Design and thermal characteristic analysis of motorized spindle cooling system

Yuan Zhang¹,², Lifeng Wang², Yaodong Zhang² and Yongde Zhang²

Abstract
The thermal deformation of high-speed motorized spindle will affect its reliability, so fully considering its thermal characteristics is the premise of optimal design. In order to study the thermal characteristics of high-speed motorized spindles, a coupled model of thermal-flow-structure was established. Through experiment and simulation, the thermal characteristics of spiral cooling motorized spindle are studied, and the U-shaped cooled motorized spindle is designed and optimized. The simulation results show that when the diameter of the cooling channel is 7 mm, the temperature of the spiral cooling system is lower than that of the U-shaped cooling system, but the radial thermal deformation is greater than that of the U-shaped cooling system. As the increase of the channel diameter of U-shaped cooling system, the temperature and radial thermal deformation decrease. When the diameter is 10 mm, the temperature and radial thermal deformation are lower than the spiral cooling system. And as the flow rate increases, the temperature and radial thermal deformation gradually decrease, which provides a basis for a reasonable choice of water flow rate. The maximum error between experiment and simulation is 2°C, and the error is small, which verifies the accuracy and lays the foundation for future research.

Keywords
High-speed motorized spindle, cooling system, thermal-fluid-structure coupling, thermal characteristics, optimal design

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Introduction
High-speed machine tools are an important strategic industry in the equipment manufacturing industry.¹ The high-speed motorized spindle will generate a lot of heat during the process of rotation, which will lead to the change of thermal characteristics of motorized spindle, and then affect the processing quality of the product.² According to reports, 60%–80% of the causes of machining errors of parts come from thermal deformation, and the thermal deformation of motorized spindle is the main reason.³–⁵ Therefore, the motorized spindle has a direct impact on the machining accuracy of the machine tool.⁶–⁸ High precision, high speed, and high intelligence have become the development direction in the future, and the resulting temperature rise and thermal deformation problems have always been research focus in the academic circles.

At present, thermal compensation and cooling are the main control methods of thermal deformation. Thermal compensation is predicted and compensated by algorithms such as BP neural network⁹ and adaptive chaos particle swarm optimization algorithm,¹⁰ but it is not universal. The cooling methods are divided into

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active and passive methods. Passive methods include heat sink cooling,\textsuperscript{11} heat pipe cooling.\textsuperscript{12} The active method mainly uses water, air, and oil as cooling coolant.\textsuperscript{13} Active cooling has higher efficiency and better adjustability than passive cooling.

Bossmanns and Tu\textsuperscript{14,15} successively proposed a finite difference thermal model and a power flow model for the power distribution of the electric spindle, analyzed heat transfer and heat dissipation characteristics, established a heat source model, and calculated and verified the temperature rise of each part of the electric spindle. Holkup et al.\textsuperscript{2} proposed a thermal-structure coupled electric spindle model, which can predict the temperature field distribution and thermal deformation of the spindle under specified conditions. Lee et al.\textsuperscript{16} combined finite element method with thermal analysis method to analyze the thermal characteristics of the motorized spindle, and obtained the influence of rotating speed on the temperature distribution, temperature rise, and thermal deformation of the spindle. The relationship between thermal deformation and vibration was obtained by experiments. Uhlmann and Hu\textsuperscript{17} quantitatively described the heat source and related heat transfer coefficients of the motorized spindle, and analyzed the thermal characteristics of the spindle by using the finite element method, and obtained the temperature field distribution and thermal deformation. Ma et al.\textsuperscript{18} proposed a three-dimensional finite element analysis model, which considered contact thermal resistance and the thermal displacement of the bearing parts. After analyzing the temperature field distribution and thermal deformation of motorized spindle, it is found that this method can improve the prediction accuracy. Zhang et al.\textsuperscript{19} established a temperature field prediction model based on the measured data of the outer surface temperature of motorized spindle, and optimized the heat transfer coefficient by genetic algorithm, which improved the prediction accuracy.

