A Thermal Modeling Approach and Experimental Validation for an Oil Spray-Cooled Hairpin Winding Machine

Fengyu Zhang, David Gerada, Senior Member, IEEE, Zeyuan Xu, Chuan Liu, He Zhang, Senior Member, IEEE, Tianjie Zou, Member, IEEE, Yew Chuan Chong, and Chris Gerada, Senior Member, IEEE

Abstract—While the electromagnetic aspects of hairpin windings are actively being investigated and discussed in recent literature, including the design rules together with the loss calculation and reduction techniques, the thermal performance and modeling aspects have received less attention to date. In hairpin windings, the conductors (pins) are comparatively larger and arranged as separate components in parallel within the slot. In contrast, conductors randomly overlap and contact each other for traditional random windings. The differences in the aforementioned winding physical characteristics result in a different methodology to develop the thermal network. This article presents a 3-D lumped parameter thermal network (LPTN) approach for an oil-spray cooled hairpin winding, which includes the slot thermal model configuration, the end-winding connections, together with different methodologies of analyzing the end-winding sprayed oil characteristics. The aforesaid thermal model captures unique features related to the winding technology and cooling mechanisms, such as the nonuniform end-winding temperature caused by the uneven oil-spray cooling effects. Finally, taking an existing propulsion drive hairpin stator and a bespoke-designed test setup, the presented steady-state thermal modeling approach is experimentally validated covering various experimental tests, including different spray conditions.

Index Terms—End winding, hairpin winding, oil spray, propulsion, steady state, thermal network.

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Fengyu Zhang, David Gerada, Zeyuan Xu, Tianjie Zou, and Chris Gerada are with the Power Electronics, Machines and Control Group, University of Nottingham, Nottingham NG7 2 RD, U.K. (e-mail: fengyu.zhang1@nottingham.ac.uk; david.gerada@nottingham.ac.uk; zeyuan.xu@nottingham.ac.uk; chuan.liu2@nottingham.ac.uk; tianjie.zou@nottingham.ac.uk; chris.gerada@nottingham.ac.uk).

Chuan Liu is with the Power Electronics, Machines and Control Group, University of Nottingham, Nottingham NG7 2 RD, U.K., and also with Motor Design Ltd., Wrexham LL13 7YT, U.K. (e-mail: chuan.liu2@nottingham.ac.uk).

He Zhang is with the Key Laboratory of More Electric Aircraft Technology of Zhejiang Province, University of Nottingham Ningbo China, Ningbo 315100, China (e-mail: he.zhang@nottingham.edu.cn).

Yew Chuan Chong is with Motor Design Software Technology (Shanghai) Company Ltd., Shanghai 200030, China (e-mail: eddie.chong@motor-design.com).

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I. INTRODUCTION

W ITH the requirements of high efficiency, high power density (kW/kg or kW/L) together with the equally important cost performance ($/kg) for transportation applications, motors with higher operational current densities and enhanced cooling techniques are currently active research and development topics [1], [2]. Besides the developments in electromagnetic and thermal aspects, efficient manufacturing is also a big source of entitlement for making stronger electrification business cases. Hairpin windings are increasingly being adopted in traction motors due to their reduced manufacturing time [3] and specific advantages such as the uniformity/accuracy in conductor placement[4], [5], which eliminates the uncertainties in both the ac losses as well as the end-winding shape, allowing for the design of more reliable cooling systems [6]–[8]. Furthermore, the copper fill factor is almost doubled with respect to the traditional random windings, which further helps with improving the overall performance metrics [9], [10]. The procedure of designing a hairpin stator is described in [11]–[13], including different winding design options and connection patterns.

