Research on Thermal Capacity of a High-Torque-Density Direct Drive Permanent Magnet Synchronous Machine Based on a Temperature Cycling Module

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ABSTRACT It is difficult to meet the engineering requirements for a high-torque-density direct drive (HTDDD) permanent magnet synchronous machine (PMSM) using the thermal load design method for traditional motors. In this study, a waterway structure design method for a HTDDD PMSM depending on the motor thermal capacity is proposed based on a temperature cycling module. The design program of the motor waterway structure is based on a waterway thermal calculation and an equivalent thermal network method. The waterway structure can be rapidly designed by cyclic checking of the temperature at each node in the motor. A fluid-solid coupled temperature field simulation is used to analyze the influences of the thermal capacity, the waterway structure, and the water velocity on the temperature rise and the cooling effect of the motor. Finally, the algorithm is used to create a prototype, and the proposed method is experimentally validated.

INDEX TERMS Direct drive permanent magnet synchronous machine, thermal capacity, waterway structure, equivalent thermal network, fluid-solid coupled temperature field.

I. INTRODUCTION
A high-torque-density direct drive permanent magnet synchronous machine (HTDDD PMSM) combines the functions of speed reduction and torque amplification of a reducer into a single motor to directly drive a load. This machine is characterized by a low speed, a high torque density and a high efficiency and is widely used in the coal mine industry [1], [2]. However, the continuous working system of the HTDDD PMSM makes heat dissipation difficult in the high ambient temperature of the mine. The motor temperature can easily become excessively high and burn the motor. In severe cases, irreversible demagnetization of the permanent magnet compromises safe motor operation. The HTDDD PMSM thermal capacity is defined as the electromagnetic load on the armature unit volume. The direct drive permanent magnet synchronous machine (PMSM) for a belt conveyor with the same power level is characterized by a high torque density but has a larger volume than and a different selection of the thermal capacity from the traditional motor in the original system. Therefore, the appropriate selection of the thermal capacity of the HTDDD PMSM for a belt conveyor and determining the temperature rise and cooling effect of the motor to improve the heat dissipation capacity is extremely important for the safe operation of a HTDDD PMSM for a belt conveyor.

In recent years, the design of the motor cooling structure and the calculation of the temperature field have become an active research area in the field of motors. For example, Joo et al. [3] in Korea in 2011 used the finite element method (FEM) to calculate the temperature field distribution of each component of a 7.5-kW air-cooled interior PMSM for an agricultural electric vehicle. The results of the 3D numerical analysis were in good agreement with experimental results. In 2015, Zhao et al. [4] at the Xi’an University of Architecture and Technology used a lumped-circuit thermal analysis to comparatively analyze a surface-mounted permanent magnet (SPM) motor and a hybrid magnetization interior permanent magnet (IPM) motor, showing that the IPM motor had the higher gross loss, whereas the SPM motor had the...
higher rotor temperature rise. In 2016, Dong et al. [5] at Southeast University used lumped parameter thermal networks to compare the thermal performances of SPM and IPM motors for high-speed applications. The results showed that the SPM motor was cooler with a higher overall efficiency. In 2017, Fasquelle et al. [6] in France implemented cold plates in the magnetic core for a 3-MW permanent magnet synchronous motor: the thermal performance of the cold plate system was experimentally determined, and computational fluid dynamics (CFD) and FEM were used to analyze the flow structure and its effect on the thermal properties of the cold plates. In 2018, Zhu et al. [7] at the Shenyang University of Technology used a double-circulatory technique in conjunction with a two-way coupling of physical parameters to determine the temperature distribution in a permanent magnet traction motor. The fluid temperature and the heat convection coefficients were iterated in the inner loop of the procedure. The outer loop was used to modify the heat convection coefficients and the electromagnetic losses. A modified cell method thermal model was used to reduce the iterative computation. In 2019, Xia et al. [8] at the Harbin University of Science constructed a wind resistance network model to investigate the transient fluid flow and temperature rise in a 2500-kW high-power-density and high-voltage induction motor during starting. The results showed that a low fluid flow and starting current result in a rapid rise in the winding temperature. The winding temperature during starting also rises rapidly under high loads. Aldo Boglietti et al. [9] established a lumped parameter thermal model for a wound-rotor induction machine, which was validated in a 3.8-hp wound rotor machine experiment. Lu et al. [10] at Zhejiang University developed a finite element model and a thermal network model to obtain the temperature distribution of a water-cooled permanent magnet linear motor (PMLM), which was validated by experimental results. Jungreuthmayer et al. [11] established a comprehensive CFD model of a radial flux permanent magnet motor, performed a heat flow analysis of the machine, and used a simple fan with rectangular blades to increase heat motion in the machine. Tong et al. [12] at the Shenyang University of Technology used Fluent software to simulate the temperature rise distribution of a high-speed surface-mounted PMSM, which was verified by comparison with results from tests on the thermal performance of the prototype. Most of the papers use the equivalent thermal network method and FEM to calculate the temperature rise of induction motor and conventional PMSM, and their cooling structures are more conventional. For most motors, the traditional thermal load design method is combined with a software simulation to verify the temperature rise [13], [14]. However, it is difficult to meet engineering requirements for the HTDD PMSM using the traditional thermal load design method. At present, there are few researches on the waterway structure design and thermal capacity of HTDD PMSM.