Chien and Jang\textsuperscript{20} uses a combination of experiment and numerical analysis to study the effects of different heat sources and different flow rates on fluid movement and temperature distribution. Weber et al.\textsuperscript{21} used lumped parameter network models and computational fluid dynamics (CFD) methods, the temperature field distribution of motorized spindles with single spiral and double spiral channel structures is simulated and predicted. Li et al.\textsuperscript{22} used the design of experiment (DOE) method to optimize the size of the parallel cooling channel + spiral cooling channel, and analyzed the cooling effects of different performance cooling oils. The analysis methods of various scholars on the thermal characteristics of the electric spindle are shown in Table 1.

The above research on the thermal characteristics of high-speed motorized spindles is based on thermal and thermal-structural coupling. The heat transfer coefficient of coolant needs to be calculated manually, and it is a fixed. It can’t change with the size of the cooling channel, resulting in a decrease in calculation accuracy. The fluid analysis can only obtain temperature field distribution, but not the thermal deformation. At present, scholars who use the thermal-fluid-solid coupling analysis method only analyze and study cooling oil with different properties and optimize the size of the cooling channel. At present, there is no article that analyzes the influence of water cooling, flow rate, and different materials on the thermal characteristics of the electric spindle based on the thermo-fluid-solid coupling method. In view of this, this paper established a thermal-fluid-structure coupling analysis model to analyze the thermal characteristics of the motorized spindle, optimize the design of the spiral cooling system, and study the influence of the cooling water flow on it. The correctness of the simulation results was verified through experiments. This provides a favorable basis for the optimal design of the thermal characteristics of motorized spindle.

### Analysis of heat generation and heat transfer of motorized spindle

The main heat source of a high-speed motorized spindle consists of two parts, one is the internal motor, the other is the front and rear bearings.\textsuperscript{23} During the

| Year | First author | Analytical method |
|------|--------------|-------------------|
| 1999 | Bossmanns\textsuperscript{14,15} | Thermal analysis |
| 2001 | Chien\textsuperscript{20} | Fluid analysis |
| 2008 | Holkup\textsuperscript{2} | Thermal-structure coupling analysis |
| 2010 | Uhlmann\textsuperscript{17} | Thermal-structure coupling analysis |
| 2012 | Lee\textsuperscript{16} | Thermal-structure coupling analysis |
| 2015 | Ma\textsuperscript{18} | Thermal-structure coupling analysis |
| 2016 | Weber\textsuperscript{21} | Fluid analysis |
| 2017 | Zhang\textsuperscript{19} | Thermal analysis |
| 2020 | Li\textsuperscript{22} | Thermal-fluid-structure coupling analysis |
rotation of motorized spindle, due to motor loss and bearing friction, a large amount of heat will be generated. The accumulation of heat will cause thermal deformation of motorized spindle, which will affect the machining accuracy and quality. Therefore, water cooling systems is usually used to cool the stator and bearing components of the motor in engineering. The structure and flow field model of spiral cooled motorized spindle are shown in Figure 1. The material properties are shown in Table 2. Characteristics of cooling water are shown in Table 3.

Motor loss and heat generation

Three-phase asynchronous motor is used for motorized spindle. Analysis of its working principle shows that the loss and heating of the motor consists of mechanical loss, electrical loss, magnetic loss, and additional loss. The first three parts account for the majority of the total energy loss. The additional loss caused by fundamental leakage flux and high-order harmonic leakage flux only accounts for 1%–5% of the total loss, which can be ignored. Therefore, the formula for calculating motor loss is as follows.

**Mechanical loss.** There is a gap between the rotor and the stator, so that the outer surface of the rotor contacts with the air. In the running process, the friction loss between the rotor and the air is called mechanical loss. It can be calculated by equation (1):

\[ P_a = C_a \omega^3 r^4 L \pi \rho \]  

Where \( P_a \) is the mechanical power loss, \( \rho \) is the air density, \( \omega \) is the angular velocity, \( r \) is the rotor radius, \( L \) is
the length of the rotor, and \( C_a \) is the air flow resistance coefficient.

