Thermoelectric-based cooling system for high-speed motorized spindle II: Optimization and validation strategy

Kaiguo Fan · Rongfei Xu · Ruoda Wang · Rui Gao

Received: 12 September 2021 / Accepted: 7 January 2022 / Published online: 21 January 2022
© The Author(s), under exclusive licence to Springer-Verlag London Ltd., part of Springer Nature 2022

Abstract
With the development of motorized spindle, the cooling effect and the distribution of cooling capacity become the crucial problem of cooling system. An optimization method for thermoelectric-based cooling system (TECS) is proposed based on the conservation of energy to distribute the cooling capacity. The main strategy of the proposed optimization method is to make the cold and heat input at different regions of the spindle sleeve equal in real-time through optimizing the contact area between the heat conduction sleeve (HCS) and spindle sleeve. The numerical simulation and thermal characteristics experiments are carried to verify the effect of the proposed optimization method and the TECS. The simulation and experimental results show that the maximum temperature rise and thermal elongation of the TECS-based motorized spindle are reduced 56.7% and 58.6% compared with water-cooled motorized spindle, and the temperature distribution of the spindle sleeve is more uniform. It is meaningful to improve the accuracy of motorized spindle.

Keywords Optimization strategy · Heat transfer path · Simulation and experiment · Cooling system · High-speed motorized spindle

1 Introduction

The heats generated by the internal heat sources transfer to the environment and the cooling system through different components and contact surfaces, which forms a heat transfer path. The purpose of thermal optimization is to optimize the heat transfer path to reduce the temperature rise and thermal deformation. The internal heat sources, thermal conductivity, thermal contact resistance, convective heat transfer, and the structure of these components are the main factors that affect the heat transfer path.

In order to optimize the heat transfer path, the first step is to obtain the thermal behavior of a machine tool which is usually realized through numerical simulation. Before simulation, the thermal boundary conditions, which include the heat generation of internal heat sources, the thermal contact resistance, and the convection coefficient, should be determined firstly. The internal heat sources of a machine tool mainly include the heat converted from the motor energy consumption which can be calculated through the measured motor torque [1, 2], empirical formulas, etc., the heat generated by friction of rotation and transmission components which can be determined by calculating the friction torque using the empirical formula [3, 4], etc., and the heat generated by cutting which can be calculated through reverse solving of differential equation of unsteady heat conduction according to the actually measured temperature of the cutting tool [5]. The commonly used identification methods of thermal contact resistance mainly include the theoretical calculation method [6–8] and the experimental method [9–11] to obtain the thermal contact resistance under a specific working condition. For the calculation of convection coefficient, Chow et al. [12] proposed a prediction model using the method of finite difference and experiment. Barrios et al. [13] used the finite volume method and Gaussian method to establish a three-dimensional transient model for convection coefficient which can realize the calculation of convection coefficient under cutting conditions.

After determining the thermal boundary conditions, the finite element numerical simulation is usually used to analyze the thermal behavior of the machine tool. References [14, 15] used the finite element numerical simulation method to analyze the temperature field and thermal deformation of

Kaiguo Fan
kg11@163.com

1 School of Mechanical Engineering, University of Shanghai for Science and Technology, Shanghai 200093, China
machine tool’s spindle and ball screw, which can reveal the heat transfer path of a machine tool and can guide the optimization design of the machine tool. However, the simulation accuracy depends on the calculation accuracy of boundary conditions. There are many parameters that should be calculated accurately to obtain the most realistic boundary conditions and simulation results. Some parameters, such as the spindle speed, rated power, surface roughness, hardness, thermal conductivity, elastic modulus, Poisson’s ratio, and the dimension of part, can be identified accurately according to the material properties and the machine tool’s parameters. However, some parameters, such as the friction torque of bearing and contact pressure between the contact surfaces are very difficult to calculate accurately due to the influence of manufacturing and assembly accuracy of machine tool’s parts. Furthermore, the friction torque of bearing and contact pressure between contact surfaces will change with thermal deformation of machine tool’s components. In order to improve the calculation accuracy of thermal boundary conditions, Kang et al. [16] established a dynamic heat generation model for bearing considering the thermally induced preload and the viscosity–temperature effects, which greatly improves the calculation accuracy of heat generation of bearing. Fan [17] used the temperature rise of thermal key points to real-time correct the heat generation of internal heat sources which can effectively improve the simulation accuracy. However, since the thermal boundary conditions of a machine tool are very complicated, it is difficult to obtain its actual value.

Using the simulation results, the heat transfer path of the machine tool can be optimized. For the structural optimization, the main methods include the construction theory, topology optimization, and bionic optimization to optimize the material distribution within a specific load, constraint, area, and performance [18–20]. For the optimization of convective heat transfer, the commonly used method is to increase the heat dissipation of the cooling system and balance the temperature field through controlling the cooling parameters and optimizing the cooling channel [21–24]. For the optimization of thermal conductivity, the material with high thermal conductivity becomes the best choice, through which the heats generated by the internal heat sources can be transferred to other areas quickly to balance the temperature field [25]. The thermal contact resistance between the contact surfaces is affected by factors such as the intermediate medium, material properties, roughness of contact surfaces, temperature, and pressure [6, 7, 9, 10]. The current research mainly focuses on the analysis of above factors and attempts to optimize these factors. In summary, the above studies provide a basis for the optimization of heat transfer path.