In this study, the relationship between the thermal capacity and temperature rise of a direct drive PMSM is used in a temperature cycling module to develop a water structure design method for the HTDD PMSM that satisfies the project requirements of a direct drive belt conveyor of a PMSM in the Wang Lou coal mine. A fluid-solid coupled temperature field simulation is used to analyze the influences of the thermal capacity, the waterway structure, and the water velocity on the temperature rise and cooling effect of the motor. The algorithm proposed in this paper is used to develop a direct drive PMSM for the 200-kW belt conveyor according to the project requirements. The waterway structure design method for the motor based on the temperature cycling module is verified experimentally, and provides a reference for research on the thermal capacity of a HTDD PMSM.

II. STRUCTURE AND WATERWAY CALCULATION

A HTDD PMSM used for belt conveyors combines components for torque conversion and torque transmission to directly drive a load. The operating condition and structure of a HTDDD are shown in Fig. 1. The HTDDD PMSM is directly connected with the roller of the belt conveyor and eliminates the gear reducer and other transmission components, which makes the transmission mechanism more compact, while improving the efficiency of the drive system.

A multipole structure is used in a HTDDD PMSM to satisfy a low-speed and high-torque requirement of several or even dozens of revolutions per minute. A large number of poles can be used to reduce the yoke height of the stator and rotor and the motor volume, while increasing the effective diameter of the motor, the utilization ratio of the ferromagnetic material, and the torque density. Each coil is wound on multiple teeth and the ends of each coil are overlapped in a traditional distributed winding motor. The large coil end results in a large copper loss. The slot utilization rate in a traditional distributed winding motor is relatively low. Therefore, a fractional slot concentrated winding is used in the HTDDD PMSM. Each coil is wound on only one tooth, the coil end is short and the copper loss is reduced. It effectively reduces motor size and improves motor efficiency and power density. The magnetic pole is created by punching a 0.5-mm silicon steel sheet; the magnetic pole is fixed on the rotor sleeve, and a permanent magnet is inserted between the adjacent magnetic poles; the sectional structure is shown in Fig. 2. The reinforced finned axial tandem waterway structure is used in the HTDD PMSM. The angle steel is arranged between the base barrel and the frame plate, which greatly improves the
FIGURE 2. Sectional structure of HTDDD PMSM used for belt conveyor:
(a) structure of HTDDD PMSM; (b) reinforced finned waterway; and (c) angle steel: a-average width of the cooling channel in a angle steel, b-height of the cooling channel, and m-thickness of angle steel fin.

FIGURE 3. Equivalent thermal network model.

