Effect of a cooling unit on high-speed motorized spindle temperature with a scaling factor

Zheng De-xing¹,² · Chen Weifang²

Received: 20 August 2021 / Accepted: 19 February 2022 / Published online: 24 February 2022
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Abstract
Forced cooling, as an efficient way of heat dissipation, significantly affects the spindle temperature. Although a full cooling passage was factored into the finite element analyses by some scholars, often only the model for the front/rear of a spindle is need for the purpose of the thermal evaluation simplification and data overhead reduction in engineering. For this, we devote to exploring how the coolant passage affects the heat dissipation of the front/rear halves of spindles by a constructed scaling factor, and accordingly create a simpler thermal evaluation model of spindles. Firstly, the experiments about a coolant unit effect on the thermal performance of spindles were first implemented, and thus, the temperature difference was noticed under various coolant parameter settings. Through the regressive analysis on the test data, the peak temperature area along axial direction was found and then the scaling factor was proposed to describe the effect of a cooling system on the temperature field of the front/rear half spindle. A similar structural coefficient, meanwhile, was also set up to state clearly the role of motor armature heating in the spindle temperature rise. Next, the thermal equivalent convection for a coolant passage was modeled based on the thermal resistance theory. Based on the abovementioned work, we planned a novel thermal network for only the front half of a motored spindle, in which the cooling mechanism was integrated, and the structural constraints were considered by the aid of the proposed scaling factors. Finally, the simplification and solution for this developed thermal grid were done, and the corresponding numerical results were compared with the test values. At the same time, the bearing heating with no or only 50% of a coolant unit, for the sake of better contrast and validation, was also simulated. The comparison results are indicative of a better agreement with real values when the proposed scaling factors are employed to considering the cooling unit impact on the high-speed motorized spindle temperature.

Keywords Motorized spindle · Coolant system · Scaling factor

1 Introduction
Motorized spindle unit, as the core component of high speed, efficiency, and precision machining, is widely employed in various machine tools [1, 2]. Many studies demonstrate that the performance of high-speed motorized spindles is closely related to their thermal growth. To accurately assess the thermal behaviors of spindles, the heat generation and dissipation were investigated successively [3, 4]. As two key parts influencing the operation characteristics, bearing and motor are also the main heat sources to determine the thermal behaviors of spindles. To dissipate their heating for lower operation temperature and then more ideal working accuracy has been fully valued by researchers. In this process, many numerical models by finite element theory, lots of physical equations relating to heat transfer mechanism, and various empirical formulas based on test data were constructed to analyze quantitatively the corresponding heat exchange [5–7]. In the meantime, numerous optimization algorithms were also proposed to further explore the heat sources, temperature profile, and thermal deformation of spindles [8, 9].

In current studies, however, an enormous amount of research effort only considered the power loss of bearings, the heat conduction between bearings and surrounding parts, and the convection effect between bearings and
coolant/lubricant as well, and yet how the heat given out by motor armatures is transmitted has not been thoroughly addressed thus far. In this area, some scholars also discussed the motor heating effect on the spindle temperature variation, for example Zhang and Chen [10] constructed a power flow model of built-in motor to explore the temperature field change and thermal error of a motorized spindle unit, while the contact statuses between parts were ignored; Zhang et al. [11] established a finite element model for motorized spindles, in which the motor is only a thermal source. Lately, Meng et al. [12] took various fractal parameters into a mathematical model to describe the thermal contact conduction between rotor and stator, but the detailed thermal transmission between rotor/stator and surroundings was not fully considered. Well, to sum up, the detailed heat transfer of motor armatures was often dismissed.

Furthermore, cooling, as an efficient way to lower the operating temperature of high-speed motorized spindles, is usually needed in two locations: bearings and motor [13]. The cooling for bearings is often accompanied by lubrication, whose corresponding impact on spindle temperature rise was investigated by many scholars, such as Michael et al. [14]. In comparison, the energy loss of motor armatures is much bigger and therefore is regarded as the most important thermal source of a spindle. So, a separate coolant system, for the purpose of dissipating the motor heating fast, is very essential. Here are some passive/active ways to cool electric motors, for example the forced/free air cooling, the liquid cooling by oil/water, the loop thermosyphon, the evaporative cooling using liquid boiling, and the additional circuit for enhanced heat transfer [13, 15, 16]. Among them, the air cooling is more suitable for small power motors, and the evaporative cooling is mainly employed in super high-power applications. The forced air and liquid cooling with simpler cooling structures, by contrast, are more practical. After further comparison, the conventional forced convection by oil or water is more efficient and as a result is most popular in various rotation spindles, especially high-speed motorized spindles.