For traditional random-wound motors, with high stator winding losses, water jacket cooling around the stator periphery is more efficient compared to air cooling [14] and is reliably used in many high technology readiness-level applications. Novel cooling topologies, such as slot water jacket cooling targeting the heat generated inside the slot directly, are also an efficient way for reducing winding temperature, albeit the increased copper losses [15], [16]. With the aforementioned cooling techniques, the concern and investigation on the end-winding heat transfer arises. Oil-spray cooling in the end-winding region is investigated in [17] and [18], highlighting its effectiveness. Local winding temperature analysis is performed and plotted as a function of both the oil flow rate and rotor speed, indicating that the rotational speed influence is less sensitive on the winding temperatures compared to the oil flow rate [17]. However, the oil convection heat transfer is not quantified in this study, which is subsequently addressed in [19] and [20], with the heat transfer coefficient (HTC)
investigated and plotted as a function of the spray parameters, such as nozzle height, inlet pressure, subcooling, and spraying angle. Nevertheless, in the aforementioned studies, the thermal behavior of the sprayed end winding in an actual spray-cooled machine is not modeled and the complicated end-winding heat transfer characteristics are simplistically approximated due to: 1) end-windings’ complex geometry and 2) the uneven temperature distribution in the end-winding region for different slots. There are research approaches to tackle the problems related to complicated end-winding geometry for air-cooled machines in [21]–[23] using empirical formulas and for directly cooled machines using a computational fluid dynamic (CFD) analysis [24], [25]. In these works, the temperature distribution at the end-winding region is assumed uniform along the end-winding circumference. However, with oil-spray cooling, the situation is more complex, as the cooling effects on different sections of the end winding are quite dissimilar, which results in a significantly uneven temperature distribution for the end winding [17]. The thermal modeling at a machine level to consider the uneven temperature distribution in the end-winding caused by the sprayed-cooling effects is not investigated within any of the existing literature.

Lumped parameter thermal network (LPTN) thermal models are widely used due to their quick solving characteristics and are a mature tool in the thermal modeling of motors with traditional random windings [26]–[29]. Exploiting motor symmetries, the motor part corresponding to half a stator slot is simulated and validated [14]. For the thermal model, the effective thermal conductivity is applied to obtain the slot thermal condition [22] and empirical formulas are derived based on the fluid characteristics in the air-gap/end-winding region [21]. However, inaccurate temperature distribution and hot-spot temperature prediction will result if the aforesaid traditional simplified half-slot symmetry models are used to conduct thermal analysis on machines with oil-spray cooling due to the uneven temperature distribution in the end-winding region.

The contributions of this article are as follows.

1) A thermal approach to analyze the hairpin winding thermal behavior is proposed, which highlights the differences between a classic random-wound machine and a hairpin winding machine. This is especially so for the slot thermal conditions together with the important coupling between electrical connections and the thermal aspects of the electrical machine.

2) The HTC and convection area formulas are derived and presented in this article for the uneven heat transfer characteristics in the end-winding region due to the sprayed cooling effects. The proposed methodologies are applicable to any form of thermal analysis, including LPTN, FEA, and CFD, and to any general motor setup employing a sprayed cooling system.

3) An LPTN model is built, which considers 1) and 2). This is experimentally validated on a specially designed test rig with a hairpin winding stator.

This article is organized as follows. In Section II, the thermal model of the whole hairpin-winding stator is presented, including the modeling technique and the thermal model parameters selection. In Section III, the convection heat transfer, which captures the uneven temperature distribution caused by the sprayed-oil in the end-winding region, is detailed for the thermal network, including the HTC and convection area calculation. Using a bespoke-developed test rig, the results of the thermal model are experimentally validated when the test rig runs to steady state in Section IV. Finally, conclusions from this research are summarized in Section V.

II. THERMAL MODELING OF HAIRPIN WINDING

A 3-D thermal network based on an existing hairpin-wound stator is developed in this section, describing the methodologies and specific considerations. For reference, the key formulas of this network are included in the Appendix. The stator investigated in this article is an off-the-shelf Delco-Remy HVH250 machine featuring a two-layer hairpin winding configuration, with parameters listed in Table I.

| TABLE I |
|---|
| **Hairpin Winding Stator Used in This Article** |
| Outer diameter | 215 mm |
| Inner diameter | 160 mm |
| Active length | 60 mm |
| Cooling Type | Oil-spray cooled |
| Winding type | Hairpin winding |
| Layer number | Two |
| Pin dimensions | 3mm × 4mm |
| Slot dimensions | 4mm × 12mm |

A. LPTN of Hairpin Winding Machine

The thermal network for machines with traditional windings considers the periodic and symmetrical geometry of machine, and typically, only a half slot or one slot is modeled for reducing computation time [14]. There are different types of LPTN developed for traditional winding in recent literature: (Type “i”) a different number of nodes are selected based on the balance between the results’ accuracy and the computational speed [30]; (Type “ii”) simplified thermal model using cuboidal element method or “T” type using average temperature in the slot [31], [32]; and (Type “iii”) nodes with improved power losses are investigated in [33] and [34]. Both the aforementioned methods require the determination of the equivalent slot thermal conductivity, considering all slot constituent materials, including copper, wire insulation, as well as impregnation resin.