In the equation above, $Nu_f$ is the Nusselt number of the water; $\lambda_w$ is the thermal conductivity of water; and $D_e$ is the equivalent diameter of the cooling channel.

The heat absorbed by the cooling water is expressed as

$$\Phi = \alpha_f \cdot S \cdot \Delta T_f.$$  \hfill (5)

In the equation above, $S$ is the area of the cooling wall; and $\Delta T_f$ is the temperature difference between the cooling water and the cooling wall.

The pressure difference between the inlet and outlet is expressed as follows:

$$\Delta p = \rho g h + \rho g (h_j + h_w).$$ \hfill (6)

In the equation above, $g$ is the gravitational acceleration; $h$ is the height difference between the inlet and outlet; $h_j$ is the resistance along the path; and $h_w$ is the local resistance.

III. TEMPERATURE CYCLING MODULE

A. EQUIVALENT THERMAL NETWORK MODEL

The equivalent thermal network method is to discretize the motor according to different regions of different materials. It converts the temperature field of the motor into equivalent thermal network using the concept of centralized parameter. An orthogonal grid is used to divide each component of the HTDDD PMSM into multiple regions. The center of each region is regarded as a temperature node. These nodes are subsequently connected by conduction and convection thermal resistances. By solving each thermal parameter and calculating the thermal balance equation of each node, the temperature distribution of each component of the HTDDD PMSM can be obtained. The calculation results for the waterway structure are used to establish an equivalent thermal network model for the HTDDD PMSM, as displayed in Fig. 3.

In Fig. 3, 0 represents an ambient temperature node; 1 and 2 represent the front and rear air nodes, respectively, in the motor chamber; 3-5 represent water-cooled frame nodes; 6-8 represent stator yoke nodes; 9-13 represent winding nodes, of which 9 and 13 are end winding nodes; 14-16 represent stator tooth nodes; 17-19 represent rotor magnetic pole nodes; 20-22 represent permanent magnet nodes; 23-25 represent rotor sleeve nodes; 26-29 represent...
the front and rear end covers of frame nodes; and 30 and 31 represent front and rear bearing nodes, respectively.

The heat sources in the motor are distributed as follows. The stator yoke and tooth iron core losses are equably distributed over nodes 6-8 and 14-16; the copper loss is equably distributed over nodes 9-13; the mechanical loss is equably distributed over nodes 30 and 31; and the stray loss is equably distributed over nodes 14-16 and 17-19.

### B. HEAT SOURCES

The various losses incurred during the conversion of electromechanical energy are mainly distributed in the form of heat in the motor, which generates a temperature rise. A nonuniform air gap is used for the rotor magnetic pole to reduce the harmonic content of the air gap magnetic density, it reduce the torque ripple and has no unbalanced electromagnetic force. The rotor magnetic pole has a laminated structure, and the eddy current loss from the space harmonic on the rotor magnetic pole is negligible. In addition, the built-in components magnet, the sleeve, the front air in the motor chamber, the rotor magnetic pole, the permanent magnet and the sleeve are presented here.

1) THERMAL RESISTANCE OF WATER-COOLED FRAME

We consider the water-cooled frame node 3 as an example. There are three paths for heat transfer. These paths correspond to the temperature boundary, internal heat conduction within the frame, and the stator yoke. The calculation process is given below.

The thermal resistance between nodes 3 and 0 can be expressed as follows:

\[
R_{30} = \frac{h_f/2}{\lambda_f S_{30a}} + \frac{1}{\alpha_f S_{30c}} 
\]

(10)

\[
S_{30a} = \pi D_f L/3 
\]

(11)

\[
S_{30c} = CL/3. 
\]

(12)

In the equation above, \( D_f \) is the outer frame diameter; \( h_f \) is the frame thickness; \( L \) is the axial length of the stator; \( C \) is the wetted perimeter of the water channel; \( \lambda_f \) is the heat conductivity coefficient of the water-cooled frame; \( S_{30a} \) is the radial heat conduction area of the frame; and \( S_{30c} \) is the convective heat dissipation area between the water and the frame.