After optimization, the numerical simulation and thermal characteristics experiment are usually used to verify the optimization effect. The advantage of numerical simulation is that it can solve complex nonlinear problems and simulate the thermal characteristics of whole machine. But the simulation accuracy depends on the calculation accuracy of thermal boundary conditions, finite element model, etc. In order to obtain the more realistic thermal behavior of a machine tool, the thermal characteristics experiment has been the best choice. The thermal characteristics experiment of a machine tool includes the spindle idling experiment under different spindle speed and running time, the environmental temperature variation error (ETVE) test, the test for thermal distortion caused by moving linear axes, the test for thermal distortion caused by rotary motion of components [26], etc. The commonly used test devices include the temperature and displacement sensors to measure the temperatures of thermal key points and the thermal deformation of key components of the machine tool. With the development of detection technology, thermal imaging and laser techniques have been applied to the temperature and displacement sensors to achieve non-contact measurement. Wu et al. [27] used the thermal infrared imager and laser triangulation sensors to measure the temperature field and thermal deformation of a motorized spindle, the significant effect of the high rotational speeds, preload oil viscosity, and heat transfer coefficients on the temperature or thermal failure of the bearing has been revealed. However, the thermal characteristics test method is limited by the number of measuring sensors and the installation location, and cannot fully reflect the thermal behavior especially the internal thermal behavior of a machine tool.

In summary, the thermal behavior of a machine tool is like a “black box,” and there are many factors that can only be obtained through experience and theory, which limit the accuracy of thermal optimization and thermal behavior analysis. In order to solve this problem, an online correction method for thermal boundary conditions is proposed in Part 1 which can improve the simulation accuracy significantly. In this study, the corrected thermal boundary conditions are used to simulate the thermal behavior of a motorized spindle, and the simulation results are used to optimize the heat transfer path. The optimization strategy is introduced in Sect. 2. The optimization effect of the thermoelectric-based cooling system (TECS) is verified through numerical simulation and thermal characteristics experiment in Sects. 3 and 4. Finally, some discussions about the cooling effect and cooling system are given in Sect. 5.

2 Optimization strategy of TECS

The optimization strategy of the TECS is to make the cold and heat input at different regions of the spindle sleeve equal in real-time through optimizing the contact area
between the heat conduction sleeve (HCS) and spindle sleeve as shown in Fig. 1. The detailed optimization strategy is as follows:

1. Extracting the temperature rise of the spindle sleeve along the axial direction as shown in Fig. 2.
2. Fitting the temperature rise of the spindle sleeve as a linear function of axial position.
3. Modifying the linear function established in previous step using the following equations:
   \[
   c_{SP}\rho_{SP}2\pi rt\left(\Delta l \sum_{i=1}^{n} \Delta T_i - \int_{0}^{1} f(x)dx\right) = 0 \tag{1}
   \]
   \[
   f(x) = p_1x + p_2 \tag{2}
   \]
   where \(c_{SP}\) and \(\rho_{SP}\) are the specific heating capacity in J/(kg·℃) and density in kg/m³ of the spindle sleeve, respectively, \(r\) and \(t\) are the average radius and wall thickness of the spindle sleeve in meter, respectively, \(x\) is the axial position of the spindle sleeve in meter, \(\Delta l\) and \(\Delta T_i\) are the length in meter and temperature rise of each section of the spindle sleeve, respectively, \(n\) is the number of the extracted temperature rises.

Substituting Eq. (2) into Eq. (1), the following equation can be obtained:

\[
\Delta l \sum_{i=1}^{n} \Delta T_i - \frac{1}{2}p_1l^2 - p_2l = 0 \tag{3}
\]
where \(l\) is the length of the spindle sleeve in meter, \(p_2\) is the coefficient to be revised. In Eq. (3), \(\Delta l\), \(\Delta T_i\), and \(l\) are all known, \(p_1\) can be obtained through the linear fitting of temperature rise established in step 2). In order to make Eq. (1) true, we only need to calculate the coefficient of \(p_2\) through Eq. (3). Then, a new linear function of temperature rise can be established as shown in Fig. 2.

4. Calculating the intersections of the temperature rise curve and the new fitting line, and marking the intersections as A, B, C, and D as shown in Fig. 2.