However, since in the LPTN developed for the electrical machines equipped with hairpin winding, there are a small number of conductors placed in the slot compared to the traditionally wound machine, it is practical to model each individual conductor with one node without incurring time penalties. Furthermore, since the conductor has high thermal conductivity, the temperature gradient across the conductor area is small, and thus, a low node number with one node per conductor does not decrease LPTN model accuracy. Oil-sprayed cooling is commonly used in hairpin winding
electrical machines, which results in an uneven temperature distribution in the end-winding region and also in considerable axial heat transfer along the high thermal conductivity conductor. Therefore, in this article, a 3-D full-slot LPTN model is proposed on an oil-spray cooled hairpin winding featuring multiple nozzles.

In the 3-D thermal network, the temperature of a hairpin conductor is considered to be constant within its cross section, with the number of nodes in the radial direction of the slot determined by the conductor layer number per slot, i.e., two in this case study. The two layers of hairpin winding are geometrically in parallel inside the slot, i.e., an upper conductor and a lower conductor for each slot, as shown in Fig. 1. Fig. 1 shows an example of the radial thermal network for two adjacent slots, where the gaps between the conductors themselves and between the conductors and stator core are exaggerated for illustrative purposes. It is also worth noting that the thermal resistance of the insulation material (enamel) that surrounds the copper is also included.

The axial thermal network connecting three adjacent radial planes and two corresponding end-winding planes is shown in Fig. 2. Differently to traditional windings with crossed and twisted conductors, there are accurately defined gaps between the upper and lower layer conductors and also between the conductors from any neighboring slots in the end-winding region, as shown in Fig. 3. These enable all conductors to be directly cooled by oil droplets from the sprayed oil.

In the thermal network, on the nonsprayed end, fluid (air) temperature is then considered evenly distributed. Meanwhile, the convection heat transfer between the fluids and the stator back-iron and teeth are also considered and applied to the thermal network [21].

The convection heat transfer between the two hairpin windings from the same slot and the fluid is considered independent, with the resistances shown in Fig. 2 colored by and representing the thermal resistance between the fluid and the upper layer winding, while the resistances colored by and representing the thermal resistance between the fluid and the lower layer winding.

B. Correlation Between Electrical Connection and Thermal Network

In this article, for hairpin windings, thermal paths of the end windings from different slots follow the same connection patterns as with the electrical connections between the three phases. In constructing the thermal network, it is, therefore, important to map out each hairpin’s electrical connections to describe the physical connections in the end-winding region. Fig. 3 shows an example of the physical (and thermal/electrical) connection corresponding to the motor case study in Table I, where the same conductor is crossing across six slots on one side.

Fig. 4 shows a map of hairpin stator electrical connections for one phase, and also, the corresponding thermal resistances at both the nonsprayed and the oil-sprayed sides are shown. The winding for the phase shown in Fig. 4 starts from the upper winding in slot 8 and ends in the bottom winding in slot 68, as indicated by the (●) within these slots. The solid fill (■) resistances represent the thermal resistances on the oil-sprayed side, whereas the hollow-fill (□) resistances represent those on the thermally insulated (i.e., nonsprayed-oil) side. Lines in different colors (○, ⚫, ■, □) represent one single conductor, with the solid lines representing the conductor on the oil-sprayed side and the dashed lines representing the conductor on the nonsprayed side. Conductor axial thermal resistances for the straight section (i.e., within the slot/core) are not shown on purpose, in order to give a clear view of the end-winding thermal connections.