The thermal resistance between nodes 3 and 4 can be expressed as

\[
R_{34} = \frac{L_f/3}{\lambda_f S_{34}}. 
\]

(13)

In the equation above, \( L_f \) is the frame axial length; and \( S_{34} \) is the circumferential cross-sectional area of the frame.

The heat is transferred from node 6 to node 3 through the thermal resistance of the assembly gap between the frame and the stator. The thermal resistance between nodes 3 and 6 can be expressed as follows:

\[
R_{36} = \frac{h_c/2}{\lambda_c S_{36c}} + \frac{L_{fr-st}}{\lambda_o S_{36}} + \frac{h_f/2}{\lambda_f S_{36}} 
\]

(14)

\[
S_{36c} = \pi (D_1 - h_c/2) L/3 
\]

(15)

\[
S_{36} = \pi D_1 L/3. 
\]

(16)

In the equation above, \( D_1 \) is the stator outer diameter; \( h_c \) is the stator core yoke thickness; \( \lambda_c \) is the radial heat conductivity coefficient of the silicon steel sheet; \( \lambda_o \) is the heat conductivity coefficient of air; \( L_{fr-st} \) is the mounting clearance between the frame and the stator; \( S_{36c} \) is the average heat dissipation area of the stator yoke; and \( S_{36} \) is the heat dissipation area between the frame and the stator.

2) THERMAL RESISTANCE OF ROTOR MAGNETIC POLE

The rotor is mainly composed of a rotor magnetic pole, a permanent magnet and a rotor sleeve, as shown in Fig. 4. We consider rotor magnetic pole node 17 as an example. There are five heat transfer paths. Heat is transferred through the air gap between the rotor and the stator, and by the permanent magnet, the sleeve, the front air in the motor chamber, and as internal heat conduction within the magnetic pole. The calculation process is given below.
In the equations above, \( m \) is the number of pole pairs; \( S_{1714} \) is the radial heat dissipation area of a magnetic pole; \( S_{g1714} \) is the heat dissipation area between a magnetic pole and the air gap; and \( \alpha_g \) is the heat dissipation coefficient of the air gap calculated using the Taylor number [22]. We define
\[
\alpha_g = \frac{Nu_\alpha \lambda_S}{g}. \tag{20}
\]

In the equation above, \( g \) is the size of the gap between the rotor and the stator; and \( Nu_\alpha \) is the Nusselt number of air. We also define
\[
T_a = \frac{0.5 \rho \omega^{1.5} n}{\nu_a}. \tag{21}
\]

In the equation above, \( \rho_a = (D_{i1} + D_2) / 4 \), where \( D_{i1} \) is the stator internal diameter; \( n \) is the motor rated speed; and \( \nu_a \) is the kinematic viscosity of air.

For laminar flow \((T_a > 41.19)\),
\[
Nu_\alpha = 0.128 \left( T_a^2 / F_g^2 \right)^{0.367}. \tag{22}
\]

For turbulent flow \((T_a < 41.19)\),
\[
Nu_\alpha = 0.409 \left( T_a^2 / F_g^2 \right)^{0.241}. \tag{23}
\]

In the equation above, \( F_g \) is a geometrical factor.

The thermal resistance between nodes 17 and 18 can be written as follows:
\[
R_{1718} = \frac{L/3}{\lambda_{co} S_{1718}} \tag{24}
\]
\[
S_{1718} = \pi \left( D_{i2}^2 - D_{i2}^2 \right) / 4 - 2p(eH + h_m b_m). \tag{25}
\]

In the equation above, \( S_{1718} \) is the heat dissipation area of the magnetic pole; and \( \lambda_{co} \) is the axial heat conductivity of the silicon steel sheet.