**Electrical loss.** Electrical loss, also known as copper loss, is composed of basic copper loss and additional copper loss. It can be calculated by equation (2):

\[
P_b = mI^2R = mI^2\rho_sK/U
\]  
(2)

Where \( P_b \) is the power loss, \( m \) is the number of motor phases, \( I \) is the current passing through the stator and rotor windings of the motor, \( \rho_s \) is the resistivity of the winding coil, the size is related to the temperature, \( R \) is the single-phase winding resistance, \( K \) is the single-phase winding length, \( U \) is the cross-sectional area of the winding coil.

**Magnetic loss.** Magnetic loss includes hysteresis loss and eddy current loss. It can be calculated by equation (3):

\[
P_c = k_iB_{max}^2 + K_c(B_{max})^2
\]  
(3)

Where \( P_c \) is the magnetic loss, \( k_i \) is the related constant of silicon steel sheet, \( f \) is the electromagnetic field exchange frequency, \( B_{max} \) is the maximum value of magnetic induction, \( i \) is an empirical constant, and its value is related to the type of material and \( B_{max} \), when \( B_{max} < 1 \text{T} \), \( i \) takes 1.6, when \( B_{max} \geq 1 \text{T} \), \( i \) takes 2, \( K_c \) is the coefficient related to the material.

**Bearing friction heat**

During the process of high-speed rotation of angular contact ball bearings, a large amount of heat is generated due to the friction between rolling bodies and inner and outer rings. The generation of heat is related to the viscosity of the lubricating oil and the contact between the rolling bodies and the inner and outer rings. The heat can be calculated according to Pallgren’s formula (4):\(^{24}\)

\[
Q = 1.047 \times 10^{-4}Mn
\]  
(4)

Where \( Q \) is the heat generated by the bearing, \( M \) is the total friction torque of the bearing, \( n \) is the bearing speed.

The total friction torque of the bearing consists of two friction torques: \( M_1 \) is related to the applied load, and \( M_2 \) is related to the viscosity of the lubricant. Therefore, the total friction torque can be expressed by equation (5):

\[
M = M_1 + M_2
\]  
(5)

The friction moment \( M_1 \) related to the applied load can be calculated by formula (6):

\[
M_1 = z\left(\frac{F_s}{C_s}\right)^\gamma (0.9F_a \cot \alpha - 0.1F_r)d_m
\]  
(6)

Where \( d_m \) is the bearing pitch circle diameter, \( F_s \) is the equivalent static load of the bearing, \( C_s \) is the basic static load rating of the bearing, \( z \) and \( \gamma \) are respectively 0.001 and 0.33, \( F_r \) is the bearing radial load force, \( F_a \) is the bearing axial load force.

The friction torque \( M_2 \) related to the viscosity of the lubricant can be calculated by equation (7):

\[
M_2 = \begin{cases} 
160 \times 10^{-7}f_0d_m^3 & \text{if } \nu_0n < 2000 \\
10^{-7}f_0(\nu_0n)^3d_m^3 & \text{if } \nu_0n \geq 2000
\end{cases}
\]  
(7)

Where \( \nu_0 \) is the kinematic viscosity of the lubricant, \( n \) is the bearing speed, \( f_0 \) is related to the bearing type and lubrication method.

Using the above calculation formula, the motor loss is 3.41 kW, the friction heat of the front bearing is 470.4 W, and the friction heat of the rear bearing is 235.6 W. It is assumed that all motor losses are converted into heat, one third of which is generated by the motor rotor and the remaining two thirds by the motor stator.\(^{25}\)

**Coolant Reynolds number**

The flow of cooling water can be divided into three types: laminar flow, transitional flow, and turbulent flow, which have a direct impact on the cooling effect of the motorized spindle and can be judged by Reynolds number. The Reynolds number can be calculated by equation (8):

\[
R_e = \frac{VD}{\mu}
\]  
(8)

Where \( R_e \) is the Reynolds number of water, \( D \) is the final size of the water channel, \( \mu \) is the kinematic viscosity of water, \( V \) is the flow rate of water. It can be calculated by formula (9):

\[
V = \frac{Q}{A}
\]  
(9)

Where \( Q \) is the flow of water, \( A \) is the cross-sectional area of the water pipe.