The connections between one node and the adjacent nodes (“n” nodes in total) in the described LPTN are built with the heat conservation equation

\[ q_i + \sum_{j \neq i} \frac{T_j - T_i}{R_{ij}} = 0 \]
where “$R_{ij}$” is the thermal resistance between nodes “$i$” and “$j$,” while node temperature is represented by “$T_i$” and “$T_j$,” respectively. In (1), “$n$” represents the total number of nodes that are adjacent to the “$i$” node, while “$q_i$” is the heat loss generated in node “$i$.” For this article which focuses specifically on the thermal modeling, only dc copper losses are considered. This assumption, as in previous similar research [14], for the purpose of thermal model validation is considered valid since next-generation high-power-density machines are copper loss dominant [2], and furthermore, the main effect of the winding ac copper losses would be to increase the copper loss magnitude, hence increasing the dominance of copper losses [13], [35]. The additional copper losses from ac effects in hairpin windings are predominantly concentrated on the periphery of the conductor due to skin effect, which in a way makes their heat removal by spray cooling easier.

Furthermore, only thermal conduction within the solid parts together with convection at the end-winding region are considered, while any heat dissipation by radiation is ignored. In general, thermal conduction resistances are described with

$$R = \frac{x}{kA_{cross}} \quad (2)$$

where “$x$” is the distance between adjacent nodes along the heat flow, “$A_{cross}$” is the cross-sectional area that is perpendicular to the heat flow, and “$k$” is the material thermal conductivity between the two nodes and is determined as follows for different parts in the stator. The stator lamination thermal conductivity is 30 and 0.87 W/(m·K) in the radial and axial planes, respectively. Differently from the traditional windings, where an equivalent thermal conductivity is typically used in the slot [0–2 W/(m·K)] [36], for hairpin windings, the actual copper thermal conductivity value [401 W/(m·K)] is used in the thermal network for the conductors in slot. The impregnation (Epoxylite TSA220) thermal conductivity is 0.21 W/(m·K), while the slot liner (Nomex 410) thermal conductivity is 0.17 W/(m·K).

Convection heat transfer occurs mainly in two places: 1) between the housing and ambient air and 2) in the end-winding region. The following equation is used to calculate the convection thermal resistance:

$$R = \frac{1}{hA_{conv}} \quad (3)$$

where “$h$” is the HTC and “$A_{conv}$” is the effective convection area. For convection heat transfer between housing and ambient air, “$A_{conv}$” corresponds to the surface area of the housing frame, while “$h$” is the natural convection HTC with the experimental values adopted from [37]. For convection that occurs in the end-winding region, “$h$” is determined by the experimental results, as detailed in Section III.

### III. Convection Heat Transfer in End-Winding Region

This section presents in detail the methodologies to deal with the convection heat transfer between the sprayed oil and the end winding. Section III-A describes the equivalent convection area calculation, whereas the HTC calculation formulas are presented in Section III-B. The combinations of different convection areas and HTC formulas are incorporated in the thermal model detailed earlier in Section II for thorough experimental validation in Section IV.

#### A. Equivalent Convection Area Calculation

Due to the rectangular shape of the hairpin conductors and the nature of axially oriented sprayed oil, only the surfaces facing the sprayed oil are directly impinged by the oil droplets and achieve a better cooling performance. On the other hand, for the conductor surfaces “hidden” from the direct sprayed-oil flow (shaded in blue in Fig. 3), the cooling performance is somewhat weaker as it depends on the splashed oil droplets from the aforementioned “direct” surfaces, which result into an oil mist flowing around the end winding. Furthermore, oil films, covering the conductor surfaces, also contribute to the cooling improvement of hairpin end windings. Unfortunately, it is almost impossible to identify and separate all the aforesaid effects of the sprayed-oil cooling performance based on the experimental data of end-winding temperatures. Thus, in this work, an effective surface area coefficient “$\beta$” is introduced to reflect the effective convection area between the hairpin end winding and the sprayed oil, considering the effects of the impingement as well as the physical distance between the nozzle and the end winding. This coefficient is then used to calculate the effective convection thermal resistance and is implemented into the thermal network model.
Based on different assumptions and levels of complexities, three methods are presented to calculate “A_{conv}” for the end-winding region in this article, as per (5)–(7)

\[ A_{\text{end}} = 2(a + 2b) \times l \]
\[ A_1 = A_{\text{end}} \]
\[ A_2 = \beta_0 \times A_{\text{end}} \]
\[ A_3 = \beta \times A_{\text{end}} \]
\[ l = H + 3c. \]

In the most basic form, when \( A_{\text{conv}} = A_1 \), it is assumed that the effective convection area is the total surface area of the hairpin conductor “A_{end}” (i.e., the entire blue-shaded area of Fig. 3 is also considered to be impinged by the sprayed oil). In this case, “A_1” is the same as “A_{end}” calculated by multiplying the perimeter (profile) of the rectangular bar by half the arithmetic end-winding length “L” as described by (4) and (8). In these equations, referring to Fig. 3, “a” and “b” are the conductor width and depth, respectively, “c” is the distance between the centers of two adjacent slots, and “H” is the end-winding height.