Firstly, Biesack and Loomis [17] set the helical passage for a motorized spindle housing. With the aid of a separate channel, Shigenori and Narashino [18] later separated the cooling from bearing lubrication. After this, Xia et al. [19] designed the fractal tree-like model and applied it into the cooling sleeve of spindle; Kang et al. [20] proposed a novel shaft core cooling approach to further reduce the spindle temperature; by ELM and GA, Liu et al. [21, 22] developed an intelligent control method and accordingly constructed a thermal simulation speculation-based active coolant control strategy.

Various forced cooling structures by water/oil, as stated above, were designed successively. However, there is still a long way to go till these novel cooling schemes can be used in engineering. By comparison, to improve the existing designs is more pragmatic in making heat exchange better. In this connection, Grama et al. [23] constructed a control strategy (Cooler Trigger Model) with higher cooling effectiveness; Mori et al. [24] developed an on--off control way to reduce the power loss of the coolant unit with a hot gas bypass. Again, Chien and Jang [25] integrated the helical cooling passageway into a 3-D fluid model of motorized spindle, and next, both numerical and experimental studies of spindle temperature distribution were implemented; Zhu et al. [26] carried out the finite element analysis and experimental study to discuss the effect of shaft core cooling. However, these above finite element models are with excessive data overhead, and thus are more usually adopted in scientific researches. The concise heat transfer models based on thermal network method, in contrast, have a much lower data overhead and show more specific heat exchange between parts, and therefore are more practical in engineering applications [27]. Lately, the coolant passage was modeled and planned into a complete thermal network of spindle as a whole by Zhou et al. [28], but the developed thermal evaluation model of spindles is still too complex.

For a faster model, often scholars only modeled the first half of a spindle, yet it is a fact of life that the cooling units are either not integrated into the presented thermal grid models [4, 7], or they are taken for granted by researchers that 50% of a cooling unit is involved in the heat dissipation of the front/rear half spindle [29]. Obviously, how the substructures outside a coolant passage affect the heat dissipation of cooling system is ignored.

As noted above, a lot of work has been done by researches; nevertheless, the thermal performances of cooling systems have not been thorough up to now. Some scholars investigated how the cooling parameters impact the temperature, and the cooling systems were also integrated as a whole into the finite element models to explore the spindle temperature by some others, but how the coolant passage affects the heat dissipation of the front and rear half of a spindle separately was seldom mentioned. Consequently, there is still a lack of an accurate, simple, and suitable model to evaluate the influence of cooling units on spindle temperature so far.

In this paper, the coolant passage effect on spindle heating was explored experimentally. Through the regressive analysis on test data, the scaling factor was introduced to represent the role of a coolant passage in the thermal estimation.

### 2 Experimental study about coolant system effect on motorized spindle temperature

#### 2.1 Experiment setup

To investigate the role of coolant systems in cooling motorized spindles, the test project to detect the temperature of
motorized spindle (1), as shown in Fig. 1, was designed to analyze the effect of various cooling settings on heat exchange, in which the initial axial load was 400 N. In experimenting, we changed the coolant flux and working temperature by adjusting oil cooler (7), and the ambient temperature, shaft speed, and oil/air flow rate for lubrication were regulated through the air conditioner, variable-frequency drive (6), and oil-air lubrication device (4) respectively. In order to monitor the spindle temperature change under different cooling parameters, 8 thermal resistance temperature sensors (12–19) were evenly placed on the outer surface of spindle cooling sleeve and 2 built-in thermistors (10–11) on the outer rings of front and rear spindle bearings as well. At the same time, the thermal resistances (8) and (9) were planned to measure the temperature of discharged oil-air and coolant separately. It should be noted that the spindle in the test is Setco231A240, whose supporting bearing is 7009AC from SKF. The double helical design is employed in the cooling system of spindle, and the corresponding cooling media is the spindle oil no. 5. More details on test rig are listed in Table 1.

### Table 1 Description on test rig

| Number | Description                  | Model       | Manufacturer                      |
|--------|------------------------------|-------------|-----------------------------------|
| 1      | Motorized spindle            | Setco231A240| Setco Inc. in American            |
| 2      | Data acquisition system      | DH5922      | Jiangsu Donghua Co., Ltd in China |
| 3      | Industrial personal computer | IPC-510     | Advantech. Co., Ltd in Taiwan     |
| 4      | Oil-air lubrication rig      | AMO-IIIDS   | Lube Corp in Japan                |
| 5      | Air compressor               | HET-120     | Xiamen East Asia Machinery        |
|        |                              |             | Industrial Co., Ltd. in China     |
| 6      | Variable-frequency drive     | HS71        | Emerson in American               |
| 7      | Oil cooler                   | CO4PTS-446 | Point Corp in China               |
| 8–19   | Thermal resistance sensor    | PT100       | Jiangsu Donghua Co., Ltd in China |

2.1.1 Experiment results

In this test project, the mean temperature of checkpoints was employed to evaluate the impact of cooling unit on the thermal characteristics of the motored spindle. Figure 2 depicts the temperature difference on the spindle housing surface from front bearing to rear bearing when the coolant flow rate was changed from 3 to 4.5L/min. The other test conditions are as follows:

1. The ambient air is 22 °C, and the oil cooler operates at 20 °C.
2. The flow rate of lubricating oil is 0.6 mL/h, and it’s viscosity is 22 mm²/s.
3. The air flow flux is regulated by adjusting the air compressor, and it’s value is 2.5e-3 m³/s.