**Mathematical model of heat transfer**

During the working process, the heat generated by the motor and the bearing is transferred to each component through heat conduction, and each component conducts heat radiation and convection heat exchange with the surrounding air. Meanwhile, the heat exchange between the components follows the law of...
conservation of energy. Therefore, the heat conduction can be solved according to Fourier’s law formula (10):

\[ \Phi = -K \frac{\partial T}{\partial r} A = qA \]  

(10)

Where \( \Phi \) is the heat through the section, \( K \) is the thermal conductivity, \( T \) is temperature, \( r \) is the coordinate on the heat conducting surface, \( A \) is the cross-sectional area of heat transfer, \( q \) is the heat flux density.

In \( d_r \) time, the net heat of the micro-element body imported and exported in the \( x, y, z \) direction is

\[
\Delta \Phi_d = \left[ \frac{\partial q_x}{\partial x} + \frac{\partial q_y}{\partial y} + \frac{\partial q_z}{\partial z} \right] d_x d_y d_z d_r
\]  

(11)

The calorific value of the heat source in the micro-element body in \( d_r \) time is

\[
\Delta \Phi_v = \Phi d_x d_y d_z d_r
\]  

(12)

The increase in the enthalpy of the micro-element body in \( d_r \) time is

\[
\Delta E = \rho c \frac{\partial T}{\partial r} d_x d_y d_z d_r
\]  

(13)

According to the energy balance, \( \Delta E = \Delta \Phi_d + \Delta \Phi_v \), and by inserting equation (10) into the above equation, the three-dimensional unsteady heat conduction equation can be obtained

\[
\rho c \frac{\partial T}{\partial r} = \frac{\partial}{\partial x} \left( r K \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left( K \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left( K \frac{\partial T}{\partial z} \right) + \Phi
\]  

(14)

Where \( \rho \) is the density, \( c \) is the specific heat capacity at constant pressure.

When the heat transfer coefficient \( \varepsilon \) and temperature between the boundary surface and the adjacent fluid are known conditions, the boundary condition is

\[
K \left( \frac{\partial T}{\partial r} \right) = \varepsilon \left( T - T_f \right)
\]  

(15)

Heat convection can be calculated according to Newton’s law of cooling formula (16):

\[
\Phi = hA \Delta T
\]  

(16)

Where \( \Phi \) is the heat transfer power, \( h \) is the heat transfer coefficient, \( A \) is the heat transfer area, \( \Delta T \) is the temperature difference.

The convective heat transfer between the cooling water and the cooling system should follow the conservation laws of mass, momentum, and energy. The cooling water is a kind of incompressible fluid, so the three conservation laws are:

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0
\]  

(17)

\[
\rho c_p \left[ \frac{\partial T}{\partial t} + \frac{\partial (uT)}{\partial x} + \frac{\partial (vT)}{\partial y} + \frac{\partial (wT)}{\partial z} \right]
\]  

(19)

Where \( u, v, w \) are the speeds in \( X, Y, \) and \( Z \) directions respectively, \( P \) represents the pressure on the fluid cell, \( f_x, f_y, f_z \) represent the unit mass force on the micro-element, \( c_p \) represents the specific heat capacity, \( T \) is the temperature, \( \rho \) represents the density of the fluid.