From the foregoing description of “A_1,” it can be understood that this is an optimistic (best case) scenario. An opposite assumption can be made that one side of the conductor is entirely hidden from the benefits of spray cooling (i.e., blue side of Fig. 3 is dry), which corresponds to the effect represented by the coefficient \( \beta_0 \) in (6), for which \( A_{\text{conv}} = A_2 \). In this case, it follows that the coefficient \( \beta_0 \) is formulated from the ratio of three conductor sides to the entire conductor perimeter as described by the following equation:

\[ \beta_0 = (a + 2b)/(2a + 2b). \]

For the case in hand, considering the pin dimensions, \( \beta_0 \) is equal to 0.7857.

The third and the last approach for the determination of the effective convection area, \( A_{\text{conv}} = A_3 \), rely on using measured data to modify the convection area between the sprayed-oil nozzle and the slot location for each individual hairpin conductor. This, therefore, considers the physical distance of the slot to the spray nozzles in the experiment. In this case, the definition of “\( \beta \)” is based on the experimental data measured from the end winding of different hairpin slots

\[ \beta = \frac{\beta_0(T_{\text{ave}} - T_{\text{oil}})}{T_{\#\text{slot}} - T_{\text{oil}}} \]

where “\( T_{\text{ave}} \)” is the measured average end-winding temperature, “\( T_{\text{oil}} \)” is the sprayed oil temperature, and “\( T_{\#\text{slot}} \)” is the measured temperature specific to that slot. Details of measurement points are described in Section IV.

B. HTC Calculation

The foregoing discussion dealt with the considerations and calculation methods for the effective convection area “A_{conv}.” From (3), the other important aspect is the HTC “\( h \).” For this, again, three approaches are considered in this article. In the simplest calculation, from the experimentally measured data (end-winding temperature and heat input), the average HTC “\( h_{\text{ave}} \)” of all the end winding is obtained as follows:

\[ q = \frac{Q}{2 \times N} \]
\[ h_{\text{ave}} = q/[A_{\text{end}}(T_{\text{ave}} - T_{\text{oil}})] \]

where “\( q \)” is the corresponding loss of a single conductor dissipated to the sprayed oil, “\( Q \)” is the heat loss dissipated to the sprayed oil (heat losses dissipated to the ambient environment have been excluded in the sprayed-oil experiment), “\( N \)” is the slot number (72 for the case in hand), and “\( T_{\text{ave}} \)” and “\( T_{\text{oil}} \)” are average conductor end-winding temperature and sprayed-oil temperature, respectively.

A second approach for the calculation of “\( h \)” which puts more resolution, is suitable for the situation where nozzles are placed in symmetric locations, and thus, thermal conditions are approximately symmetrical (bar the effects of oil gravity) while exhibiting a degree of periodicity. To illustrate this, considering the motor in Table I, a case is investigated where 12 nozzles are evenly distributed circumferentially around the end winding, while they spray onto the end winding in the axial direction, one at each hour position (using the clock positioning system), as shown in Fig. 5. In such a case, Fig. 5 shows that the end-winding area for six slots is covered by each nozzle. Considering the various symmetries, this results in four independent HTCs (i.e., “\( h_{\text{ave1}} \), “\( h_{\text{ave2}} \), “\( h_{\text{ave3}} \), and “\( h_{\text{ave4}} \)”), as shown in Fig. 5. Table II summarizes the HTCs distribution on the oil-sprayed side and formulas from (13) to (16) are used to calculate the average HTC. For example, “\( h_{\text{ave1}} \)” is calculated with (13), where “\( T_i \)” is the average temperature of conductors measured
TABLE III
FOUR METHODOLOGIES OF END-WINDING OIL-SPRAY SIMULATION IN THE THERMAL NETWORK