The evident results are as below.

1. The temperature distribution is not uniform along the spindle axis. The maximum error is nearly 0.8 °C when the spindle operates at 12000 rpm, and this deviation is less than 1.2 °C at 8000 rpm.
2. For Fig. 2, all temperature peaks under various coolant fluxes appear at the 5th checkpoint on the cooling sleeve surface. Obviously, this checkpoint is not the midpoint of the spindle. A clear conclusion, then, is that the front half of the spindle is not same to the rear half in heat dis-
sipation. And this may be attributed to the influence of the substructures outside the cooling unit on heat transfer.

3. To compare 1st thermistor (close to the front-end of spindle) with 8th thermal resistance sensor (near the rear end of spindle), there is only a slight difference in temperature, whose value is about 0.4 °C.

To analyze the heat sources and bearing arrangement of the motorized spindle Setco231A240, the added findings are as follows:

1. In contrast with the heat generation of bearings, the motor armature generates more heat in operation. Consequently, there is still a higher temperature on the outer surface of spindle cooling sleeve than the front/rear end, though a lot of heat is taken away by the cooling passageway.

2. The bearings, as seen Fig. 1, are arranged in Setco231A240 back to back, and therefore, the axial load is distributed equally to each bearing set. As a result, the bearings have a same power loss and a little temperature difference.

Based on the above experimental findings, the measurements of the 1st, 5th, and 8th checkpoints are obviously more representative in reflecting the thermal change of the spindle. So, we employed them to further explore the effect of a cooling unit on the spindle temperature field.

The test results with coolant temperature is presented in Fig. 3, where the coolant flow rate (4.5L/min) is fixed, the coolant temperature is regulated between 18 and 22 °C by setting oil cooler, and the other test conditions are as above.

Additionally, it is noteworthy that we also adjusted the setting of air conditioning synchronously in the test so that the ambient temperature was always 2 °C higher than coolant.

1. The curves shown in Fig. 3 underline that the temperature of cooling sleeve rises with increasing the coolant temperature gradually. Evidently, higher operating temperature of cooling unit retards the convection heat transfer with spindle.

2. The temperature rise on cooling sleeve surface is nearly linear to the temperature change of cooling oil. This clearly indicates that the cooling system has an indispensable role in reducing the spindle temperature.

The thermal energy taken away by coolant is another cogent report to assess the cooling unit influence on spindle temperature. Hence, we also gauged the temperature of discharged cooling oil, and the corresponding changes are depicted in Fig. 4.

2.2 Regressive analysis

To closely examine the data shown in Fig. 2, one further finding is that the data on both sides of the 5th checkpoint are almost linear. Figure 5 presents the corresponding results of linearization after regressive analysis. It is obvious that the structural constraints outside coolant passages significantly impact the heat dissipation of spindles.
1. In the front half spindle (from the 1st checkpoint to 4th checkpoint), the slope changes of regression lines are very little under different coolant flow rates (0.024 for 8000 rpm, and 0.021 for 12000 rpm). Similarly, the regression lines are also almost parallel in the rear half spindle, and more specifically, the corresponding slope variations are 0.022 for 8000 rpm and 0.02 for 12000 rpm separately. This indicates that the heat transfer capacity between the spindle parts is almost not affected by the coolant flow rate.

2. There is a clear difference in the slopes of regression lines for the front and rear sections of the spindle, whose mean values are 0.339 and 0.336 for 8000 rpm and 0.32 and 0.247 for 12000 rpm, respectively. As a result, the intersections between regression lines, namely, the temperature peaks, appear at different positions when changing the operation conditions (see Figs. 7 and 8), but these positions are still very close to the 5th checkpoint. Evidently, the location of the spindle temperature peak depends on the structural constraints outside the cooling unit than operation conditions.

3. From Fig. 5, a further finding is that the slopes of regression lines for the front half spindle are closer than the rear half. To be specific, the variations are 0.022 at 8000 rpm and 0.02 at 12000 rpm for the front half spindle, and the corresponding values are 0.024 and 0.021 for the rear half spindle separately. Clearly, the structural constraints outside the front half of the spindle show more stable in affecting the spindle temperature field. And this also provides a theoretical basis for researchers to choose the first half of a spindle as the thermal analysis object.