After calculation, the Reynolds number is greater than 4000, which belongs to the state of turbulent motion. Choose \( k-\varepsilon \) model to solve the problem. Turbulence kinetic energy \( k \) equation and the dissipation rate \( \varepsilon \) equation are expressed as follows:

\[
\frac{\partial}{\partial t} \left( \rho k \right) + \frac{\partial}{\partial x_j} \left( \rho \varepsilon u_j \right) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_s}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_M + G_h - \rho e - Y_M + S_M
\]  

(20)

\[
\frac{\partial}{\partial t} \left( \rho \varepsilon \right) + \frac{\partial}{\partial x_j} \left( \rho \varepsilon u_j \right) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_s}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_{1_\varepsilon} \frac{\varepsilon}{k} \left( G_k + C_{3_s} G_h - C_{2_s} \rho \varepsilon \right) + S_\varepsilon
\]  

(21)

Where \( \mu_s \) is the turbulence viscosity coefficient, the values of \( a, b, \) and \( c \) are 1.44, 1.92, and 0.09, Prandtl numbers \( \sigma_k \) and \( \sigma_\varepsilon \) are 1.0 and 1.3 respectively.

The heat exchange between the outer surface of motorized spindle and the surrounding environment includes natural convection and thermal radiation.
According to related literature, the combined heat transfer coefficient is 9.7 W/m²·K.

Based on the above theoretical foundation, the finite element method can be used to solve the heat at any point, which provides a basis for the following calculations.

**Simulation analysis and optimal design of thermal characteristics of motorized spindle**

The motorized spindle generates heat due to the mechanical loss, electrical loss, magnetic loss and bearing friction of the internal motor, and then dissipates heat through heat conduction, heat convection, and heat radiation, finally keeps the temperature and thermal deformation unchanged. At this moment, the motorized spindle reaches thermal balance. In order to reduce the temperature and thermal deformation, this paper takes the cooling system as the research object, and studies and optimizes its thermal characteristics.

**Boundary condition setting and meshing method**

The study of thermal characteristics of the motorized spindle is to analyze and calculate its heat source, temperature rise, temperature field distribution, and thermal deformation. Therefore, it is necessary to determine the boundary conditions before analyzing the thermal characteristics. At first, it is assumed that the operating conditions of the motorized spindle are: The rotation speed under no-load condition is 18,000 r/min, The ambient temperature is 20°C, The cooling water flow rate is 12 L/min, the water temperature is 20°C, The kinematic viscosity of lubricating oil is 68 cst. According to the calculation of the above conditions, the boundary conditions are shown in Table 4.

![Figure 2. Mesh model.](image)

**Table 4. Simulation boundary conditions.**

| Boundary conditions                           | Value   |
|----------------------------------------------|---------|
| Stator heat generation (W)                   | 2273    |
| Rotor heat (W)                               | 1137    |
| Heat generation of front bearing (W)         | 470.4   |
| Heat generation of rear bearing (W)          | 235.6   |
| Heat transfer coefficient between surface and air (W/m²·K) | 9.7     |

Define the cooling water inlet and outlet of the cooling system model, optimize the local mesh, set the expansion layer in the fluid domain, select the CFD fluid mesh type, and then divide into tetrahedron and hexahedron meshes. The number of meshes is 11,555,714. The mesh model is shown in Figure 2.

Under the platform of ANSYS Workbench, the coupling analysis module is constructed by using Fluent module and Static Structure module. The turbulence model adopts the standard $k-\varepsilon$ model, and the solver adopts the coupled algorithm for first order implicit transient analysis.

**Thermal characteristics analysis and mesh independence verification of spiral cooling motorized spindle**

In order to study the thermal characteristics of the motorized spindle of the spiral cooling system, the above boundary conditions are loaded on the finite element model to solve the problem. The thermal characteristics are shown in Figure 3.

