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Thermal modelling of a high speed motor spindle

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Abstract

This paper presents a 3D FEM model to predict the thermal behaviour of a high speed motor spindle. This model allows a transient simulation which considers complex boundary conditions like heat sources as well as contact and convective heat transfer between spindle parts. Verification experiments are implemented on a test spindle. Compared with the measured data, it is shown that the developed model can predict well the thermal behaviour of the test spindle.

1. Introduction

High speed motor spindles are mainly developed for the machining of dies, moulds, aerospace parts and printed circuits. Enhancing the efficiency of compact drive motors, improving the accuracy and lifetime of bearings are main goals of spindle design. However, power losses by the motor and friction within the bearings generate an amount of heat, which is transferred to the ambient air, the coolant and the spindle structure. This leads to uneven thermal expansion of spindle parts and displacement of the tool centre point (TCP). Eventually, the thermal error of high speed motor spindles is very difficult to calculate efficiently, so experience in thermal behaviour is the key factor in the design process. Nevertheless, experience alone is often insufficient. Therefore, there is a critical need to develop a scientific and systematic modelling tool that can predict the thermal error of high speed motor spindles to assist industrial engineers in achieving an optimal design and error compensation [1].

Currently, none theoretical analysis can predict the temperature distribution and resulting thermal error of high speed motor spindles with sufficient accuracy. It is necessary to search for accurate functions and procedures to describe the energy losses, the geometry, the heat exchange as well as the dynamics and nonlinearity of the involved phenomena [2].

In this paper a holistic 3D FEM thermal model of a high speed motor spindle including the headstock body is presented and illustrated. This model characterizes quantitatively the major heat sources, the heat transfer between spindle parts and allows for a precise transient analysis of the thermal behaviour.

2. Thermal model of the high speed motor spindle

2.1. The spindle setup

Fig. 1. Model of the high speed motor spindle including the headstock

The setup which has to be modelled is a high speed milling spindle including the headstock, equipped in a HSC vertical machining centre, as shown in Fig. 1. It is
derived by a permanent magnet synchronous motor, which has a maximum rotation speed of \( n = 18000 \text{ min}^{-1} \) and a maximum power of \( P = 16 \text{ kW} \). A water circuit cooling jacket is placed around the stator of the motor. Two sets of hybrid angular contact ball bearings are mounted on the spindle in an O arrangement according to the fixed-floating bearings principle, the fixed bearings near the spindle nose and the floating bearings behind the rear of motor. Oil-air lubrication is used for the bearings.

2.2. FEM model

The FEM model for the motor spindle is established using the FEM system Marc® & Mentat®, which is a combination of a heat transfer model and load-deformation model with thermal load, as shown in Fig. 2. 3D 8-node solid heat transfer elements that are compatible with stress analysis are used for the modelling of the spindle. The materials of all spindle parts are defined in the FEM model, as shown in table 1.

### Table 1. Materials of the motor spindle

<table>
<thead>
<tr>
<th>Spindle parts</th>
<th>Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft</td>
<td>42CrMoV4</td>
</tr>
<tr>
<td>Housing</td>
<td>GGG80</td>
</tr>
<tr>
<td>Headstock</td>
<td>ALSi9Cu3</td>
</tr>
<tr>
<td>Races of bearing</td>
<td>X30CrMoN</td>
</tr>
<tr>
<td>Rolling elements</td>
<td>Si3N4</td>
</tr>
<tr>
<td>Motor</td>
<td>FeSi, Cu</td>
</tr>
</tbody>
</table>

The model of transient heat transfer is based on the equation:

\[
\rho C_p \frac{\partial T}{\partial t} + \nabla \cdot q = Q \quad (1)
\]

where \( T \) is the temperature distribution in space and time, \( \rho \) is the mass density, \( C_p \) is the specific heat capacity, \( \nabla \) is the gradient operator, \( q \) is the energy flow density and \( Q \) is the energy density.

The stress analysis computes deformations of the spindle caused by thermal expansion. The general equation based on HOOKES’s law is as follows:

\[
\varepsilon = \alpha \cdot \Delta T \quad (2)
\]

where \( \varepsilon \) is the strain vector, \( \alpha \) is the thermal expansion coefficient.