3 A factor to assess the coolant unit effect on temperature

Based on the above regression analysis, one obvious conclusion is that the structural constraints outside a coolant passage significantly impact the heat dissipation of spindles. It is absolutely essential to create the detailed model of a whole spindle for discussing the heat transfer in depth. Considering the complexity of spindle structures, however, this will result in a complex model and then the data overhead increases greatly. For a simple and practical model, many papers aim at one end of the spindle; nevertheless, they ignore the substructures between the front and rear end of the spindle, which significantly impact the accuracy of thermal estimation. In this section, the “Sect. 3.1” is proposed, and next, the structural factor is constructed to locate the spindle temperature peak and describe the effect of the structural constraints on spindle temperature. And this work benefits simplifying the thermal estimation modeling.

3.1 Peak temperature surface

According to the above regression analysis, we find that all the temperature peats appear in almost the same area
with the coolant flow rate and spindle speed changed. For the spindle with a specific structure, quite evidently, the change of operation conditions has little effect on the maximum temperature area, which depends more on the spindle structure influence on heat transfer. From Fig. 5, this area is very narrow, approximately a plane. Therefore, we may use the plane \( x = L_p \) to represent the peak region, as seen in Fig. 6 and \( x = L_p \) can be determined by

\[
\begin{align*}
    y &= m_R x + b_R \\
    y &= m_F x + b_F
\end{align*}
\]  

(1)

where \( m_R \) and \( m_F \) are the mean values of first order terms for the front and rear halves of the spindle, and \( b_F \) and \( b_R \) represent the corresponding average values of constant terms, respectively.

Obviously, an indisputable fact is that the temperature is highest at \( x = L_p \), and so we may assume that heat can only be transferred from this plane to the front and rear ends of the spindle, namely the heat exchange is carried out independently on both sides of the plane \( x = L_p \). As a result, the spindle can be divided into two parts so as to simplify the thermal model.

### 3.2 A scaling factor

An accurate and fast thermal grid model is the key to evaluate the spindle temperature promptly and efficiently in engineering. The role of a coolant system on spindle heating, however, is not well characterized so far. In most cases, the established thermal network models of spindles are with no or only 50% of coolant units considered. Although some scholars modeled a whole coolant passage, the planned thermal network for a complete spindle is still too complex. So there is an urgent need for the accurate thermal resistance model to describe the coolant unit effect on spindle temperature.

In the above section, the peak temperature plane is proposed, based on regressive analysis. Clearly, this plane is closely related to the spindle structure. For one given spindle, the peak temperature area relying on structural constraints is certain. With the aid of the conclusions in Sect. 3.1, it is obvious that we can divide the spindle into two parts, and next, discuss their thermal performance separately. And this will greatly reduce the complexity of models. This section devotes to constructing a scaling factor to represent the place of peak temperature plane.

As seen in Fig. 6, a structure sketch of the motorized spindle is drawn, in which \( R_F \) and \( R_R \) are the structural constraints of front and rear spindle ends, and \( L_F \) and \( L_R \) denote the distances between the temperature peak plane \( x = L_p \) and the spindle ends separately. Based on \( x = L_p \), the cooling passageway is divided into two parts: \( L_{CF} \) and \( L_{CR} \). The stator and rotor of spindle motor, too, are divided into two parts.

To further describe the effect of coolant unit, stator, and rotor on the temperature field of spindle, Let us set the corresponding factors as follows

\[
\begin{align*}
    K_C &= \frac{L_{CF}}{L_{CF} + L_{CR}} \\
    K_S &= \frac{L_{SF}}{L_{SF} + L_{SR}}
\end{align*}
\]  

(2)

Considering the consistency in length, the stator and rotor have the same scaling ratio.

\[
K_R = K_S
\]  

(3)

Clearly, by introducing \( K_R \) and \( K_C \), we can consider the impact of coolant unit, stator, and rotor on the temperature of front/rear half spindles more accurately.

### 4 Application in thermal estimation of spindles

To create a more simple and efficient thermal estimation model with the proposed structure scaling factors is the end of this study. The validity of these coefficients, of course, also needs to be proved. For this reason, we need to select a
research object to build the corresponding thermal networks and subsequently determine it for contrast.

4.1 Thermal networks model with a cooling system

Based on the experimental findings in Sect. 2, the heat transfer model for the front half of the motorized spindle Setco231A240 was develop with the help of constructed scaling factors, as shown in Fig. 7, where the bearings’ substructures, contact between parts, spindle motor heating, and coolant and lubricant system are all taken into account. Some idealizations were also done to facilitate modeling.

1. The rotor and stator are both seen as cylinders; the structure after simplifying is as illustrated in Fig. 7.
2. For the surroundings outside a pair of front bearings, we referred to our previous work [30] and accordingly reduced them, namely they are predigested to be the locating ring, regulator, housing, and shaft, respectively.
3. In view of the abovementioned facts in Sect. 2, the coolant passage and motor armature are both divided into two parts with the help of the scaling factors, namely only the front halves of the rotor, stator, and cooling system are integrated into the proposed thermal grid when we predict the front-end temperature of the spindle.
4. The rotor heating is assumed to be a separate point thermal source, so is the power loss of bearing and stator. They are arranged separately.