The heat is transferred from node 17 to node 20 through thermal resistance of the assembly gap between the rotor magnetic pole and the permanent magnet. The thermal resistance between nodes 17 and 20 can be written as follows:
\[
R_{1720} = \frac{b_m}{\lambda_{pm} S_{1720}} + \frac{2L_{ca}}{\lambda_a S_{1720}} + \frac{Q_1}{\lambda_S S_{1720}} \tag{26}
\]
\[
S_{1720} = 2p_{bm} L/3. \tag{27}
\]

The thermal resistance between nodes 17 and 23 can be written as follows:
\[
R_{1723} = \frac{(D_2 - D_{i2}) / 4}{\lambda_S S_{1723}} + \frac{(D_2 - D_{i1}) / 4}{\lambda_{sl} S_{1723}} \tag{28}
\]
\[
S_{1723} = (\pi D_{i2} - 2p_{bm}) L/3. \tag{29}
\]

In the equation above, \( \lambda_{sl} \) is the heat conductivity coefficient of the sleeve; and \( S_{1723} \) is the heat dissipation area between the rotor magnetic pole and the sleeve.

The thermal resistance between nodes 17 and 1 can be written as
\[
R_{171} = \frac{L/6}{\lambda_{co} S_{1718}} + \frac{1}{\alpha, S_{1718}}. \tag{30}
\]

In the equation above, \( \alpha_r \) is the heat dissipation coefficient of the rotor magnetic pole end face, as defined in [23]:
\[
\begin{align*}
\alpha_r &= \frac{Nu_r \lambda_a}{D_2 / 2} \\
Nu_r &= 1.67 Re_r^{0.385} \\
Re_r &= \frac{\pi D_2^2 n}{120 \nu_a}.
\end{align*} \tag{31}
\]

In the equation above, \( Nu_r \) is the Nusselt number of air at the rotor magnetic pole end face; and \( Re_r \) is the Reynolds number of air.

3) THERMAL RESISTANCE OF PERMANENT MAGNET

We consider permanent magnet node 20 as an example. The heat transfer paths are similar to those for the magnetic pole. The calculation process is as follows.

The thermal resistance between nodes 20 and 14 can be calculated as
\[
R_{2014} = \frac{(D_2 - D_{i2}) / 4}{\lambda_{pm} (2p_{bm} L/3)} + \frac{1}{\alpha_g (2p e L/3)}. \tag{32}
\]

The thermal resistance between nodes 20 and 21 can be calculated as
\[
R_{2021} = \frac{L/3}{\lambda_{pm} (2p_{bm} b_{km})}. \tag{33}
\]

The thermal resistance between nodes 20 and 23 can be calculated as
\[
R_{2023} = \frac{(D_2 - D_{i2}) / 4}{\lambda_{pm} (2p_{bm} L/3)} + \frac{(D_2 - D_{i1}) / 4}{\lambda_{sl} (2p_{bm} L/3)}. \tag{34}
\]

The thermal resistance between nodes 20 and 1 can be calculated as
\[
R_{201} = \frac{L/6}{\lambda_{pm} (2p_{bm} b_{km})} + \frac{1}{\alpha_r (2p_{bm} b_{km})}. \tag{35}
\]
4) THERMAL RESISTANCE OF ROTOR SLEEVE
We consider rotor sleeve node 23 as an example. There are four heat transfer paths. Heat is transferred through the rotor magnetic pole, the permanent magnet and the front air in the motor chamber, and as internal heat conduction within the sleeve. The calculation process is given below.

The thermal resistance between nodes 23 and 24 can be calculated as

\[ R_{2324} = \frac{L/3}{\lambda_{sl} S_{2324}}. \]  

(36)

In the equation above, \( S_{2324} \) is the radial cross-sectional area of the rotor sleeve. This area is given by

\[ S_{2324} = \pi \left( D_{12}^2 - D_1^2 \right) / 4. \]  

(37)

The thermal resistance between nodes 23 and 1 can be calculated as

\[ R_{231} = \frac{L/6}{\lambda_{sl} S_{231}} + \frac{1}{\alpha_{sl} S_{231}}. \]  

(38)

In the equation above, \( \alpha_{sl} \) is the heat dissipation coefficient of the sleeve end face, as defined in [24]:

\[
\begin{align*}
\alpha_{sl} &= \frac{Nu_{sl} \lambda_{sl}}{D_{12}/2}, \\
Nu_{sl} &= 0.456 Re_{sl}^{0.385}, \\
Re_{sl} &= \frac{\pi D_a D_n}{120 \nu a}.
\end{align*}
\]  

(39)

In the equation above, \( Nu_{sl} \) is the Nusselt number for air at the end face of the rotor sleeve; and \( Re_{sl} \) is the Reynolds number for air.

