An Experimental and Numerical Study of the Thermal Issues of a High-speed Built-in Motor Spindle

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ABSTRACT
The heat generated from a built-in motor and bearings in a high-speed spindle results in a rise in temperature which causes a severe deformation of the spindle. This leads to changes in the relative position of the components of the spindle head which in turn causes machining errors and possible damage to the spindle. In this study, experiments and simulations of a spindle with a built-in motor were carried out. The results showed that the primary deformation of the cutter endpoint of the spindle was in the Z-axis direction. Experimental measurements showed that after the spindle had been running at 18,000 rpm for 10 min, the bushing temperature reached 33.6 °C and the endpoint of the spindle had been extended by 11.18 μm. Numerical results also revealed that the thermal deformation of the spindle in the radial direction is mainly thermal expansion without appreciable bending deformation. The simulation model was further used to analyze the effects of the cooling channel design. It showed that the single or double helical cooling channels were less effective than the original reciprocating cooling channels. This is because the reciprocating cooling channel design allowed staged partial heat dissipation of the coolant in the cooling loop.

1. Introduction
Recent demand from the industry for better cutting efficiency and machining quality has caused more attention to be paid to the technology of high-speed cutting. In a typical machining process involved the usage of machine tools, about 75% of all geometric errors in workpiece
accuracy is caused by fluctuations of temperature.[1] This, in turn, makes thermal error one of the most important factors affecting machine precision. The continual development of high-speed cutting technology has resulted in a growing demand for high rotational speeds of the spindle. The best way to achieve this is to build the motor into the spindle assembly. If the motor is inside the spindle, transmission power loss could be reduced, and this arrangement is also particularly well suited for high-speed machine tools. However, the heat generated by the motor will significantly increase the temperature of the spindle, and thus the thermal deformation. The thermal deformation will cause changes of the relative position of the spindle components and lead to machining errors and even damage to the spindle components themselves.

During the operation of the spindle, its temperature distribution is mainly affected by the following factors [2]:

1. Heat generated by the motor: the amount of heat is related to the motor torque and rotational speed.
2. Heat generated by bearings: the amount of heat is related to the bearing load and rotational speed.
3. Thermal energy generated due to viscous dissipation of lubricants during rotation.

Simulation can be used for the thermal analysis and design of the spindle and to rapidly analyze its thermal deformation characteristics. For example, Bossmanns and Tu utilized the Finite Element Method (FEM) to construct a spindle heat transfer model [3] and verified it with actual measurement results in a study of the effect of spindle rotational speed, bearing preload, and lubrication on heat transfer of the spindle. Zahedi and Movahhedy [4] built a coupled analysis model for a high-speed spindle to study thermal deformation and the consequent dynamic interactions of the mandrel, bushing, and bearing, which was further verified by experiment. The result indicated that rotational speed affected the amount of thermal displacement, as well as the rigidity and the natural frequency of the bearing.

As previously noted, during operation, the spindle temperature increases due to heat from friction, heat generated by the motor, and the cutting operation. The effect of a rising spindle temperature and the thermal errors in consequence can be reduced by various means, such as heat insulation, cooling systems, and thermal error compensation. The most common method is the use of circulating cooling fluid in a system of cooling channels that dissipate heat from the motor and bearings. For example, Chien and Jang [5] used the numerical method to analyze and study flow and temperature of spiral cooling channel around a spindle with a variety of heat generation amount and coolant flow velocity.

Although most of the spindles available on the market have cooling channels, the vast majority of them only reduce the exterior temperature of the spindle assembly. The high temperature of the motor at the core, which usually contributes to the major thermal deformation/damage of the spindle, is much more difficult to mitigate. Moreover, the uneven temperature distribution between the outside and the inside of the spindle leads to a significant thermal stresses on the bearings which in turn reduces its service life. Furthermore, the design of the cooling channels needs careful consideration because the path, shape, and direction of cooling liquid flow are closely related to cooling efficiency as is the temperature distribution around the spindle. In this study, we conducted a three-dimensional thermo-fluidic structure multiphysics analysis for a built-in motor high-speed spindle using commercial FEM software. Results of the temperature distribution and thermal deformation of the spindle were compared and discussed with experimental measurements. Furthermore, this study also investigated the effect of the design of cooling channels on heat transfer in the spindle assembly. Suggestions to the design considerations on the built-in motor high-speed spindle cooling system are provided.