It can be seen from the analysis in Figure 3(a) that the diameter of the cooling channel is 7 mm, and the maximum temperature is 32°C, and the temperature band at the front end spiral part is wide, which is caused by insufficient cooling, while the middle part is fully cooled, the temperature band is narrow, and the front end temperature is slightly lower. According to the analysis in Figure 3(b), the total thermal deformation is 0.039785 mm from the analysis in Figure 3(c), it can be seen that the maximum radial thermal deformation occurs in the motor part, and the deformation amount is 0.018898 mm, the spiral groove is a part of the cooling channel and mainly cools the stator part of the motor. Analysis shows that the large thermal deformation is mainly caused by the opening of the spiral groove on the shell, and the thin carbon fiber shell does not provide support, which is the reason of local deformation. At present, in the industry, axial thermal deformation can be controlled by thermal compensation technology, but there is no better method to control radial thermal deformation. Therefore, we need to improve and optimize the structure of the cooling channel without changing the external dimensions.

Before optimizing the design, we need to verify the validity of the simulation results. Therefore, we carried out mesh independence verification analysis. The divided meshes are still tetrahedral meshes and hexahedral meshes, with the number of meshes is 12,156,226,
and the cloud picture of temperature field distribution is shown in Figure 4.

The comparison between Figures 3 and 4 shows that when the mesh number changes, the temperature field distribution and temperature are basically unchanged, which can verify the mesh independence.

**Design and optimization of U-shaped cooling system**

Aiming at the problems of high local temperature and large thermal deformation of the spiral cooling system, a U-shaped cooling system with internal cooling channels is designed optimally. Because the main body of the internal cooling channel is a whole, the resistance to thermal deformation can be improved. In order to determine the size of the cooling channel and meet the design requirements, four models with diameters of 7, 8, 9, and 10 mm were designed for analysis. When the flow rate is 12 L/min, the relationship curve of temperature, radial thermal deformation, and channel size are shown in Figure 5.

From the analysis of Figure 3(a), when the channel diameter is 7 mm, the temperature of U-shaped cooling system is 3°C higher than that of spiral cooling system,
and the radial thermal deformation is 0.005242 mm less. With the increase of channel diameter, the temperature and radial thermal deformation of U-shaped cooling system gradually decrease. When the diameter is 10 mm, the temperature is 1.5°C lower than that of the spiral cooling system, and the radial thermal deformation is 0.009077 mm lower. Therefore, a model with a diameter of 10 mm is chosen for modeling. The structural model of the design is shown in Figure 6. The properties of materials are shown in Table 5.

As the cooling system is optimized only for the design, the rest is unchanged, and the boundary conditions are the same as the above. When the flow rate is 12 L/min, the thermal state characteristics of the U-shaped cooling system are shown in Figure 7.

It can be seen from the analysis in Figure 7(a) that the temperature of the motor part is the highest, followed by the front bearing, and the temperature of the rear bearing is the lowest, with the highest temperature being 30.5°C. From the analysis in Figure 7(b), the total deformation of the U-shaped cooling system is 0.026229 mm. From the analysis of Figure 7(c), it can be seen that the rear end of the cooling system has the largest radial thermal deformation, but the rear end deformation has little effect on processing and can be ignored. Therefore, the largest radial thermal deformation is in the motor part, with a deformation of 0.009821 mm. Compared with Figure 3, it can be seen that the temperature, total thermal deformation and radial thermal deformation are lower than those of the spiral cooling system, and the performance of the U-shaped cooling system optimized based on the above aspects is better.

In order to make better use of the performance of the U-shaped cooling system, reasonably select the cooling water flow rate and achieve the purpose of energy saving and effective cooling, the influence of the flow rate on the thermal characteristics of the motorized spindle was studied. The nephogram of temperature distribution under different flow rates is shown in Figure 8, and the cross section of radial thermal deformation is shown in Figure 9.

According to the analysis of Figures 8 and 9, the overall distribution rule is still consistent. When the cooling water flow rate is 4, 6, 8, and 10 L/min, the temperature is 41, 36, 33.5, and 31.5°C, and the radial thermal deformation is 0.02314, 0.016813, 0.013341, and 0.011238 mm. The relationship curve among flow rate, temperature, and radial thermal deformation is shown in Figure 10.