In Fig. 7, the improved node arrangement for ball bearings in Ref. [30] is employed, and more explanation on the thermal resistance planning is as shown in Table 2.

In the meantime, the corresponding descriptions on resistance determination are also classified and further summarized in Tables 3 and 4, where the one-dimensional transfer within the parts themselves, the thermal contact transfer between parts, the heat exchange between various spindle parts and fluid media, etc. can be evaluated by the models given.

Besides, we also arrange $R_{c-1}$ and $R_{c-2}$ to assess the heat exchange between fluid media and cages, whose determination is presented in paper [30].

Furthermore, $R_c$ is planned to represent the convective resistance of cooling system, and $R_{r-s}$ characterizes the heat transfer capacity between rotor and stator. And the detailed description of these two thermal resistances is arranged in Sects. 4.2.2 and 4.2.3 separately.

![Fig. 7 Thermal model for the motorized spindle with a cooling unit](image-url)
### Table 2: Explanation on thermal resistance planning

| Planning | Description |
|----------|-------------|
| $T_a$    | Ambient temperature |
| $T_c$    | Oil-air temperature in bearing cavity |
| $R_{hitr}, R_{htr}, R_{htr}/R_{hitr}$ | Axial heat conduction for housing, shaft, and outer/inner ring |
| $R_{htr}, R_{st}, R_{st}/R_{htr}$ | Radial heat conduction for housing, shaft, and outer/inner ring |
| $R_{ct}, R_{ct}, R_{ct}/R_{ct}$ | Heat convection between outer/inner ring and oil-air |
| $R_{ht}, R_{ht}, R_{ht}/R_{ht}$ | Heat exchange between locating ring, regulator, balls, and oil-air |
| $R_{ht}, R_{ht}, R_{ht}/R_{ht}$ | Heat transfer by conduction for ball-cage |
| $R_{ht}, R_{ht}, R_{ht}/R_{ht}$ | Thermal contact transfer for ball-out/inner ring |
| $R_{ht}, R_{ht}, R_{ht}/R_{ht}$ | Heat exchange between housing, shaft, and ambient air |
| $R_{ht}, R_{ht}, R_{ht}/R_{ht}$ | Radial thermal contact between housing, shaft, and bearing |
| $R_{ht}, R_{ht}, R_{ht}/R_{ht}$ | Heat transfer for rotor-shaft and stator-housing |

### 4.2 Models for thermal discussion

#### 4.2.1 Heat sources

The heat of motorized spindle is composed of the bearings' energy consumption and the armature energy loss.

**Motor** The energy expenditure caused by spindle motor heating is mainly composed of copper loss, core loss, and mechanical loss, which can be expressed as

$$P_M = P_f + P_h + P_c + P_{cu} \quad (4)$$

If only the front half of a spindle is analyzed, the heat generation of spindle motor is calculated by

$$P_{MS} = K_f(P_f + P_h + P_c + P_{cu}) = K_f P_f + K_h P_h + K_c P_c + K_{cu} P_{cu} \quad (5)$$

Here, $P_f, P_h, P_c,$ and $P_{cu}$ denote the mechanical loss, hysteresis loss, eddy current loss, and copper loss, separately. They are estimated by [39]

### Table 3: Resistances on thermal conduction

| Thermal conduction description | Employed model | Applicable object | Reference |
|--------------------------------|----------------|------------------|-----------|
| Ball-raceway                   | $R = \frac{1}{\kappa} \left( \frac{d}{\rho} \right) \left( \frac{\delta}{\mu} \right)^{-1}$ | $R_{br}, R_{bo}$ | Muzychka and Yovanovitch [31] |
| Thermal contact transfer       | $R = \frac{1}{\kappa_{htr}} = \frac{1}{h_{htr}}$ | $R_{ht}, R_{ht}$ | Xu et al. [32] |
| Interference                   | $R = \frac{1}{\kappa_{htr}} = \frac{1}{2} \left( \frac{k_{ht}}{k_{ht}} + \frac{k_{ht}}{k_{ht}} \right)$ | $R_{ht}, R_{ht}$ | Xu et al. [32] |
| Transition                     | $R_{L} = L_s (A_{L} k_{D} + (1 - A_{L} + k_{D}))^{-1}$ | $R_{L}, R_{L}$ | Xu et al. [32] |
| One-dimensional transfer       | $R = \ln(d_{cm}/d_{min}) \left( 2 \pi k_{D} L \right)^{-1}$ | $R_{sh1}, R_{sh2}$ | Holman [33] |

where $\rho_{air}$ is the air density, $\omega_m$ is the spindle speed, $L$ and $R$ are the rotor length and diameter, $h$ is the protrusion of rotor end, $\delta_{air}$ is the air gap between rotor and stator, $R_s$ is the Reynolds number, $C_{ ls}$ is the constant relying on material properties, $B_{max}$ is the maximum of magnetic induction, $f$ is the magnetic field frequency, $v_s$ and $\delta_s$ are the volume and thickness of silicon steel, $\rho_{is}$ is the resistivity, and $I_L$ and $R_L$ represent the current and resistance of stator/rotor winding, separately.