Transient heat transfer is a boundary value problem, so proper boundary conditions must be prescribed to the spindle in order to obtain a realistic solution. Fig. 3 shows the heat sources and boundary conditions of the spindle. The main heat sources are the motor and bearings [1]. The boundary conditions include contact heat transfer within spindle parts and between spindle, spindle headstock and water cooling jacket for the motor stator as well as air convections at the gaps and convection to the ambient air.

Some assumptions are made for the thermal modelling:
- Heat radiation to the surrounding is ignored.
- The flow rate of the cooling fluid is large enough so that the fluctuating temperature of the fluid is not considered.
- Air friction loss is much smaller than other heat sources, so that it is ignored.
2.3. Heat generation in the motor

The power losses occurring in the permanent magnet synchronous motor are armature winding losses, armature and rotor core losses [3].

The armature winding losses $\Delta P_{Cu}$ are defined in [4] by:

$$\Delta P_{Cu} = sI^2Rk_R$$  \hspace{1cm} (3)

where $s$ is the number of phases, $I$ is the armature current, $R$ is the armature winding resistance and $k_R$ is the skin effect coefficient for armature conductors. The armature and rotor core losses $\Delta P_{Fe}$ can be calculated on the basis of the specific core losses:

$$\Delta P_{Fe} = \Delta p_{1/50}(f/50)^{4/3}k_aB^2m_c$$  \hspace{1cm} (4)

where $k_a$ is the factor taking the increase in losses due to metallurgical and manufacturing processes into account, $B$ is the magnetic flux density, $m_c$ is the mass of the core and $\Delta p_{1/50}$ is the specific core loss at $B = 1T$ and $f = 50Hz$.

2.4. Thermal model of bearings

The rolling bearings are modelled simply as a three layer hollow cylinder structure, in which the middle layer represents all rolling elements [5]. The heat is exchanged between the inner race, rolling elements and outer race through mainly conduction. The thermal contact conductance between rolling elements and races $h_b$ can be empirical obtained with the rotating speed:

$$h_b = z\sqrt{14 + 2hnv - 2ln\frac{d}{d_w}d_w^2}/2400$$  \hspace{1cm} (5)

$$v = n(d + d_w)/19099$$  \hspace{1cm} (6)

where $d_w$ is the diameter of rolling elements, $d$ is the bore diameter of bearings, $z$ is the number of rolling elements, $v$ is the speed of the rolling elements, $n$ is the rotation speed of the motor spindle.

There are two main sources of friction which occur in bearings: one is the load friction caused by rolling and the other by viscosity of the lubricant. HARRIS [6] gives empirical formulas to calculate the friction torque for a given roller of the bearing:

$$M_v = f_0(\eta m)^{2/3}d_m^310^{-7}$$  \hspace{1cm} (7)

$$M_i = f_1Q_id_m$$  \hspace{1cm} (8)

where $M_v$ is the frictional torque due to lubricant viscosity and $M_i$ is the frictional torque due to the applied load, $f_0$ and $f_1$ are bearing parameters dependent on the bearing and lubrication type, $\eta$ is the kinematic viscosity of the lubricant fluid, $d_m$ is the mean diameter of the bearing, $Q_i$ is the bearing load. Thus, the heat generated in bearings $\Delta P_b$ can be obtained from

$$\Delta P_b = n(M_v + M_i)/9550$$  \hspace{1cm} (9)

It is assumed that, half of the heat is distributed to the races, and the other half is applied to the rolling elements.

2.5. Convection boundary condition

The heat transfer coefficient $h_f$ [9] for convection is defined by

$$h_f = Nu k_f/d_h$$  \hspace{1cm} (10)

where $k_f$ is the thermal conductivity of the fluid, $Nu$ is the Nusselt number, and $d_h$ is the hydraulic diameter. For the description of free convection of the outer surface of the spindle stock, the value of $d_h$ is equal to the height of the spindle stock $l$.