It can be seen from the analysis of Figure 10 that with the increase of the cooling water flow rate, the temperature and radial thermal deformation of the motorized spindle gradually decrease, and the changes

| Part number | Materials             | Density (kg/m³) | Poisson's ratio | Elastic modulus (N/m²) | Specific heat (J/kg °C) | Thermal conductivity (W/m °C) | Thermal expansion coefficient (°C⁻¹) |
|-------------|-----------------------|-----------------|----------------|------------------------|-------------------------|-------------------------------|-----------------------------------|
| 1           | Aluminum alloy (3.0205) | 2700            | 0.3897         | 7e + 10                | 940                     | 204                           | 2.4e-05                           |
| 2           | 42CrMo                | 7820            | 0.29           | 2.06e + 11             | 460                     | 44                            | 1.3e-05                           |
| 3           | Ordinary carbon steel  | 7800            | 0.28           | 2.1e + 11              | 440                     | 43                            | 1.3e-05                           |

Figure 6. U-shaped cooling system motorized spindle: (a) motorized spindle model of U-shaped cooling system and (b) U-shaped cooling system fluid domain model. 1. Front cover; 2. Cooling system body; 3. Rear bearing support; 4. Rear cover.
Figure 7. Thermal characteristics cloud diagram of electric spindle of U-shaped cooling system: (a) cloud picture of temperature distribution, (b) cloud picture of total thermal deformation, and (c) radial thermal deformation cloud picture.

Figure 8. Temperature field distribution under different flow rates: (a) cloud picture of temperature distribution when the flow rate is 4 L/min, (b) cloud picture of temperature distribution at a flow rate of 6 L/min, (c) cloud picture of temperature distribution under the flow rate of 8 L/min, and (d) cloud picture of temperature distribution at a flow rate of 10 L/min.
is obvious at the beginning, and then gradually becomes gentle. So as to reasonably select the cooling water flow rate and achieve the aims of energy saving and effective cooling.

Motorized spindle temperature rise experiment

In order to better monitor the state of the machine tool and reduce the influence of thermal errors, Clough et al. tested various monitoring methods and emphasized the value of online monitoring of thermal imaging. Infrared thermal imaging camera has the advantages of small size, light weight, high sensitivity, wide measurement range, and non-contact measurement. It is an advanced and effective technology in fault diagnosis and detection, and is widely used in industry, aerospace, transportation, medical, and other fields. The motorized spindle experimental device is shown in Figure 11.

Experimental program

The Testo 865 thermal imaging camera is used to measure the temperature field distribution of the motorized spindle. The scheme is as follows:

1. Check whether the oil and gas lubrication system, water cooler, thermal imager, and other equipment are normal, and set the oil and gas lubrication quantity, emissivity, and other values.
2. Set the cooling water temperature and ambient temperature to 20°C, and adjust the flow rate to 4, 6, 8, 10, 12 L/min and the speed of 18,000 r/min as required. Use the infrared camera to capture the initial state of the spindle as the initial data, and then start the motorized spindle.
3. Shoot the same position every 30 s until the temperature of the electro-spindle stabilizes and no longer changes. In order to ensure that the motorized spindle is completely cooled, the experimental interval will not be less than 24 h.
Figure 11. Experimental motorized spindle: (a) spiral cooling motorized spindle and (b) U-shaped cooling motorized spindle.

Figure 12. Thermography: (a) thermal image of spiral-cooled motorized spindle at a flow rate of 12 L/min, (b) thermal imaging of U-shaped cooling motorized spindle at a flow rate of 4 L/min, (c) thermal imaging of U-shaped cooling motorized spindle at a flow rate of 6 L/min, (d) thermal imaging of U-shaped cooling motorized spindle at a flow rate of 8 L/min, (e) thermal imaging of U-shaped cooling motorized spindle at a flow rate of 10 L/min, and (f) thermal imaging of U-shaped cooling motorized spindle at a flow rate of 12 L/min.
Because the materials of different parts of the motorized spindle are different, the emissivity is also different. In order to improve the accuracy of the experiment, we use polyvinyl chloride insulating tape to stick on the surface of the electric spindle so that the surface has the same emissivity. Query shows that the emissivity is 0.94.27,28 To see more intuitively and compare more conveniently, we processed the experimental results. The thermal imaging of the spiral-cooled motorized spindle is shown in Figure 12(a), and the thermal imaging of the U-shaped cooling motorized spindle is shown in Figure 12(b) to (f).

