Numerical Model for Thermo-Mechanical Spindle Behavior

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Abstract: - The main spindle has a strong influence on the metal removal rate and quality of the machined pieces and therefore it requires rigidity and stable thermal behavior. The paper presents a simulation model for predicting the effects of the temperature distribution and of the vibrations in the spindle and rolling bearings generated by the operation conditions. The heat generated in the bearings and motor is transferred by conduction and thermal radiation to the motor housing and spindle structure leading to the thermal expansion of the spindle parts. The finite element method controlled by an ANSYS program was used to highlight the thermo/dynamic behavior in different cases, free and processing. To validate the results of the numerical simulation used for determining the thermal and dynamic behavior, an experimental protocol is achieved.

Key-Words: - Spindle, Angular bearings, Stiffness, Preload, Temperature

1 Introduction

The impact of the thermal effect on the machine tool processing accuracy has been increasingly dealt with since 1960 [1]. Consideration has been given to many procedures to reduce the thermal error, such as: the machine tool design taking into consideration the main heat sources, the cooling of the unit, the compensation procedures for the thermal error [2-9]. Recently it has been shown that the thermal error of a machine tool is strongly dependent on the specific operation parameters and the conditions the machine is setup in order to generate a specific thermal condition [7]. So, the theoretical accurate modeling of thermal errors becomes indispensable.

The major heat amount resulted from the machine tool operation is generated by the bearings and the cutting processes. It is assumed that most of the cutting heat is taken away by the coolant and the heat generated by bearings is the dominant heat source causing the thermal distortions. Therefore the calculation of temperature in the ball bearings under friction has been of great scientific interest over the last years.

The development of the high speed machining and the thermal characteristics of machine tools have been given much consideration by many researchers. The thermal behavior is one of the important factors that affect the performances of machine tool systems. There are many thermal or thermo-mechanical models [10-13] available to investigate the thermal behavior of machine tool spindles.

This paper presents a simulation model to predict the effects of the temperature distribution in the spindle and rolling bearings, temperature generated by the operation conditions, using ANSYS program, based on the finite element method.

Simulations of the thermal transient’s analysis have also been performed. The static, thermal characteristics of the motorized spindle were analyzed and verified by the simulation results.

2 Research Formulation

In the today research developments concerning the spindle lifetime the thermal behavior represents one of the most important factors. The mechanical constraints on radial and axial directions coupled with the dynamic effects require a thorough understanding of the phenomenology. For that purpose a numerical
model of the thermal behavior is being developed. The spindle used for this research is a belt driven type spindle.

3 Experimental Procedure

The testing configuration was designed to obtain a dynamic and thermal characterization using a rotating spindle driven by a synchronous motor and a transmission belt. The thermal behavior is highlighted using thermocouple sensors for each bearing while the dynamic behavior is revealed by a tri-axial accelerometer fixed on the front bearing and one axial accelerometer fixed on the rear bearing. The rotation speed is obtained with a laser tachometer. The signal acquisition and processing are done by a Fastview Digitline equipment that allows synchronization of the vibrations and temperature signal with the rotational speed.

Fig. 1 Experimental setup for main spindle

4 Model Description

The heat is mainly generated at bearing raceways and balls due to the friction influenced by speed, preload and lubricant; the bearing temperature distribution rises due to the heat generated by friction losses. The following equation gives the heat in the bearing [10, 26].

\[ N_R = M_R \cdot \frac{n}{9500} \]  

(1)

The total friction of a ball bearing under preload, lubricant and speed conditions is given by the sum of the load torque and viscous friction torque.

\[ M_R = M_0 + M_1 \]  

(2)

In the below equation \( M_0 \) is the viscous friction torque as a function of speed and can be expressed as [1]:

\[ M_0 = 10^{-7} f_0 \cdot (v \cdot n)^{2} d_m^{3} \quad v \cdot n \geq 2000 \]  

(3)

\[ M_0 = 160 \cdot 10^{-7} f_0 \cdot d_m^{3} \quad v \cdot n < 2000 \]  

(4)

in which parameter \( f_0 \) is a factor depending on the bearing and lubrication type, and for the angular contact ball bearing \( f_0 = 2 \), \( v \) is the kinematics viscosity of lubricant at operation temperature.