The $Nu$ in this work is calculated based on different heat convection models listed in Table 2, where $Pr$ is the Prandtl number, $Ta$ is the Taylor number, $Re$ is the Reynolds number, $Gr$ is the Grashof number, $r$ is the radius of the rotating cylinder, $l_t$ is the length of the water tube, $w$ is the velocity of the water, $d_k$ is the loop diameter, $d_s$ is the diameter of the headstock, $g$ is the gravity of earth, $\beta$ is the thermal expansion coefficient and $\xi$ is the friction factor obtained from

$$\xi = 1/[0.79ln(Re) - 1.64]^2$$  \hspace{1cm} (11)
Table 2. Nusselt number formulations for the thermal model of the spindle

<table>
<thead>
<tr>
<th>Convection boundary conditions</th>
<th>Characteristic dimension</th>
<th>Convection model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Forced convection in the annular gap [7]</td>
<td>$d_h$</td>
<td>$Nu = 0.42(TaPr)^{0.29}$ with $Ta = 4n^2n^2/2\eta^2/3600, 0 \leq Ta \leq 10^5$</td>
</tr>
<tr>
<td>Forced convection in the water jackets of the motor [8]</td>
<td>$d_h$</td>
<td>$Nu = \left{ \frac{0.125\left(Re - 1000Pr\right)}{1 + 12.7\sqrt{0.125\left(Pr^{1/3} - 1\right)}} \right} \left[ 1 + \frac{(d_h/l_t)^{2/3}}{1 + 3.6(1 - d_h/d_0)(d_h/d_0)^{1/3}} \right]$ with $Re = \frac{vd_h}{\eta}$ for $2 \cdot 10^4 \leq Re \leq 1.5 \cdot 10^5$ and $5 \leq d_h/d_0 \leq 84$</td>
</tr>
<tr>
<td>Free convection on the headstock [9]</td>
<td>$l$</td>
<td>$Nu = \left{ \frac{0.825 + 0.387(GrPr)^{1/6}}{1 + (0.492/Pr)^{5/12}} \right}^2 + 0.97l/d_s$ with $Gr = \beta\rho\Delta Tl^3/\eta^2$ for $10^{-1} \leq GrPr \leq 10^{-2}$</td>
</tr>
</tbody>
</table>

2.6. Contact heat transfer at joints of spindle parts

It is considered that the heat conduction through the actual contact spots and through the fluid in the gaps are two significant components of the heat transfer through a joint [10]. Therefore, the thermal contact conductance at the contacting surface of the joints is given by

$$h_c = h_s + h_g$$  \hspace{1cm} (12)

where $h_s$ is the contact spots conductance, $h_g$ is the gap fluid conductance. COOPER ET AL. [11] developed a theoretical thermal contact conductance model for contacting surfaces. YOVANOVICH [12] re-examined the model and proposed a new, more accurate, correlation equation:

$$h_s = 1.25(k_s\tan\theta/\sigma)(P/H_c)^{0.95}$$  \hspace{1cm} (13)

where $k_s$ is the harmonic mean of the thermal conductivities of the joint, $\sigma$ and $\tan\theta$ are the effective Root Mean Square roughness and the effective mean absolute slope of the combined profile of the joint surfaces, respectively. The apparent contact pressure is $P$ and $H_c$ is the plastic contact hardness.

The simple correlation equation for the gap conductance model is given by YOVANOVICH [12, 13]:

$$h_g = \frac{k_gf_g}{B + \lambda\sigma}$$  \hspace{1cm} (14)

$k_g$ is the thermal conductivity of the gas, $f_g$ is the correlation factor, $\lambda$ is the relative mean plane separation, $B$ is the complex gas gap rarefaction parameter.

3. Transient simulation

Heat transfer analysis is used for computing the temperature development, while stress analysis predicts deformations caused by thermal expansion. In this section, as an example the constant rotation speed of $n = 12000 \text{ min}^{-1}$ during a given time of $t = 480 \text{ min}$ is simulated. It is assumed that the initial temperature of the spindle structure is $T = 18.5 \degree \text{C}$ and environment temperature of $T = 19 \degree \text{C}$ is a constant value during the simulation.