Table 6 shows the temperature simulation results, experimental results and errors of the spiral cooling system and the U-shaped cooling system at different flow rates, as well as the radial thermal deformation data. By analyzing Figure 12, it can be seen that the temperature field distribution of motorized spindle of the spiral cooling system and the U-shaped cooling system obtained by the simulation analysis is consistent with that of thermal imaging obtained by experiment. It can be seen from Table 6 that the error between the simulation results and the experimental results is very small, and the maximum error is 2°C, which can verify the accuracy of the simulation analysis. Figure 13 shows the temperature contrast curve of simulation and experiment for motorized spindle of U-shaped cooling system.

It can be seen from Figure 13 that the temperature gradually decreases with the increase of cooling water flow rate. At first, the temperature drops significantly, and then it gradually leveled off. The change trend of experiment is consistent with that of simulation, which further proves the accuracy of the simulation. The simulated value is slightly higher than the experimental value, because the calculated power loss is higher than the actual power loss, coupled with the simplification of the finite element model and the change of the ambient temperature, the calculation errors is finally caused.

**Analysis of results**

Because the materials of different parts of the motorized spindle are different, the emissivity is also different. In order to improve the accuracy of the experiment, we use polyvinyl chloride insulating tape to stick on the surface of the electric spindle so that the surface has the same emissivity. Query shows that the emissivity is 0.94.27,28 To see more intuitively and compare more conveniently, we processed the experimental results. The thermal imaging of the spiral-cooled motorized spindle is shown in Figure 12(a), and the thermal imaging of the U-shaped cooling motorized spindle is shown in Figure 12(b) to (f).

Table 6 shows the temperature simulation results, experimental results and errors of the spiral cooling system and the U-shaped cooling system at different flow rates, as well as the radial thermal deformation data. By analyzing Figure 12, it can be seen that the temperature field distribution of motorized spindle of the spiral cooling system and the U-shaped cooling system obtained by the simulation analysis is consistent with that of thermal imaging obtained by experiment. It can be seen from Table 6 that the error between the simulation results and the experimental results is very small, and the maximum error is 2°C, which can verify the accuracy of the simulation analysis. Figure 13 shows the temperature contrast curve of simulation and experiment for motorized spindle of U-shaped cooling system.

It can be seen from Figure 13 that the temperature gradually decreases with the increase of cooling water flow rate. At first, the temperature drops significantly, and then it gradually leveled off. The change trend of experiment is consistent with that of simulation, which further proves the accuracy of the simulation. The simulated value is slightly higher than the experimental value, because the calculated power loss is higher than the actual power loss, coupled with the simplification of the finite element model and the change of the ambient temperature, the calculation errors is finally caused.
spiral-cooled motorized spindle, the cooling system of the spiral-cooled motorized spindle is optimized, and the original cooling system is changed to a U-shaped cooling system.

2. According to the simulation analysis, the optimized motorized spindle temperature is 1.5°C lower than the original, the total thermal deformation is reduced by 0.013556 mm, and the radial thermal deformation is reduced by 0.009077 mm, which improves the performance of the motorized spindle. Within a certain range, with the increase of cooling water flow rate, the temperature and radial thermal deformation decrease gradually, which provides a basis for the reasonable selection of the cooling water flow rate.

3. It can be seen from the experimental results that the overall temperature distribution law and change trend of the experiment and the simulation are consistent. The maximum error is 2°C, and the error is small, which can verify the accuracy of the simulation analysis and lay a favorable foundation for further research.

At present, due to the limitation of experimental conditions, there is no thermal deformation measurement experiment. In future work, we will further improve the experiment and carry out the experimental study on thermal deformation.

Declaration of conflicting interests
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