\( M_1 \) is the load torque of the mechanical friction and it is a function of load for ball bearings with spherical roller:

\[ M_1 = f_1 P_r d_m \]  

(5)

Where \( f_1 \) is a bearing factor for frictional torque as a function of preload, \( P_r \) depends on the value and direction of the load, \( d_m \) is the mean bearing diameter depending on the bearing design.

For angular contact ball bearing:

\[ f_1 = 0.001 \left( \frac{P_0}{C_0} \right)^{0.33} \]  

(6)

for single bearing

\[ P_1 = F_a - 0.1 \cdot F_r \]  

(7)

for bearing pair

\[ P_1 = 1.4 \cdot F_a - 0.1 \cdot F_r \]  

(8)

where: \( C_0 \) is basic rating static load, \( P_0 \) is equivalent static radial load, \( F_a \) axial load and \( F_r \) radial load.

\[ P_0 = X_s \cdot F_r - Y_s \cdot F_a \]  

(9)

where: \( X_s \) and \( Y_s \) values for the single row angular contact ball bearings with different contact angles are given in Table 1 [1].

Table 1 Values of \( X_s \) and \( Y_s \) for angular contact ball bearings.

<table>
<thead>
<tr>
<th>Contact angle</th>
<th>15°</th>
<th>20°</th>
<th>25°</th>
<th>30°</th>
<th>35°</th>
</tr>
</thead>
<tbody>
<tr>
<td>( X_s )</td>
<td>0.58</td>
<td>0.5</td>
<td>0.5</td>
<td>0.5</td>
<td>0.5</td>
</tr>
<tr>
<td>( Y_s )</td>
<td>0.47</td>
<td>0.42</td>
<td>0.38</td>
<td>0.33</td>
<td>0.29</td>
</tr>
</tbody>
</table>

The parameters of the spindle and for one front and two rear bearings mounted in configuration “O” are listed in Tables 2 and 3, respectively. The bearings are lubricated with Kluberspeed BF 42-12 lubricant, and Table 4 gives the base oil viscosity–temperature relationship.
Table 2 Parameters of the test spindle.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotational speed (rpm)</td>
<td>0-4500</td>
</tr>
<tr>
<td>Bearing span (mm)</td>
<td>322</td>
</tr>
<tr>
<td>Max diameter of shaft(mm)</td>
<td>100</td>
</tr>
<tr>
<td>Length of shaft(mm)</td>
<td>476</td>
</tr>
<tr>
<td>Preload</td>
<td>570</td>
</tr>
</tbody>
</table>

Table 3 Parameters of ball bearings.

<table>
<thead>
<tr>
<th>Bearing</th>
<th>Front bearing</th>
<th>Rear bearing</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>B7211-C-T-P4S</td>
<td>B7208-C-T-P4S</td>
</tr>
<tr>
<td>Material</td>
<td>Steel</td>
<td>Steel</td>
</tr>
<tr>
<td>Inner diameter (mm)</td>
<td>55</td>
<td>40</td>
</tr>
<tr>
<td>Outer diameter (mm)</td>
<td>100</td>
<td>80</td>
</tr>
<tr>
<td>Width (mm)</td>
<td>21</td>
<td>18</td>
</tr>
<tr>
<td>Ball diameter (mm)</td>
<td>14</td>
<td>13</td>
</tr>
<tr>
<td>Number of balls</td>
<td>14.3</td>
<td>11.1</td>
</tr>
<tr>
<td>Contact angle(deg.)</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td>Basic rating static load (N)</td>
<td>26500</td>
<td>16100</td>
</tr>
</tbody>
</table>

Table 4 The base oil viscosity–temperature relationship from DIN 51 802 of Kluberspeed BF 42-12.

<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>Kinematics viscosity (mm²/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>45</td>
<td>24</td>
</tr>
<tr>
<td>100</td>
<td>5</td>
</tr>
</tbody>
</table>

Fig. 2 Generated heat calculated in function of operating speeds.

5 Numerical Simulations

To calculate the heat generated in function of the operating range speed (500 - 4500 rot/min) and the constant 570N preload by the above equation (1), the experimental assembly was modelled employing the parametric Siemens NX 8.5 design software.