5. Calculating the lengths of \(l_1\) through \(l_5\) according to the calculated intersections of A, B, C, and D.

Using the above-mentioned method, the length of each section of the spindle sleeve can be calculated as \(l_1 = 0.013\) m, \(l_2 = 0.033\) m, \(l_3 = 0.089\) m, \(l_4 = 0.032\) m, and \(l_5 = 0.013\) m. Figure 2 shows that the temperature rises at
regions $l_1$, $l_3$, and $l_5$ are higher than that at regions $l_2$ and $l_4$. On the one hand, because the heats at regions $l_2$ and $l_4$ are transferred from regions $l_1$, $l_3$, and $l_5$ and then transferred to the HCS, reducing the heat transfer from regions $l_2$ and $l_4$ to the HCS can increase the temperature rise at regions $l_2$ and $l_4$. On the other hand, because the cooling capacity generated by the TEC are constant at a certain moment, reducing the input of cooling capacity at regions $l_2$ and $l_4$ can increase the input of cooling capacity at regions $l_1$, $l_3$, and $l_5$. Therefore, reducing the contact area between the HCS and spindle sleeve at regions $l_2$ and $l_4$ can increase the temperature rise at regions $l_2$ and $l_4$ and decrease the temperature rise at regions $l_1$, $l_3$, and $l_5$. The removed widths are equal to $l_2$ and $l_4$ as shown in Fig. 3. Thereby, the structure of the HCS is optimized through the above-mentioned optimization method.

In summary, the design and optimization strategy of the proposed TECS includes five steps. The first step is calculating the parameters of HCS to achieve the purpose of rapid heat conduction using the constrained optimization method. The second step is determining the required maximum cooling power and number of thermoelectric cooling chips used in thermoelectric cooler (TEC) to cool the heat transferred to the HCS completely according to the maximum heats generated by the internal heat sources and the structure of the motorized spindle. The third step is designing the cooling sleeve to cool the hot side of the TEC according to the maximum heat generation of the motorized spindle and the power consumption of the TEC. The fourth step is designing the TEC controller to control the cooling capacity of the TEC based on the proportion integral differential (PID) algorithm. The fifth step is optimizing the structure of HCS to distribute the cooling capacity accurately according the temperature distribution of the spindle sleeve.

Using the above-mentioned five steps to design and optimize the TEC and HCS, a TECS-based motorized spindle can be designed or re-equipped. Figure 4 shows the structure of the motorized spindle before and after re-equipped. In order to compare the cooling effect of TECS with water-cooling system, the numerical simulation of the motorized spindle before and after re-equipped are carried out in Sect. 3.

3 Simulation validation

3.1 Thermal behavior simulation

The simulation conditions are that the spindle speed is 6000 rpm, the temperatures of ambient and cooling water are all 20 °C, the temperature of the cold side of TEC is set as 20 °C, that is, the temperature of HCS is 20 °C due to the cold side of TEC is HCS. The structures of the stator and bearings are simplified as cylinders, and some small structures in the finite element model, such as bolts, bolt holes, and chamfers are removed. All the corrected boundary conditions (as shown in Table 1) and structures of the motorized spindle before and after re-equipped are the same except for the cooling system in numerical simulations. Applying the corrected boundary conditions to the finite element model, the temperature field of the motorized spindle before and after re-equipped can be obtained as shown in Fig. 5.

Figure 5a shows the temperature field of the motorized spindle with water-cooling system. The maximum temperature rise of the motorized spindle is 32.302 °C which occurs in the rotor. The minimum temperature rise is 5.513 °C which occurs in the cooling water jacket. That is, the water-cooling system cannot dissipate the heat transferred to the cooling channel completely when the water temperature is the room temperature. The temperature distribution of the spindle sleeve is ununiform, the temperature rises at the regions related to the internal heat sources are higher than that at other regions, and the maximum temperature difference of the spindle sleeve is 3.105 °C.
Fig. 4 Structure of the motorized spindle before and after re-equipped shows that the water jacket of the water-cooled motorized spindle is replaced by the TECS.

Fig. 5 Simulation result shows the temperature field of the motorized spindle before and after re-equipped. The temperature field of the spindle sleeve with TECS is more uniform than that with water-cooling system.
Figure 5b shows the temperature field of the motorized spindle with TECS. The maximum temperature rise of the motorized spindle is 15.611 °C which occurs in the rotor. The temperature distribution of the sleeve is more uniform, and the maximum temperature difference is 0.547 °C. The minimum temperature rise is 0 °C which occurs in the HCS. That is, the TECS can cool the heat transferred to the HCS completely.

Comparing Fig. 5a with b, the temperature distribution of the sleeve with TECS is more uniform than that with water-cooling system, the maximum temperature difference of the sleeve is reduced from 3.105 °C to 0.547 °C. That is, the optimization design of HCS can distribute the cooling capacity effectively. The maximum temperature rises of the TECS-based motorized spindle is reduced 16.691 °C compared with water-cooled motorized spindle. That is, the cooling effect of the TECS is significantly better than that of water-cooling system when the water temperature is the room temperature. The simulation results also show that the temperature distribution of the motorized spindle before and after re-equipped is basically the same except for the sleeve and cooling system. That is, changing the structure of the motorized spindle can improve the temperature field distribution, which can be used to guide the thermal optimization design of the motorized spindle.