Fig. 4 shows the simulated temperature distribution of the motor spindle by the end of 480 minutes of rotation. The highest heating is to be expected in direct contact to the motor and bearings. The maximum temperature of $T = 50 \degree \text{C}$ occurs at the motor stator. However, temperatures of the housing and spindle stock are the lowest. This can be explained with the powerful water cooling, which dissipated the most amount of heat from the stator. In contract, the heat from the rotor and the bearings is transferred to the shaft and cannot be dissipated directly due to the insufficient cooling. Therefore, the shaft is heated highly.

Fig. 5 shows the simulated spindle deformation due to the temperature variation. In consequence of the unexpected heating of the shaft, the maximum displacements of $\Delta X = 2 \mu\text{m}$, $\Delta Y = 10 \mu\text{m}$ and $\Delta Z = 50 \mu\text{m}$ appear at the TCP after 480 minutes of rotation. Due to the thermo-symmetric construction of the considered machine tool, only a slight displacement arises in x-direction of the TCP.
4. Experimental verification

In order to verify the FEM model, local temperature changes and thermal displacements of the motor spindle were analysed using different measurement methods. Firstly, temperature variations around the motor spindle surface were determined with the help of a thermographic camera. Fig. 6 shows the thermographic image of the spindle by the end of 150 minutes of rotation with rotation speed of \( n = 15000 \text{ min}^{-1} \). The horizontal temperature distribution is mostly homogenous. A vertical temperature difference of \( \Delta T = 5 \, ^\circ\text{C} \) can be determined around the spindle structure. Due to water cooling, temperatures on the motor surface (middle side of the spindle) are lower than on bearing surfaces (upper and lower sides). This result validates the simulation model (Fig. 6 right).

In order to obtain accurate statements on heat conduction and temperature development on the spindle, four thermocouples were used to measure the temperature at defined positions of the spindle headstock. These positions are 1) the spindle end, 2) the floating bearing, 3) the spindle motor and 4) the fixed bearing, Fig. 7 a). In addition, TCP displacements were measured with the aid of laser triangulators, Fig. 7 b). The measurement took 480 minutes with a constant rotation speed of \( n = 12000 \text{ min}^{-1} \). The temperatures and displacements were monitored once every one minute.
Fig. 8. Comparison of measured and simulated temperature variations (top) and displacements of the TCP (bottom)

Fig. 8 (top) shows the comparison of measured and simulated temperature variations at the defined measuring positions. It can be seen clearly that simulated temperatures match with measured data very well. The temperature development shows that the highest heating is in direct proximity to the positions 1 and 4. The permanent operating temperature of about $T = 27.5 \, ^\circ C$ is reached after approximately 60 minutes at position 1. This is due to the close location to the spindle bearing. Similar temperature increases can be measured at position 2 at the upper side of the spindle headstock. However, the increase is slightly smaller. The temperature at position 3 is the lowest due to the water cooling of the motor. Thus, the temperature measurement with use of thermocouples validates the simulation results.

Fig. 8 (bottom) shows the comparison of measured and simulated displacements of the TCP in x-, y-, and z-direction. The displacement in z direction increases rapidly to the maximum value of $\Delta z = -50 \, \mu m$ after approximately 60 minutes. Subsequently, the negative displacement remains constantly until the end of measuring time. Similar developments arise in x- and y-directions. The maximum displacement in y-direction is approximately $\Delta y = 10 \, \mu m$ and $\Delta x = 5 \, \mu m$ in x-direction. There is only a small difference of $2 \, \mu m$ between the calculated and the measured displacements in x-direction. Therefore, the simulated result is very satisfying.

5. Conclusion

In this paper a holistic 3D FEM thermal model of a high speed motor spindle is presented, which characterizes quantitatively the heat sources in motor and bearings, the contact heat transfer at joints of the spindle parts, the convection heat transfer to water cooling, surrounding air and in the air gap. This model allows the prediction of transient temperature distributions and induced displacements. The validation of the presented model is executed by using temperature and displacement measurements of a test spindle. The comparison between measured and simulated results shows a very good agreement. In future simulation based sensitivity analyses focusing on the description a modelling of boundary conditions will be arranged in order to improve the simulation results.

References