**Bearing** In assessing the energy loss of rolling bearings, the experimental models are first employed, for example the best-known Palmgren’s formulas [40]. On this basis, many improved estimation was proposed successively. Despite
some enhancement of the forecast accuracy, more parameters are integrated, which makes the evaluation of heat generation more difficult. In this discussion, we still referred to Palmgren’s findings to calculate the power loss of bearings. The energy loss caused by the spin motion of bearing balls is detailed in Table 5.

\[ W_T = W_r + W_F + W_s \] (7)

Here, \( W_T \) is the total friction heat, and how to determine the sub-sources \( W_r, W_F, \) and \( W_s \) is detailed in Table 5.

### 4.2.2 Heat exchange capacity about the cooling channel of a spindle

The forced convection through cooling channels is an efficient way to dissipate heat. In the light of our latest work [29], the coolant passage can be regarded as a tube with the inner diameter \( d_i \). Referring to the relevant findings summarized in the book [41], the corresponding thermal resistance to describe the heat exchange capacity of cooling passageways is formulated as below.

\[ R_{th} = \frac{1}{A h_v} = \frac{1}{A k_D N_u} \] (8)

where \( h_v \) is the convection heat transfer coefficient, \( k_D \) is the thermal conductivity of coolant, and \( A \) represents the inner area of passage.

\[ A = \pi^2 d_i D_i n_i \] (9)

Here, \( D_i \) is the helix diameter of the spiral cooling channel, and \( n_i \) denotes the number of helix turns.

### Table 5 Sub-sources of bearing heating

| Sub-source               | Employed model                                                                 | Applicable object                                                                 | Reference |
|-------------------------|------------------------------------------------------------------------------|----------------------------------------------------------------------------------|-----------|
| \( W_F \)               | \( f_0 f_1 P \cdot d_m \cdot \alpha_r d(t) \)                                | The friction heat induced by the radial and axial loads                           | Palmgren  [40] |
| \( W_r \)               | \( \left\{ \begin{array}{l} W_r = f_0 f_1 M_r \cdot \alpha_r d(t) \\ M_r = f_0 (v \cdot n)^3 d_m^2 \end{array} \right. \) |
|                         | \( v \cdot n \geq 2000 \) \hspace{1cm} \( M_r = 160 f_0 d_m^2 / 1000 \) \hspace{1cm} \( v \cdot n < 2000 \) |
| \( W_s \)               | \( Z_s \cdot (6 \alpha Q_{oil-p} \cdot \xi + \frac{6 \alpha Q_{oil-air} \cdot \Sigma_s}{8}) \) | The energy loss caused by the spin motion of bearing balls                          |           |
In addition, $N_u$ in Eq. (8) is the Nusselt number to describe the heat transfer capacity between coolant and passageway, which can be determined through introducing the Reynolds number $R_r$ and the Prandtl number $P_r$ [42].

$$N_u = 0.023R_e^{0.8}P_r^{0.4} = 0.023(\rho_Vd_l/\eta_l)^{0.8}(\eta_l/(\rho \alpha_{dl}))^{0.4}$$

(10)

where $\eta_l$, $\alpha_{dl}$, and $V_l$ are the viscosity, diffusion coefficient, and circuit flow rate of coolant separately.

So,

$$R_{th} = \frac{1}{A} \left( \frac{d_l}{k_D 0.023(\rho_Vd_l/\eta_l)^{0.8}(\eta_l/(\rho \alpha_{dl}))^{0.4}} \right)$$

$$= 43.478 \times \frac{d_l^{0.2}}{A} \frac{1}{k_D(\rho_l/(\eta_l \alpha_{dl}))^{0.4}V_l^{0.8}} = \frac{43.478}{\pi^2 \zeta}$$

(11)

Here,

$$\xi = \frac{1}{k_D(\rho_l/(\eta_l \alpha_{dl}))^{0.4}V_l^{0.8}}$$

$$\zeta = d_l^{0.2}D_r \eta_l$$

(12)

Obviously, $\xi$ relies on the operation conditions of a cooling unit. On the contrary, $\zeta$ is more closely related to the structural parameters.