### 3.2 Stiffness simulation

Because the water-cooling system is replaced by the TECS, and the structure of the two is different, the modal analysis is carried out to verify the stiffness of the motorized spindle before and after re-equipped. The simulation conditions are that the front end of the motorized spindle is fixed, while the rear end only restricts the radial displacement; all contact surfaces are bounded, the density, elastic modulus, and Poisson’s ratio of steel and AL 6063 are 7800 and 2700 kg/m³, 211 and 69 Gpa, 0.3 and 0.33, respectively; a force of 100 N is applied to the front end of the spindle. The number of modes to be extracted is set as 6.

Figure 6 shows the first-, third-, and fifth-order vibration modes. Table 2 shows the first six orders frequencies of the motorized spindle before and after re-equipped. The simulation results show that the stiffness of TECS-based motorized spindle has little change compared with water-cooled motorized spindle; the vibration modes before and after re-equipped are almost the same; the first- and sixth-order frequencies are slightly lower than water-cooled motorized spindle, but the other order frequencies are higher than water-cooled motorized spindle. In theory, the first-order frequency of the TECS-based motorized spindle decreases, the other orders frequencies also should decrease. However, the simulation results are inconsistent with the theory, and the first- and sixth-order frequencies decrease but the other order frequencies increase. This is because the main structure of the motorized spindle before and after re-equipped is the same except for the cooling system, and the constraint position and mode are the same too; the stiffness of the motorized spindle changes little before and after re-equipped.

On the other hand, the finite element method is based on the discretization which will lead to deviations from the actual continuum. In addition, the boundary conditions of the actual continuum are very complicated, and the finite element method cannot fully simulate these boundaries.
Therefore, there is a certain deviation between the simulated and actual values. In order to verify the stiffness of the motorized spindle before and after re-equipped, the vibration test under cutting conditions is carried out in Sect. 4.2.

4 Experimental validation

4.1 Thermal behavior validation

In order to verify and compare the actual effect of the TECS-based motorized spindle with water-cooled motorized spindle, the thermal characteristics experiments of the motorized spindle before and after re-equipped are carried out, respectively, and the temperature rise of the front bearing cap and the thermal elongation of the spindle are used to characterize the thermal characteristics of the motorized spindle.

Figure 7a shows the experimental setup of water-cooled motorized spindle. The motorized spindle is fixed in a vise, and the frequency converter is used to control the spindle speed. A ZH3010 eddy current displacement sensor with a resolution of 0.1 μm and range of 4 mm is used to measure the thermal elongation of the spindle. The temperature sensors T1, T2, and T3 are used to measure the temperature rise of the spindle related to the internal heat sources which is the front bearings, stator, and rear bearings, respectively. The experiment is divided into three groups according to the spindle speed which is 3000, 6000, and 10,000 rpm, respectively. A pump is used to circulate the cooling water in the coolant channel, the temperature of the cooling water is 22°C, and the flow rate is 0.3 m/s. Each experiment starts from the cold state, and the ambient temperature of the laboratory is 22°C.

Figure 7b shows the measured and simulated temperature rises of the spindle housing at the measurement position T1 when the spindle speed is 3000, 6000, and 10,000 rpm, respectively. Figure 7b shows that the thermal equilibrium time of the spindle housing is about 1500 s, and the temperature rise trends of the front bearing cap under different spindle speed are the same. The maximum temperature rise of the front bearing cap is 13.2°C which occurs when the spindle speed is 10000 rpm. Figure 7b also shows that the simulation results are consistent with the experimental results, and the simulation accuracy is great than 95% using the online correction method to correct the thermal boundary conditions. This proves that the proposed online correction strategy for thermal boundary conditions can effectively improve the simulation accuracy of thermal characteristics of machine tool.

Figure 8a shows the thermal elongation of the spindle, the maximum thermal elongation of the spindle is 44.85 μm when the spindle speed is 10,000 rpm. The thermal elongation trend of the spindle is consistent with the temperature rise trend of the front bearing cap. Table 3 shows the maximum thermal elongation of the spindle before and after re-equipped when the spindle speed is 3000, 6000, and 10,000 rpm, respectively.

Figure 9a shows the experimental setup of the TECS-based motorized spindle. The water-cooling system is replaced by the TECS. The motorized spindle is fixed in a vise. The TEC controller designed in Part 1 is used to control the temperature of the HCS, and the preset temperature of the HCS is ambient temperature. The temperature and displacement sensors used in this experiment are the same with the experiment of water-cooled motorized spindle.
spindle, but the measurement positions of the temperature sensors are different. The temperature sensors T1 and T2 are used to measure the temperature rises of front and rear bearing caps, and the temperature sensor T3 is used to measure the temperature rise of the HCS in this experiment. Temperature sensor T_HCS is used to control the cooling power through the TEC controller. The experiment is divided into three groups according to the spindle speed which is 3000, 6000, and 10,000 rpm, respectively.