2. Method

ANSYS® Fluent Finite Element Method software was used to simulate the temperature field around the spindle in a three-dimensional model (Figure 1). Grid dependence analysis determined that a proper number of the simulation elements used in this model to be 23,288,092 elements, and the average grid quality was 0.84. The material of each component of the spindle is listed in Table 1. The assumptions used in the simulation analysis were the following:
2.1. Governing equation

The continuity equation, momentum equation, energy equation in fluid, and energy equation in solid were coupled together for calculation. The main heat sources include the motor and bearing assembly, and the respective heat generation is evaluated based on the method provided in the referenced articles as to be discussed later. It should be noted that the current study focused on the heat transfer characteristics of the spindle structure. The dynamics of the bearings and the contact resistances between the rolling balls and raceways were not included in the current analysis.

2.2. Heat source

The heat generation from the motor is estimated from Equation (1), wherein \( \eta \) is the efficiency coefficient (0.64):

\[
Q_{\text{motor}} = Q_{\text{input}} \times \eta
\]

\[
Q_{\text{input}} = \sqrt{3} \times V_{\text{in}} \times I_{\text{in}}
\]

In the bearing, there are two main heat sources: the friction heat generated due to loading and the friction heat generated by the frictional viscosity of the lubricant. In a roller bearing, the friction torque caused by the viscosity of the lubricant and the loading can be calculated from Equations (3) and (4) [6]:

\[
M_{\text{viscosity}} = \begin{cases} 
10^{-7} \times f_0 (v_0 \eta)^{2/3} d_m^3 & \text{for } v_0 \eta \geq 2000 \\
160 \times 10^{-2} f_0 d_m^3 & \text{for } v_0 \eta < 2000
\end{cases}
\]

\[
M_{\text{load}} = f_r F_r d_m
\]

\[
F_\rho = \begin{cases} 
0.9 F_a \cot \theta - 0.1 F_r & F_r \\
F_r & \text{for } F_r
\end{cases}
\]
the heat generation from each heat source was calculated as shown in Table 4. The initial temperature of the spindle was set to be 25 °C, and the natural convective heat transfer coefficients on the spindle surface were evaluated based on the empirical correlations in reference.[7]

2.3. Cooling channel

The actual fluid cooling system of this spindle design was comprised of two sets of reciprocating flow channels. The working fluid was ISO VG32 lubricant, and the inlet and outlet locations are shown in Figure 3. The boundary condition of the coolant inlet was set to the maximum flow rate of the cooling pump, 20 L/min; upon conversion, this was 2.536 m/s, perpendicular to the inlet boundary. Once the inlet velocity had been set, the mass flow rate can be calculated based on the conservation of mass. The outlet boundary condition of the cooling fluid was set to be the pressure at the outlet (gauge pressure equals zero). The inlet temperature of the coolant was set to be 25 °C, and the outlet condition was set as a convective flow outlet.

2.4. Experimental

The experimental apparatus is shown in Figure 4(a). The system used to measure the thermal deformation was developed by our team and has been validated with commercial device. K-type thermocouples were used to measure spindle temperatures, and a Yokogawa XL100 data logger was used to record temperature at 16 locations, (see Figure 4(b)) on the surface of the housing (with the spindle running at 18,000 rpm).

3. Results and discussions

Table 5 and Figure 5 show the experimental measurement results. From measurements, after the spindle has been running at 18,000 rpm for approximately 10 min, the maximum temperature of the housing reaches 33.6 °C, and the mandrel endpoint elongates by approximately 10 μm. The simulation result is shown in Table 5 and Figure 6. The temperatures at A, B, C, and D correspond to the average values at the locations shown in Figure 4(b). The experiment and simulation both show that the maximum temperature of the spindle is near the motor. The simulation results shown in Figure 6 indicate that although the cooling channel effectively cools the spindle in general, the high temperature at the center is not dissipated and this causes a very uneven distribution of temperature between the exterior and the interior of the spindle. Figure 7 is the temperature distribution within the cooling fluid.

| Measurement point/T | Measurement (°C) | Simulation (°C) |
|---------------------|------------------|-----------------|
| A                   | 28.6             | 35.6            |
| B                   | 28.1             | 33.3            |
| C                   | 29.9             | 30.8            |
| D                   | 33.6             | 38.9            |
| Z-axis elongation   | 11.2 μm          | 19.2 μm         |

Figure 3. Inlet and outlet locations of cooling fluid channels.