D. HEAT CONDUCTION EQUATION
A thermal balance equation is derived for each node of the HTDDD PMSM. The heat balance equation for nodes 1-31 is given by

\[-T + \sum G(i,j) T = W(i).\]  

(40)

In the equation above, \( T(i) \) is the temperature of the \( i \)th node; \( W(i) \) is the thermal source of the \( i \)th node; \( G(i,j) \) is the self-heat conductance of the \( i \)th node; and \( G(i,j) \) is the mutual heat conductance between the \( i \)th and \( j \)th nodes. \( R(i,j) \) is the thermal resistance between the \( i \)th and \( j \)th nodes.

E. SOLUTION RESULTS BY MATLAB
The heat dissipation capacity of the waterway is directly reflected in the temperature rise of the motor. Therefore, we calculate the waterway structure of the HTDDD PMSM by writing a program that incorporates the temperature cycle module of the equivalent heat network method. The calculation results of the temperature cycling module are used to rapidly calculate the waterway structure of the HTDDD PMSM. The flow chart for this calculation in MATLAB is shown in Fig. 5. The cycle is terminated when the module calculation results meet the requirements for the motor temperature rise. Thus, using this waterway structure in the HTDDD PMSM meets cooling requirements.

Table 1 shows the temperatures for the HTDDD PMSM obtained by the thermal network method. The winding temperature of 61.1°C and the average permanent magnet temperature of 65.6°C do not exceed the limit temperature and meet the requirements for a class B insulation temperature rise. The temperature cycle module is therefore terminated, and the results of the waterway structural program are shown in Table 2. This reinforced finned waterway structure is used for the HTDDD PMSM.

IV. FLUID-SOLID COUPLING
A. BASIC ASSUMPTIONS
The following assumptions are used to obtain a physical model in the solution domain [25].

(1) All the insulation in the stator slots is replaced by equivalent insulation, and the double-layer winding and the slot insulation have the same interlayer insulation thickness.
(2) The skin effect of the winding is neglected. The insulation paint is evenly distributed over the winding, and the upper and lower winding have the same heat loss.

(3) Heat radiation from the motor has a negligible effect on the ambient temperature. Therefore, the ambient temperature is considered to be constant.

(4) Cooling water enters the water channel perpendicular to the inlet.

Fig. 6 shows the physical model in the solution domain for the HTDDD PMSM obtained using the assumptions above.

B. BOUNDARY CONDITIONS
The boundary conditions are determined by requiring that the governing equation is satisfied at the boundaries of the fluid motion. The boundaries include the entrance, the exit and the wall [26]. The boundary conditions for the HTDDD PMSM are given below [27],[28].

(1) The inlet surface of the frame is set as the speed inlet, where the inlet velocity is 1.77 m/s based on the calculated cooling water inlet velocity.

(2) The outlet surface is set as the pressure outlet, and the outlet pressure is standard atmospheric pressure.

(3) No-slip boundary conditions are placed at all the outer wall surfaces, and the effect of the surface air velocity on heat dissipation at these surfaces is considered to be negligible.

(4) The rotor surface is set as the rotating boundary condition, where the speed is set to 60 r/min.

C. SIMULATION RESULTS
The fluid-solid coupled temperature field of the HTDDD PMSM designed above is simulated using Fluent software. Fig. 7 shows the temperature distributions of the motor components for operation under the rated condition.

