 15 and 16 are taken all degrees of freedom, to node 3 where is spindle connected with front bearing is taken rotational degree around X axis, and translation in Z direction, while to node 10 are taken translations in X and Z directions, and rotation around X axis.

![Fig. 5. Finite element model for dynamic analysis with constraints](image)

Reason for that kind of limitations is radial on front and radial with axial bearings on rear side of spindle.

Accordingly numerical models were obtained the results for the first five natural frequencies of model (represented in Fig. 5.) values are given in Table 2.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency [Hz]</th>
</tr>
</thead>
<tbody>
<tr>
<td>(f_{01})</td>
<td>355</td>
</tr>
<tr>
<td>(f_{02})</td>
<td>504</td>
</tr>
<tr>
<td>(f_{03})</td>
<td>5088</td>
</tr>
<tr>
<td>(f_{04})</td>
<td>5580</td>
</tr>
<tr>
<td>(f_{05})</td>
<td>9000</td>
</tr>
</tbody>
</table>

Table 2. Natural frequencies of the spindle

Based on the calculated values, plotted are the mode shapes for corresponding natural frequencies. Vibration modes according third and fifth natural frequencies are represented, Fig 6 shows third, and Fig. 7 shows fifth mode shapes. We can observe that the first spindle natural frequency of the spindle is about 355 Hz corresponding to 21300 rpm, and it is much higher than maximum spindle speed which is 1800 rpm.

![Fig. 6 Third natural frequencies of the test spindle](image)

![Fig. 7 Fifth natural frequencies of the test spindle](image)

The spindle is modeled as a stationary reference frame where the general dynamic equation includes the rotational effects as follows:

\[
[M]\ddot{u} + [G]u + [K]u = F
\]

(3)

Where:

- \([M]\) = Structural mass matrix
- \([G]\) = Gyroscopic effect originated from the rotational angular velocity applied to the structure
- \([K]\) = Stiffness matrix
- \([F]\) = External force vector

In this paper, damping is not considered.

In the considered frequency interval \(0 \div 10000 \, \text{Hz}\), given by modal analysis, was performed harmonic response analysis spindle at the effect of loads. The Fig. 8 shows the displacement of characteristic nodes of spindle.

The Fig. 8 shows harmonic response of the characteristic nodes, and can be seen that maximum responses are different for each node. All the maximum values are taking place above operational frequency of 30 Hz what corresponding to 1800 rpm. So in the normal conditions, the deformations will be with the limits.

![Fig. 8 Characteristic nodes displacement](image)
3. CONCLUSION

The spindle made of multiple steps is modeled and analyzed for different boundary conditions for static and dynamic analysis. The analysis summary is as follows.

Initial static analysis shows minimum, maximum and deformations of significant parts of spindle, as well as the forces in both hydrodynamic radial bearings. The value of spindle stiffness is validated through theoretical calculations.

Modal analysis carried out with bearing supports under flexible conditions shows that natural frequencies of spindle are a much bigger than the operational frequency, so that they can not endanger measuring process.

Harmonic response analysis of characteristic nodes represented along with natural frequencies, shows that the maximum values of displacement are different for each node, because they are depended of vibration bending mode.

All the results shows that device can operate safe and accurately with the given bearing values and spindle dimensions.

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4. REFERENCES


Authors: M.Sc. Miloš Knežev, Assistant Professor, Aleksandar Zivković, M.Sc. Cvijetin Mladenović, University of Novi Sad, Faculty of Technical Sciences, Department for Production Engineering, Trg Dositeja Obradovica 6, 21000 Novi Sad, Serbia, Phone: +381 21 485 2330, Fax: +381 21 454-495.
E-mail: milosknezev@gmail.com, acoz@uns.ac.rs, mladja@uns.ac.rs.