Figure 9b shows the temperature rises measured by the temperature sensor T1 when the spindle speed is 3000, 6000, and 10,000 rpm, respectively. The maximum temperature rise of the front bearing cap is 5.7 °C which is reduced 56.7% compared with water-cooled motorized spindle. Figure 9b also shows that the simulation results are consistent with the experimental results, and the simulation accuracy is greater than 95% using the online correction method to correct the thermal boundary conditions. The maximum thermal elongation is 18.57 μm when the spindle speed is 10,000 rpm as shown in Fig. 8b, which is reduced 58.6% compared with water-cooled motorized spindle.

Comparing the thermal characteristics of TECS-based motorized spindle with water-cooled motorized spindle as shown in Figs. (7b, 8, and 9b), the trends of temperature rise and thermal elongation of the motorized spindle before

---

**Fig. 8** Experimental result shows the thermal elongation of the motorized spindle before and after re-equipped. The thermal elongation of the TECS-based motorized spindle is significantly improved

---

**Fig. 9** Actual experimental setup shows the TECS-based motorized spindle clamped in a vise. Three temperature sensors and an eddy current are used to measure the temperature and thermal elongation of the spindle to compare the cooling effect with water-cooling system
and after re-equipped are almost the same, this is because
the main structure of the motorized spindle is the same, and
the heat transfer in the motorized spindle is the same too
except for the cooling system. But the temperature rises and
thermal elongation of the TECS-based motorized spindle is
significantly reduced compared with water-cooled motorized
spindle, which proves that the cooling effect of the TECS is
significantly better than that of water-cooling system. It is
meaningful to improve the accuracy of motorized spindle.

In order to further verify the control effect of the TECS,
another two experiments are carried out in this study. The
first experiment is to verify the control effect of the TECS
under different conditions, the ISO 230–3 standard is used
to test the thermal elongation of spindle and the tempera-
ture of HCS as shown in Fig. 10. The experimental result
shows that the temperature of the HCS is controlled within
0.2 °C, which verifies the control effect of the TECS. The
thermal elongation of the spindle increases continually, but
the increasing trend is different from that of single spin-
dle speed due to the change of spindle speed during its
running.

The second experiment is to verify the influence of the
control temperature of HCS on the thermal elongation of
the spindle. The control temperature of the HCS is set to 5, 10,
15, 20 °C lower than the ambient temperature, respectively,
and the spindle speed is 6000 rpm. The experimental results
show that the thermal elongation of the spindle decreases
with the decrease of the control temperature of HCS as
shown in Fig. 11. When the temperature of the HCS is 15
°C lower than the ambient temperature, the thermal deforma-
tion of the spindle is close to zero. That is to say, con-
trol the temperature of HCS can cool the motorized spindle
deeply, which is of significance to improve the accuracy of
motorized spindle. However, when the controlled tempera-
ture of the HCS is much lower than the ambient tempera-
ture, the condensed water will be generated on the surface
of motorized spindle, the appropriate temperature without
condensed water generated needs further study.

According to the experimental results, the cooling
effect of the proposed TECS is obviously better than that
of water-cooling system, and the proposed TECS can
realize the deep cooling of motorized spindle to achieve
the goal of zero thermal elongation of motorized spindle,
which is of great significance to improve the performance
and machining accuracy of motorized spindle.

Fig. 10 Control effect validation of TECS using the ISO 230–3 standard

Fig. 11 Thermal elongation of the spindle decreases with the
decrease of the control temperature of HCS. Δt is the temperature dif-
ference between the HCS and ambient
In order to verify the influence of TECS on the stiffness of the motorized spindle, a vibration test is carried out under the cutting condition. Figure 12 shows the experimental setup of the stiffness validation of the motorized spindle before and after re-equipped. The motorized spindle is fixed on a small-sized CNC machine tool through a flange used to fix the front end and a fixture used to restrict the radial displacement of the rear end. Because acceleration is generally used for vibration evaluation of high-speed rotating machinery, an acceleration sensor is installed on the rear end of the motorized spindle to measure its vibration acceleration in this experiment. The experimental conditions are that the spindle speed is 6000 rpm, the feed rate of the cutting tool is 0.2 mm/r, the workpiece material is aluminum alloy. The measured maximum vibration accelerations of the water-cooled motorized spindle, and the TECS-based motorized spindle are 3.89 g and 3.91 g, respectively. The experimental results show that the stiffness of TECS-based motorized

![Image](image_url)

**Fig. 12** Actual experimental setup shows the motorized spindle installed in a small-sized CNC machine tool to verify the stiffness. An acceleration sensor is used to measure the vibration acceleration of the spindle.