Based on Sect. 3, the structural scaling factor $K_C$ is employed to estimate the corresponding influence of coolant passageways on the heat exchange of the front halves of spindles exactly.

$$R_c = K_C \cdot R_{th} = \frac{43.478K_CU}{\pi^2 d_l^{0.8}D_r \eta_l}$$

(13)

### 4.2.3 Heat exchange between rotor and stator

Rotor heating accounting for about 1/3 of total energy loss of a spindle motor is transferred through the air. This heat exchange can be regarded as a free convection in the confined space. Paper [35] provides a simplified model to calculate the heat transfer capacity.

$$R_{Mr-\rightarrow} = \frac{1}{28A(1 + \sqrt{0.5} V_c)}$$

(14)

where $V_c$ is the linear velocity of rotor, and $A$ and $R_{Mr-\rightarrow}$ represent the heat convection area and heat transfer resistance between rotors and stators.

Certainly, $R_{Mr-\rightarrow}$ is also separated into two parts by the temperature peak plane. For the front half spindle, the scaling factor $K_S$ can be used to determine the corresponding thermal resistance.

$$R_{Mr-\rightarrow} = \frac{K_S}{28A(1 + \sqrt{0.5} V_c)}$$

(15)

### 4.3 Simplification and solution

As mentioned, the developed thermal grid model seems to be more complex because the cooling and lubricant units are considered and the substructures and fits between parts as well, while the complexity of a network depends more on the node number, especially the nodes with more than two inputs. Further analysis shows that there are only 16 such key nodes, as you can see the marked $N_{key1}$–$N_{key16}$ in Fig. 7. Based on the series and parallel theory of Ohm’s law, the resistances between two key nodes, moreover, can be reduced as an equivalent thermal impedance. The simplified model, as shown in Fig. 8, is not complicated.

In this discussion, the corresponding heat flux model of each key node in Fig. 8 can be first built, and next, we can establish a heat balance equation group for the whole thermal grid as below.

$$\sum_{i=1}^{N} \sum_{j=1,j \neq i}^{N} (T_i - T_j)R_{ij}^{-1} + \sum_{i=1}^{N} \dot{Q}_i = 0$$

(16)

where $R_{ij}$, $\dot{Q}_i$, and $T_i$ denote the resistance, heat generation capacity, and node temperature, respectively.

Using the popular Newton–Raphson method, the node temperature can be easy to obtain and then can assist us to verify the constructed model and structural scaling factors. And this also is of great benefit for further exploring the thermal exchange of spindle.

### 5 Results and discussion

To validate the planned thermal forecasting model for the motorized spindle and the proposed scaling factor to describe how the coolant unit influences the heat transfer of front/rear half spindle as well, a comparison with simulation results is needed. In contrast to the surface temperature of spindle housing, the heating of spindle bearings can better report the temperature variation and even the thermal error of spindles, so the outer ring temperature of front bearing 7009AC was selected as a reference. The testing scheme is the same as illustrated in Fig. 1, and the experimental conditions under various operation settings of oil cooler are as mentioned in Sect. 2.2. The corresponding measurements are separately presented in Figs. 9 and 10.
The thermal network model, equation group, and iterative algorithm described in Sect. 4, meanwhile, were employed to obtain the numerical solution of outer ring temperature when the coolant temperature was changed. The contrast with the experimental results is described in Fig. 10. The similar work for coolant flow flux was also carried out, and the corresponding results from experiments and simulation are illustrated in Fig. 9.

It has been demonstrated by a number of studies that the rotation rate has a more visible effect on bearing heating than lubricant and air flow as well. Therefore, we measured the thermal growth with the increased speed to further verify the proposed model, too. At the same time, we solved the developed thermal grid of the front half spindle with a scaling factor to obtain the forecast temperature of bearing outer ring. The simulation work for the spindles with no and 50% of a cooling unit, for further validation, was also done separately. The contrast between the test values and the simulation results based on Matlab is shown in Fig. 11.

Note that the assumptions for numerical results are as follows:

1. The temperature variation is linear from the inlet to the outlet of coolant passage.
2. Given the spindle bearing arrangement, we believe both the applied axial load and the thermal expansion-induced preload in operation are equally distributed to each bearing.
3. The temperature of discharged oil-air is used to represent the atmosphere in spindle cavity.

From Figs. 9, 10 and 11, there is an obvious good agreement between test values and forecasting results, and this validates the simplified thermal network with the structural scaling factors for cooling system and motor armature. Compared with current works, a more accurate estimation of real spindle temperature can be carried out by employing the developed model for motored spindles.

**Fig. 8** Simplified thermal grid by series and parallel theory

**Fig. 9** Contrast on bearing heating under different coolant flow rates
1. Just like the numerical results in bearing temperature, the test results (see Fig. 9) also decrease when raising the cooling oil flow rate. Obviously, more heat can be taken away with more coolant supply. But both experimental and numerical curves indicate that there is no same level in temperature reduction (no more than 2.03 °C at 8000 rpm and 1.98 °C at 12000 rpm) when the coolant flow rate is raised from 2.5 to 4.5L/min. It is obvious that a reasonable coolant flux is needed when the energy efficiency of oil cooler is factored.