| End-winding methodology | $h$ | $A_{conv}$ |
|-------------------------|-----|-----------|
| #i                      | $h_{ave}$ in (12) | $A_{conv} = A_1 = (2a + 2b) \times l$ |
| #ii                     | $h_{ave1}, h_{ave2}, h_{ave3}, h_{ave4}$ | $A_{conv} = A_2 = \beta_0 \times (2a + 2b) \times l$ |
| #iii                    | Single conductor local | $A_{conv} = A_3 = \beta \times (2a + 2b) \times l$ |
| #iv                     | $A_{conv}$ | $A_{conv} = A_3 = \beta \times (2a + 2b) \times l$ |

The third approach for the calculation of “$h$” calculates the local “$h$” for each single conductor, in which the case “$T_{ave}$” in (12) is replaced by the local temperature “$T_{#slot}$,” as in the following equation:

$$h_{#slot} = q/[A_{end}(T_{#slot} - T_{oil})].$$

C. Methodologies of End-Winding Oil-Spray Simulation

Based on the foregoing discussions for the calculation of “$A_{conv}$” and “$h$,” thermal networks are developed with four methodologies to represent the convection heat transfer for the oil-sprayed end-winding region, as shown in Table III. The thermal network with all the aforementioned end-winding methodologies will be applied to different case studies and compared to experimental results in Section IV.

IV. EXPERIMENTAL VALIDATION

Using a bespoke-developed test rig shown in Fig. 6, the thermal network of Section II is validated experimentally using two approaches: 1) natural convection, where housing in Fig. 6 is exposed to the ambient air, and 2) oil-sprayed cooling, where the end windings on the “oil-sprayed end” in Fig. 2 are exposed to the sprayed oil, and the full test rig casing is thermally insulated from the ambient environment using 6-mm-thick calcium–magnesium silicate sheets in order to minimize the uncertainty factors of the convection heat transfer between the test rig and the ambient environment. The hairpin winding is heated by a dc current from the power supply shown, while the generated losses are removed by sprayed oil onto the hairpin end windings. Twelve (12) nozzles, one at each hour position, are installed on the machine front end flange as shown in Fig. 6 and are used for generating axial sprayed oil on the oil-sprayed side. A data logger is used to record the temperatures from the multiple “K”-type thermocouples, which are directly attached to the winding and covered by a thin layer of insulation.

Thermocouples are located on both layers of the oil-sprayed end-winding region in order to capture a full image of the temperature distribution, as shown in Fig. 7. In Fig. 7, the green and red colors denote thermocouples placed on the upper and lower layers, respectively.

A. Natural Convection Experimental Validation

Natural convection tests are performed in order to validate the thermal network model of Section II under a simple cooling condition. For the case of natural convection, air in the enclosed end-winding region is driven by buoyancy only, and hence, the cooling performances for all end-winding conductors are similar and uniform. Heat generated in the conductors is conducted to the machine housing and then dissipated to the ambient air by natural convection. In Fig. 8, the average end-winding temperature is plotted versus the input current, with (■) and (–) denoting the experimental and simulation results, respectively. The experimental data agree well with the simulation results, with acceptable deviation mainly due to the thermocouple location accuracy.

The thermal resistances model developed in the thermal network, including those radially, axially, as well as the interconnections in the end-winding region, are validated with the natural convection experiment, with natural convection HTCs obtained on the machine in the lab environment. The
The natural convection experiment is conducted in a large enclosed room with a controlled dc current applied to the winding in the absence of any active cooling. Environment temperature, machine winding, as well as housing temperatures are monitored until a thermal steady condition is reached, at which point all the heat generated within the windings is dissipated via natural convection to the air surrounding the test section. The average HTC at the test section surface can be obtained from (18). In (18), \( h_{\text{natural}} \) is the average natural HTC, \( Q_{\text{rig}} \) is the heat input to the winding, \( A_{\text{rig}} \) is the total surface area of the housing, and \( T_{\text{housing}} \) and \( T_{\text{ambient}} \) are the housing temperature and the ambient environment temperature, respectively, with the data listed in Table IV. The HTC used in the modeling is 6.2 \( W/(m^2\cdot K) \).

\[
h_{\text{natural}} = \frac{Q_{\text{rig}}}{A_{\text{rig}}(T_{\text{housing}} - T_{\text{ambient}})}. \tag{18}
\]