### 4.2 Stiffness validation

In order to verify the influence of TECS on the stiffness of the motorized spindle, a vibration test is carried out under the cutting condition. Figure 12 shows the experimental setup of the stiffness validation of the motorized spindle before and after re-equipped. The motorized spindle is fixed on a small-sized CNC machine tool through a flange used to fix the front end and a fixture used to restrict the radial displacement of the rear end. Because acceleration is generally used for vibration evaluation of high-speed rotating machinery, an acceleration sensor is installed on the rear end of the motorized spindle to measure its vibration acceleration in this experiment. The experimental conditions are that the spindle speed is 6000 rpm, the feed rate of the cutting tool is 0.2 mm/r, the workpiece material is aluminum alloy. The measured maximum vibration accelerations of the water-cooled motorized spindle, and the TECS-based motorized spindle are 3.89 g and 3.91 g, respectively. The experimental results show that the stiffness of TECS-based motorized

### Table 1 Calculation of initial boundary conditions and its corrected results.

| Boundary conditions          | Assemblies/Medium      | Initial conditions | Calculation equations          | Calculated results | Corrected results |
|-----------------------------|------------------------|--------------------|--------------------------------|--------------------|-------------------|
| Heat generation             | Front bearings         | n = 6000 rpm       | Q₀ = 1.047 \times 10^{-4}nM   | 8.5 W              | 8.3 W             |
|                             | Rear bearings          |                    |                               | 6.8 W              | 6.5 W             |
|                             | Stator                 | P₀ = 0.8 kW        | Qₘ = (1 − \eta)P₀             | 32 W               | 30.8 W            |
|                             | Rotor                  | P = 6%             |                               | 16 W               | 15.4 W            |
|                             |                        | \eta = 0           |                               |                    |                   |
|                             |                        | Qₛ = 2 × Qₘ/₃      |                               |                    |                   |
|                             |                        | Qᵣ = Qₘ/₃         |                               |                    |                   |
| Thermal contact resistance  | Spindle to bearing     | \sigma₁ = 0.8 \mu m | Rᵣ = \frac{\sigma₁}{10.64} \times \left( \frac{H}{T} \right)^{0.618} | 8.6 × 10⁻⁴ m²·K/W  | 9.8 × 10⁻⁴ m²·K/W |
|                             | Bearing to bearing sleeve | \sigma₂ = 0.8 \mu m |                               | 9.5 × 10⁻⁴ m²·K/W  | 10.1 × 10⁻⁴ m²·K/W |
|                             | Bearing sleeve to spindle sleeve | \epsilon = 0.02 - 0.06 mm |                               | 9.8 × 10⁻⁴ m²·K/W  | 10.5 × 10⁻⁴ m²·K/W |
|                             | Spindle sleeve to HCS  |                    |                               | 5.1 × 10⁻² m²·K/W  | 8.1 × 10⁻² m²·K/W  |
|                             | Stator to housing      |                    |                               | 9.8 × 10⁻² m²·K/W  | 10.5 × 10⁻² m²·K/W |
|                             | Rotor to stator        |                    |                               | 1 × 10⁻² m²·K/W    | 1 × 10⁻² m²·K/W    |
| Convection coefficient      | Water                  | v_water = 0.3 m/s  | Compound heat transfer \ h = \frac{Nu \cdot \alpha_{water}}{L} | 251.6 W/(m²·K)    | 235.8 W/(m²·K)    |
|                             | Air                    |                    |                               | 9.7 W/(m²·K)       | 8.9 W/(m²·K)      |
| Temperature                 | TEC                    | 20 °C              | Preset                         | 20 °C              | 20 °C             |
|                             | Water                  |                    |                               | 20 °C              | 20 °C             |
|                             | Ambient                |                    |                               | 20 °C              | 20 °C             |

© Springer
Cooling mode

Sect. 4.1. That is, the temperature of HCS can be controlled by controlling the electric current passing through the TEC as shown in the cold side of the TEC can be controlled by controlling according to the experiment results, the temperature of spindle has little change compared with water-cooled motorized spindle, which is consistent with the simulation results as shown in Sect. 3.2.

5 Discussions

5.1 Discussion about the cooling effect

According to the experiment results, the temperature of the cold side of the TEC can be controlled by controlling the electric current passing through the TEC as shown in Sect. 4.1. That is, the temperature of HCS can be controlled real-timely due to the cold side of TEC is HCS. However, it is difficult to control the temperature of cooling channel in water-cooled motorized spindle. According to the heat transfer theory, the heat transferred to the HCS of the TECS-based motorized spindle and the cooling channel of water-cooled motorized spindle, and the cooling water keeps room temperature, then,

\[ t_{\text{int-T}} - t_{\text{HCS}} = t_{\text{int-w}} - t_{\text{channel}}, t_{\text{water}} = t_{\text{HCS}} = t_a \] (6)

According to the heat transfer theory, the temperature of the cooling channel must be higher than water temperature to achieve the heat transfer in water-cooling system. That is, \( t_{\text{channel}} > t_{\text{HCS}} \), thereby the temperature of the internal heat sources in water-cooled motorized spindle is higher than that in TCES-based motorized spindle, that is \( t_{\text{int-w}} > t_{\text{int-T}} \). It means that the cooling effect of TECS is better than that of water-cooling system.

If the heat transferred to the HCS is greater than that transferred to the cooling channel, that is,

\[ t_{\text{int-T}} - t_{\text{HCS}} > t_{\text{int-w}} - t_{\text{channel}}, t_{\text{channel}} > t_{\text{water}} \] (7)

Because the heat generation is consistent under the same condition, there are more heats are converted into the internal energy in water-cooled motorized spindle and its temperature is higher than that in TCES-based motorized spindle. It also means that the cooling effect of TECS is better than that of water-cooling system.