2. From Fig. 10, the temperature of discharged cooling oil has a very small rise (about 0.36 °C for 8000 rpm and 0.52 °C for 12000 rpm) with the decrease of oil cooler operating temperature. The variation, just liking the depictions of Fig. 4, is almost linear. Clearly, the operating temperature of oil cooler only has a little effect in cooling spindle. As a result, to set the oil cooler operating at 2 °C lower than room temperature may be a more cost-effective way.

3. From Fig. 11, the deviations between simulation results and test values is about 1.2 °C when the scaling factor is introduced to model the spindle, and the prediction differences for the current models with no and 50% of the coolant passage integrated are more than 3 °C and 2 °C, respectively, namely the corresponding forecasting errors are close to 10% and 6%. By our developed model, this difference is reduced to less than 4%. Evidently, the prediction accuracy can be greatly improved when we refer to the structural scaling factor to factor the coolant system effect.

To a certain extent, the temperature forecasting values based on simulation still deviate from the test results, however. Besides the prediction error mentioned above under different spindle speeds, the deviation, for varied coolant flux, is near to 1.1 °C when the spindle operates at 8000 rpm, and this maximum is 0.96 °C at 12000 rpm. Correspondingly, the error approaches 0.3 °C for 8000 rpm and 0.5 °C for 12000 rpm with coolant temperature though these values are not large.

1. The various sub-heat sources of bearings, as we all know, are induced by different mechanisms; in addition, they are also scattered at different locations of bearings. Yet, in this paper, the bearing heat was determined by the global model and then was assumed to be a point placed in ball center. Additionally, the scaling factors were constructed to consider the effect of structural constraints on heat transfer, but both stator and rotor were still reduced to point heat sources. Compared with the real heat generation of motorized spindles, there is a certain gap.

2. To create an efficient engineering model, some idealizations for spindle substructures were done. But the corresponding heat transfer is also dismissed in these simplifications. And this is another contribution to the deviation.

3. As is known to all, the heat convection between spindle and air/liquid under vibration will be different from no vibration, so is the contact state between parts. These changes can affect the thermal contact and convection resistances. Obviously, neglect of the heat transfer change under vibration in our temperature assessment cannot match facts.

4. The coolant can absorb and then take away a great quantity of heat by heat exchange, and this also results in a temperature rise of discharged cooling oil. In our simulation, the average value of the inlet and outlet temperatures of coolant was applied to calculate the thermal equivalent convection in coolant passage. On the contrary, the temperature field of helical passage in spindles is not uniform. This linearization may further raise the difference.
6 Conclusions

The temperature evaluation is the first stage to characterize the thermal behavior of a spindle. Although many models have been available in assessing the spindle temperature field, an accurate, lightweight, and fast (low overhead/small footprint) forecasting is not addressed in engineering till now. Apparently, the insufficient investigation for the effect of coolant systems on spindle temperature may partly contribute to this situation. In this paper, extensive experiments on the spindle temperature with various coolant parameter settings were first carried out. On the basis of the qualitative and quantitative analysis and regressive analysis on these test data, the temperature trend of spindle surface came to light. As a result, the peak temperature area was found, and then a scaling factor was proposed to divide the spindle into front and rear sections. At the same time, the structural coefficient for stator and rotor was constructed too. With the help of these proposed factors, we created the thermal grid for the front half of the motorized spindle with a cooling passage. For a better contrast, the current thermal network models with no and only 50% of the coolant unit were also simulated separately. The comparison result verifies the availability of the constructed coefficients and models.

Compared with current thermal grid models, the novel temperature estimation with the scaling factors introduced is obviously closer to the test values. Evidently, with the help of these coefficients, we can assess the effect of coolant unit and motor armature heating on spindle temperature more accurately. Furthermore, these proposed factors can also be used in the finite element analysis of spindles, by which only the finite element models for the front/rear halves of spindles need to be created. In contrast with the presented finite element modeling for a complete spindle, this can present a more simple model with lower data overhead, and thus the efficiency of finite element analysis can be greatly improved. This work, meanwhile, is beneficial in evaluating the accuracy decay and even operation life of motorized spindles, and also provides the possibility for the active adjustment of spindle temperature and even thermal error.

Author contribution Zheng De-xing is the main contributor to this paper. He implemented the experimental studies and next proposed the structural scaling factor to describe the role of a coolant unit in spindle temperature. This factor was also integrated into the developed thermal grid model for the front half of motorized spindle. Meanwhile, he validated the proposed structural scaling factor by comparing the simulation with test results. Chen Weifang gave some valuable comments in improving the technical route of this work.

Funding This work was funded by the Natural Science Foundation of Jiangxi Province, China (no. 20212BAB204033), the National Natural Science Foundation of China (no. 51775277), and the National Science and Technology Major Project of the Ministry of Science and Technology of China (no. 2015ZX0401002).

Availability of data and material Data are available.

Declarations

Conflict of interest The authors declare no competing interests.

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