Fig. 7 shows that the highest temperature of 343.8K is found at the rotor end, whereas the temperature of the subthermal component (i.e., the end of the lower stator winding) is 335.4K. The maximum in the overall temperature distribution occurs at the rotor because the stator lies near the cooling water channel, and the cooling water removes most of the heat from the stator core and winding.

V. INFLUENCE OF DIFFERENT FACTORS ON TEMPERATURE RISE
A. INFLUENCE OF THERMAL CAPACITY ON TEMPERATURE RISE
As the torque at constant volume of the HTDDD PMSM is increased, the electric load in the thermal capacity of the motor is raised, thereby the copper loss and the motor temperature are increased. The influence of the thermal capacity on the temperature rise is analyzed to improve the torque density of the HTDDD PMSM. The copper loss in the rated state of the HTDDD PMSM is 6.9 kW. Increasing copper loss in the electric load by 1 and 2 times is 15kW and 20kW, respectively. When the copper loss is 6.7 kW, 15 kW and 20 kW, respectively, the simulation results of the fluid-solid coupled temperature field are shown in Table 3. The winding temperature distributions for different thermal capacities are shown in Fig. 8.

Increasing the copper loss to 15 kW and the electric load to 683A/cm result in a maximum temperature rise of 68.4K in the winding, and the average temperature rise of 62.2K meets
TABLE 3. Comparison of temperature rise for different thermal capacities.

| Thermal Capacity | Copper Loss (kW) | Electric Load (A/cm²) | Torque Density (Nm/kg) | Winding Maximum Temperature Rise (K) | Winding Average Temperature Rise (K) | Rotor Average Temperature Rise (K) |
|------------------|-----------------|-----------------------|-----------------------|--------------------------------------|-------------------------------------|-----------------------------------|
| Class B          | 6.9             | 467                   | 12.46                 | 34.91                                | 29.35                               | 37.07                             |
| Class A          | 15              | 683                   | 17.68                 | 68.40                                | 62.20                               | 45.46                             |
| Class C          | 20              | 790                   | 23.71                 | 100.80                               | 85.51                               | 58.27                             |

FIGURE 9. Waterway structure models: (a) rectangular steel plate axial waterway and (b) spiral waterway.

FIGURE 10. Comparison of cooling effects of three waterways.

B. INFLUENCE OF WATERWAY STRUCTURE ON COOLING EFFECT

Fluid-solid coupled temperature field simulations are carried out for a HTDDD PMSM with a finned reinforced waterway, a rectangular steel plate axial waterway and a spiral waterway: the latter two waterway structures are shown in Fig. 9. The temperature rise in the main component of the HTDDD PMSM for the three waterway structures is shown in Fig. 10.

The rotor and the outlet temperatures for the three schemes are not very different. The average temperature rise in the winding for scheme 1 is 3.38K smaller than that for scheme 2 and 1.23K smaller than that for scheme 3; therefore, the reinforced finned waterway structure has the highest cooling effect, and the rectangular steel plate axial waterway structure has the lowest cooling effect. Although there is still a small amount of room for optimization by changing some parameters such as the number of water channels and the area of water channel of the spiral waterway structure, its cooling effect is still lower than that of reinforced fin waterway structure. It is difficult to process and has high processing cost. The reinforced finned waterway structure increases the heat dissipation area, improves the strength of the housing and reduces processing difficulty and cost. Therefore, the reinforced finned waterway structure is adopted.

C. INFLUENCE OF WATERWAY VELOCITY ON COOLING EFFECT

Increasing the water velocity intensifies turbulence, which increases heat removal by water constantly hitting the channel bottom. Therefore, in the motor model with the finned waterway, only the inlet water velocity is varied in the simulation. Fig. 11 shows curves of the maximum temperature and the average temperature rise in the stator winding for different water velocities.