If the heat transferred to the HCS is less than that transferred to the cooling channel, that is,

\[ t_{\text{int-T}} - t_{\text{HCS}} < t_{\text{int-w}} - t_{\text{channel}}, t_{\text{channel}} > t_{\text{water}} \] (8)

Because the temperature of cooling channel is greater than that of HCS, the temperature of the internal heat sources in water-cooled motorized spindle is also higher than that in TCES-based motorized spindle. Therefore, the cooling effect of the TECS is better than water-cooling system when the temperature of the cooling water keeps ambient temperature.

According to the theoretical discussion mentioned above, the cooling effect of the proposed TECS is better than water-cooling system when the cooling water keeps ambient temperature which is consistent with the simulation and experiment results.

5.2 Discussion about the cooling system

Generally, a cooling system of a motorized spindle includes the coolant channel, cooling medium, and drive device for...
cooling medium. The commonly used coolant channel is rectangular helical channel which is usually installed in the inside of the spindle housing to transfer the heat generated by internal heat sources to the cooling medium. The commonly used cooling medium includes the water, compressed air, and cooling oil, which flows in the coolant channel and dissipates the heat to the environment. The heat dissipation capacity of a cooling system depends on both the coolant channel and the cooling medium.

On the one hand, the structure, material, and interface thermal resistance of coolant channel affect the cooling effect of cooling system. As the heat generation of each internal heat source is different, the temperature field of the cooling sleeve is ununiform. It is difficult to design a suitable coolant channel that does not affect the flow rate and can balance the temperature field. Moreover, the interface thermal resistance will increase with the deposition of dirt. However, the form of HCS is solid, the cooling effect of HCS is not affected by the interface thermal resistance and it is easier to optimize the structure.

On the other hand, the cooling effect of a cooling system is not only related to the structure of coolant channel, but also related to the heat absorption capacity and thermal diffusivity of cooling medium, which depends on the thermal conductivity, specific heating capacity, velocity, and flow of cooling medium in traditional cooling system. Due to the influence of the structure of motorized spindle, the velocity and flow of cooling medium are all limited which affect the cooling effect of cooling system. Therefore, accurate modeling and control for heat absorption of traditional cooling system requires in-depth research. However, the thermal diffusivity of HCS depends on the material’ properties as shown in Eq. (9).

\[ a = \frac{k}{\rho \cdot c} \]  

(9)

In Eq. (9), \( k \), \( \rho \), and \( c \) are the thermal conductivity, density, and specific heating capacity of the HCS, respectively. Currently, materials with high thermal diffusivity include the diamond, carbon fiber, copper alloys, aluminum alloys, etc. Considering the factors of processability and price, the most suitable materials for HCS are copper alloys and aluminum alloys. Therefore, the HCS does not need to control the thermal diffusivity but only needs to control the cooling capacity through the TEC controller.

With the continuous development of thermoelectric technology, the cooling power of a single thermoelectric cooling chip can reach several hundred watts and the size is very small, which is possible to design a TEC with high cooling power to cool high-power motorized spindle. Therefore, the TECS is better than traditional cooling system not only in terms of structural optimization but also in terms of cooling effect.

\section{6 Conclusions}

An optimization method for thermoelectric-based cooling system (TECS) has been proposed based on the conservation of energy. The cooling effect of TECS is verified and compared with water-cooling system through numerical simulation and thermal characteristics experiment. Based on the numerical simulation and experimental results, the following conclusions can be drawn.

1. The distribution of cooling capacity can be achieved by optimizing the contact area and position between the cooling channel and the spindle sleeve. The initial temperature field of spindle sleeve can be used to optimize the structure of cooling channel through calculating the over-cooled regions of the spindle sleeve.

2. Under the same conditions, the lower the temperature of the cooling channel, the more heat transfers from the internal heat sources to the cooling system, the better the cooling effect of the motorized spindle obtains, and there has a suitable temperature that makes the thermal elongation of spindle zero.

3. The stiffness of the motorized spindle is related to the structure and constraint mode and position, changing the restraint mode and position can greatly improve the stiffness of the system.

\section{Author contributions} The corresponding author contributed the central idea and wrote the initial draft of the paper. The remaining authors contributed to carrying out additional analyses and experiments and finalizing this paper.

\section{Funding} This paper is sponsored by the “Technology of on-line monitoring system for thermal characteristics of NC machine tools” (No. H2019304021); the “Project funded of Shanghai science committee-Precision technology and its application for five-axis machine tool based on the real-time compensation” (NO. J16022).

\section{Availability of data and material} The data that support the findings of this study are available from the corresponding author upon reasonable request.

\section{Code availability} Not applicable.

\section{Declarations}

\section{Ethics approval} The authors declare that the work is original and is not published elsewhere in any form or language.

\section{Consent to participate} The authors declare that they agree to participate.