The parametric model was used as input data and discredited using the finite element method, also applying the hex dominant method as illustrated in Fig. 3.

![Fig. 3 Model meshed with hex dominant method](image)

In the motorized spindle the thermal transient’s analysis was performed and the heat values of the front and rear bearings were calculated with equation (1). The heat built-up rate on bearings depends on the operation speed (see Fig. 2).

In Fig. 4 the generated heat is applied on the contact area between the balls and the inner-outer ring. One of the applied free convection is 60 W/m² °C for the external surfaces of the spindle housing and 30 W/m² °C for the internal surfaces of the spindle.

![Fig. 4 Thermal load of the model simulation](image)

6 Results and Discussions

The testing results show a stable thermal behavior that points-out the optimum operation of the bearings. The test results presented in Fig. 5 and 6 indicate a temperature evolution characterized by a transition period that takes place in the first hour of the machine tool working time followed by a
constant temperature evolution. The temperature increases in time with the number of rotations up to 3750 rot/min. So, the front bearing temperature is about 36°C and the rear bearing temperature is about 33°C. The temperature gradient remains constant within the (3750 – 4500) rot/min operation speed range.

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The results of numerical simulations are presented in the following figures. Fig. 7 presents the maximum temperature in function of operation speed for the main components while Fig. 8 shows the maximum temperature of the main components at 3000 rot/min operation speed in function of time. Fig. 7 illustrates the stable operation temperature for all components after about 56 min of the machine working time.

The temperature distribution in the assembly at steady state for 3000 rot/min speed after one hour of numerical simulation is shown in fig 9. The two rear bearings and the spindle show the maximum temperature of the assembly.
Table 5. Residual values regarding to the experimental and simulated data.

<table>
<thead>
<tr>
<th>Operation speed (rot/min)</th>
<th>Temperature experimental (°C)</th>
<th>Temperature simulate (°C)</th>
<th>Residual values (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Front bearing</td>
<td>Rear bearing</td>
<td>Front bearing</td>
</tr>
<tr>
<td>500</td>
<td>25</td>
<td>27</td>
<td>25.3</td>
</tr>
<tr>
<td>1000</td>
<td>26</td>
<td>28</td>
<td>26.1</td>
</tr>
<tr>
<td>1500</td>
<td>26</td>
<td>28</td>
<td>27.03</td>
</tr>
<tr>
<td>2000</td>
<td>31</td>
<td>30</td>
<td>28.4</td>
</tr>
<tr>
<td>2500</td>
<td>33</td>
<td>31</td>
<td>29.9</td>
</tr>
<tr>
<td>3000</td>
<td>35</td>
<td>32</td>
<td>31.2</td>
</tr>
<tr>
<td>3500</td>
<td>36</td>
<td>32</td>
<td>33.6</td>
</tr>
<tr>
<td>4000</td>
<td>34</td>
<td>32</td>
<td>35.6</td>
</tr>
</tbody>
</table>

7 Conclusions

The results obtained using the ANSYS commercial program in order to simulate the thermal machine tool behavior are comparable with the experimental recorded data. The experimental data were obtained by temperature measurements conducted by a thermocouple placed on the spindle housing in the right of front and rear bearings and increasing the operation speed from 500 to 4170 rot/min. Employing ANSYS program the maximum temperature in front and rear bearings at operation speed range of (500 – 4500) rot/min was calculated taking into consideration the details presented in this paper. From the data presented one can see that the temperature increases by about (1 – 10) °C at rear bearing and by (1 – 6) °C at front bearing for (500 – 4170) rot/min operation speed range. Also, during the operation of the spindle, the front bearing temperature reaches even 36 °C while the rear bearing maximum temperature reaches 33 °C.

The lowest residual value between the experimental and the simulated data was obtained at 1000 rot/min and the maximum value, at 4000 rot/min.

Analyzing the experimental data and those obtained by simulation, the introduced method shows a good experimental data applicability. The research work described in this paper is a further contribution to the implementation of thermal error compensation and can be used for monitoring the bearing thermal condition in a production environment.

References:


[21] Y.-R. Jeng, C.-C. Gao, Investigation of the ball-bearing temperature rise under an oil–air lubrication system, Proceedings of the