As the water flow increases, the reduction in the winding temperature becomes increasingly smaller, and the improvement in the cooling capacity is increasingly reduced. Therefore, the inlet parameters must be selected appropriately. The optimal water velocity for the cooling system is 1.77 m/s, because there is very little change in the temperature rise for water velocities above this value. The corresponding water flow is 2.0 m³/h. At an inlet water velocity of 1.77 m/s, there is a 3.1K temperature difference between the inlet and outlet. The velocity vector nephogram for water and the channel temperature are shown in Fig. 12.

VI. EXPERIMENTAL VERIFICATION

To validate the accuracy and rationality of the proposed calculation method for the waterway structure and temperature rise of the HTDDD PMSM, a 200-kW prototype is
TABLE 4. Main prototype parameters.

| Items                  | Value |
|------------------------|-------|
| Rated power (kW)       | 200   |
| Rated voltage (V)      | 1140  |
| Rated current (A)      | 121   |
| Rated frequency (Hz)   | 30    |
| Rated speed (r/min)    | 60    |
| Rated torque (Nm)      | 31787 |
| Stator outer diameter (mm) | 1220 |
| Air gap (mm)           | 2     |
| Core length (mm)       | 700   |
| Rotor outer diameter (mm) | 976  |
| Rotor inner diameter (mm) | 830  |

devolved, and its main parameters are shown in Table 4. We measure the temperature rise for prototype operation under the rated condition. The experimental apparatus used to test the HTDDD PMSM is composed of the prototype, a torque sensor, a gearbox, a 1000-kW induction motor, a frequency converter, a water cooling system, and an observation bench, as shown in Fig. 13.

Fig. 14 shows the temperature test papers attached to the surface of the rotor magnetic pole to measure the rotor temperature. The temperatures at the end and middle of the rotor are 65°C and 62.5°C, respectively.

In addition, type PT100 thermal resistances are embedded in the winding, frame, bearings and outlet. The corresponding temperature test curves are shown in Fig. 15. The temperatures at the abovementioned test points gradually stabilize after 95 minutes. The stable temperatures of windings U, V, and W are 60.2°C, 62.5°C, and 58.3°C, respectively. The stable temperatures of the front bearing, the rear bearing, the frame, and the outlet are 36.9°C, 37.9°C, 32.0°C and 29.4 °C, respectively.

The temperature distribution of the HTDDD PMSM obtained using the fluid-solid coupled finite element method (FCFEM) is shown in Fig. 16. and Fig. 7 shows the temperature distribution of its main components.

Table 5 shows the temperatures of the HTDDD PMSM components calculated by the equivalent thermal network method (ETNM) and FCFEM, as well as the experimental value and calculation errors.

The results obtained using the equivalent thermal network mode and the fluid-solid coupled temperature field simulation are approximately the same as the experimental results, and the calculation errors of the two numerical methods are within a reasonable range. Therefore, a reasonable motor temperature is obtained from the proposed equivalent thermal
network model for the design of HTDDD PMSM waterway structures. Moreover, the experimental results show a high cooling effect from the designed waterway structure and a low motor temperature rise, which meets the requirements for the motor temperature rise.

VII. CONCLUSION

In this paper, we propose a waterway structure design method based on a temperature cycling module, present the detailed waterway structure design process for a HTDDD PMSM, and effectively analyze the influences of the thermal capacity, the waterway structure, and the water velocity on the motor temperature rise and cooling effect. The following conclusions are drawn.

1. The thermal capacity is the parameter that determines the basic size and temperature rise of the motor. When the electric load in the thermal capacity is increased to 683 A/cm, the motor temperature rise satisfies the class B insulation temperature rise limit.

2. Simulations of the fluid-solid coupled temperature field of the cooling system for different waterway structures and water velocities show that the reinforced finned waterway structure produces the highest cooling effect for an optimal cooling water velocity of 1.77 m/s.

3. The temperature rise calculated by the equivalent thermal network method and the simulated fluid-solid coupled temperature field are basically consistent with the experimental results. The cooling effect of the waterway structure designed using the temperature cycling module is higher than the experimental result and meets the temperature rise requirement. Finally, the correctness and rationality of the proposed waterway structure design method for a HTDDD PMSM are verified.

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