\section{Consent for publication} The authors declare that they agree to publish.
Conflicts of interest/Competing interests The authors declare that they have no conflict of interest.

References

1. Brecher C, Witt S (2004) Static dynamic and thermal behavior of machine tools with regard to HPC. CIRP High-Performance Cutting 227–242
2. Mayr J, Jedrzejewski J, Uhlmann E et al (2012) Thermal issues in machine tools. CIRP Ann Manuf Technol 61(2):771–791
3. Jin C, Bo Wu, Youmin Hu (2012) Heat generation modeling of ball bearing based on internal load distribution. Tribol Int 45:8–15
4. Takab J, Khonsari MM (2013) Experimental testing and thermal analysis of ball bearings. Tribol Int 60:93–103
5. Norouzifard V, Hamedi M (2014) Experimental determination of the tool-chip thermal contact conductance in machining process. Int J Mach Tools Manuf 84:45–57
6. Gopal V, Whiting MJ, Chew JW, Mills S (2013) Thermal contact conductance and its dependence on load cycling. Int J Heat Mass Transf 66:444–450
7. Chow JH, Zhong ZW, Lin W, Khoo LP (2012) An investigation of thermal contact conductance using the lumped parameter method. Int J Therm Sci 60:114–121
8. Kagata K, Kageyama K, Kinoshita S, Yoshida A (2020) Evaluation of thermal contact resistance between two solid surfaces using photoacoustic technique. Int J Thermophys 41:131
9. Zhu Z, Zhang LW, Wu QK, Gu SD (2013) An experimental investigation of thermal contact conductance of Hastelloy C-276 based on steady-state heat flux method. Int Commun Heat Mass Transfer 41:63–67
10. Ding C, Wang R (2012) Thermal contact conductance of stainless steel-GFRP interface under vacuum environment. Exp Thermal Fluid Sci 42:1–5
11. Ishizaki T, Igami T, Nagano H (2020) Measurement of local thermal contact resistance with a periodic heating method using microscale lock-in thermography. Rev Sci Instrum 91(6):064001
12. Chow JH, Zhong ZW, Lin W, Khoo LP (2015) The development of a simple semi-empirical method to obtain a predictive model of the convective heat loss of a solid surface of practical machine size by finite difference and experimentation. Appl Therm Eng 75:789–799
13. Barrios ANS, Silva JBC et al (2014) Modeling heat transfer in die milling. Appl Therm Eng 64:108–116
14. Zhao HT, Yang JG, Shen JH (2007) Simulation of thermal behavior of a CNC machine tool spindle. Int J Mach Tools Manuf 47:1003–1010
15. Xu ZZ, Liu XJ, Kim HK, Shin JH, Lyu SY (2011) Thermal error forecast and performance evaluation for an air-cooling ball screw system. Int J Mach Tools Manuf 51:605–611
16. Kang Y, Shi X, Gao J, Li F (2016) Modeling and analyzing multivariable coupling thermal characteristics for motorized spindle. J Xi’an Jiaotong Univ 50(8):32–37
17. Fan K (2017) Research on the machine tool’s temperature spectrum and its application in a gear form grinding machine. Int J Adv Manuf Technol 90:3841–3850
18. Xu M, Cheng JC, Zhang SY (1998) Reconstruction theory of thermal conductivity depth profiles by the modulated photoreflectance technique. J Appl Phys 84(2):675–682
19. Page LG, Dirker J, Meyer JP (2016) Topology optimization for the conduction cooling of a heat-generating volume with orthotropic material. Int J Heat Mass Transf 103:1075-83
20. Chen K, Wang S, Song M (2016) Optimization of heat source distribution for two-dimension heat conduction using bionic method. Int J Heat Mass Transf 93:108–117
21. Brodnianská Z, Kotšmíd S (2021) Intensification of convective heat transfer in new shaped wavy channel configurations. Int J Therm Sci 162:106794
22. Liu T, Gao WG, Tian YL, Zhang DW, Zhang YF, Chang WF (2017) Power matching based dissipation strategy onto spindle heat generations. Appl Therm Eng 113:499–507
23. Zhang L, Yu S, Wu Y, Zhang K, Shi Q, An D (2019) Parameter optimization of a motorized spindle lubrication system using biogeography-based optimization. Adv Mech Eng 11(1):1–11
24. Zhang Y, Zheng L, Qi W, Guo K, Liu H, Liu C (2017) Optimal design of a multi-branch conducting path for area-to-point heat conduction using multi-objective optimization. Appl Therm Eng 125:1354–1367
25. Zhou H, Fan K, Gao R (2020) Fast heat conduction-based thermal error control technique for spindle system of machine tools. The International Journal of Advanced Manufacturing Technology 107:653–666
26. ISO 230-3:2020, Test code for machine tools-Part 3: Determination of thermal effects, 2020.
27. Wu L, Tan Q (2016) Thermal characteristic analysis and experimental study of a spindle-bearing system. Entropy 18(7):271

Publisher’s Note Springer Nature remains neutral with regard to jurisdictional claims in published maps and institutional affiliations.