Figure 4. (a) Testing setup; (b) Temperature measurement points.

Table 5. Spindle temperatures and elongations.
is transferred from the oil to the spindle. In consequence, when the coolant reaches the front bearing area, it will have a greater cooling effect. If the coolant temperature does not drop during flow, its temperature may even be higher than the bearing temperature and it will have no cooling effect. After the coolant has left the front bearing area, it flows through a section of the spindle structure where the temperature is lower and some heat is again dissipated before it enters the motor area to effect more cooling. Furthermore, the results of both the experimental measurements and simulation analysis show that spindle temperature distribution is close to being axisymmetric. Therefore, the primary thermal displacement of the cutter endpoint is in the Z direction, and there is no obvious radial displacement, which is preferable in terms of thermal error control.

Although there are obvious differences between the simulation values and the experiment measurements, they do show similar trends and the order of magnitude of the simulation data is also reasonable. Consequently, the simulation analysis model can be used to investigate the major physics toward temperature increase and thermal deformation in this type of spindle.

The reciprocating design described here seems to be an efficient means for cooling of built-in motor spindle. The method facilitates cooling of the coolant in stages by utilizing parts of the spindle assembly that have a lower temperature to enhance the overall cooling effect and at the same time achieve more even overall distribution of the spindle temperature. To further verify these effects, changes were made to the simulated design to test the helical cooling channels on spindle temperature distribution. The helical cooling channel is commonly seen in the design of spindle cooling loops.[8] Figure 8 shows a comparison of the original design (a) with single helix (b) and a double helix (c) cooling channels. To make a fair comparison, the flow rate, channel diameter, and the length of these cooling channels were kept to be approximately the same.

A comparison of simulated results of the three types of spindle cooling channel design showed the heat dissipation performance arranged in order of heat exchange efficiency to be excellent for the reciprocating type, the single helix came next and the double helix came last. In terms of temperature uniformity, the original reciprocating channel design also yielded the best result. The double helix had a more uniform spindle temperature distribution than that of the single helix design. This was particularly evident in the area close to the motor. The reciprocating cooling channel design reduced the spindle radial temperature gradient most efficiently. This is very important for avoiding bending deformation which is directly related to temperature gradient and the thermal loading of the bearings. The simulation results clearly show that the staged

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**Figure 5.** Experimental: Spindle temperature and elongation.

**Figure 6.** Simulation: Spindle temperature distribution (a) bushing surface; (b) internal cross section.

**Figure 7.** Simulation: Cooling fluid channel temperature distribution.
gradient between the interior and exterior of the spindle. Therefore, shaft-bore cooling for the core is recommended for purpose. In addition, the fluid-solid-heat coupling simulation model used in this study was used to effectively analyze the temperature increase and the thermal deformation characteristic of a spindle with a built-in motor and can be used to predict the effect of changes in cooling design on spindle thermal characteristics.

Nomenclature

- $\nu_0$: Lubricant viscosity (kg/m•s)
- $M$: Friction torque (N•mm)
- $m$: Mass flow rate (kg/s)
- $N$: Spindle rotation (rpm)
- $d_m$: Bearing sectional diameter (mm)
- $\theta$: Roller bearing contact angle (degree)
- $F_a$: Axial load on bearing (N)
- $F_r$: Radial load on bearing (N)
- $T$: Temperature (°C)
- $\rho$: Density (kg/m³)
- $P$: Pressure (Pa)
- $\mathbf{F}$: Force (N)
- $Q_{\text{motor}}$: Heat generation from motor (W)
- $Q_{\text{input}}$: Input power of motor (W)
- $Q_{\text{bearing}}$: Heat generation from bearing (W)
- $V_{\text{in}}$: Input voltage of motor (V)
- $I_{\text{in}}$: Input current of motor (A)

Disclosure statement

No potential conflict of interest was reported by the authors.

Funding

This work was supported by the Ministry of Science and Technology of Taiwan [grant number MOST 104-2218-E-005-003].

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