### B. Prediction of Axial Oil-Spray Cooling Experiments

For forced sprayed oil (BP Turbo Oil 2389, 40 °C) to the stator, various oil flow rates for the full test rig ranging from 2.40 to 3.53 L/min and a 90.8-A current are applied to validate the thermal network model described in Section II. In this section, first, the temperature difference between the two conductor layers is discussed, followed by the temperature variation with the slot number and oil flow rate. In the final part, simulation and experimental results comparisons are presented for the purpose of thermal network model validation, involving the four cases shown in Table V. For each case, the thermal network with the four end-winding methodologies presented earlier in Table III is applied and compared.

In the first instance, the temperature difference between the two conductor layers (i.e., upper layer and lower layer) is investigated in correlation with the oil flow rate for slot#1, slot#4, and slot#16. The cooling effects on the two layers for the different slots are almost the same across the range of oil flow rates considered, with a maximum discrepancy of 10% observed from the experimental tests. In light of this, for the sake of clarity while still maintaining accuracy, only the lower layer conductor in the slot is plotted and analyzed hereafter.

Fig. 9 shows a close-up view of the oil-spray phenomenon in a configuration with 12 full-cone nozzles evenly applied to the end winding, and Fig. 10 shows the experimentally.

![Fig. 8. Average end-winding temperature distribution for natural convection.](image)

![Fig. 9. Oil spray testing for case I with 50-mm distance between the nozzle and the target surface.](image)

![Fig. 10. Experimental temperature distribution for different slot numbers.](image)
measured results of end-winding temperature for different slots under varying oil flow rates. Uneven temperatures are observed from different slots, which highlights that the thermal performance is different for each slot due to the nature of sprayed-oil cooling. At the locations from 10 o’clock to 2 o’clock (slot#60 to slot#10), the heat transfer rate values are smaller compared to the other locations due to gravity effects, which results in higher winding temperatures in this region. While the temperature pattern is similar for different flow rates, the temperature variation range increases as the flow rate reduces.

As shown in Table V, the thermal network with four end-winding methodologies (i.e., #i, #ii, #iii, and #iv) developed in Section III is applied to experimental cases I and II, both having 12 evenly distributed full-cone nozzles. In Case I, shown in Fig. 11, a higher flow rate (3.5 L/min) is used, whereas for Case II, corresponding to Fig. 12, the flow rate is reduced down to 2.4 L/min. For both the aforesaid flow rates, the temperature variation range increases as the flow rate reduces.

The contact area between the end winding and the oil is modified in the thermal network with end-winding methodology “#ii.” The simulation results plotted with (—) provide a more accurate temperature profile, compared to the thermal network with end-winding methodology “#i.” However, it should be borne in mind that with the repeating “average $h$” used, the impingement effects from oil spraying on HTC are captured only based on the physical distance of slot to the spray nozzles. Effects of oil film flow on the end winding are not included, hence why the simulation pattern plotted by (—) follows the experimental data, though with larger deviations, compared to thermal network with end-winding simulation “#iv.”

Considering the oil gravity effects, which make the oil film thicker around the 4–8 o’clock locations (slots #25–#48), indeed when using single conductor local “$h$,” Figs. 11 and 12 show that the simulated end-winding temperatures agree quite well with the experimental data for the thermal network with end-winding methodology “#ii,” with small deviations. For reference, in case I with end-winding methodology “#iv,” the HTC is ranging from 212 to 625 W/m²·K, while the surface heat flux varies from 3610 to 5038 W/m² for different slots.

To compare further the thermal model developed in Section II-A, experimental cases III and IV are performed by considering different nozzle numbers and types. Fig. 13 shows the simulation and the experimental results for case III, featuring six full-cone nozzles evenly placed around the end winding with a flow rate of 1.8 L/min. In this case, six temperature cycles are observed, and as in the previous two experimental plots, the temperature range is higher around the

### Table VI

| Tool   | Thermal network with end-winding methodology | Measured tool end-winding temperature (°C) |
|--------|---------------------------------------------|-----------------------------------------|
| pe  | #i                                           | 51.22 (±2.27%)                          |
| pe  | #ii                                          | 51.46 (±2.09%)                          |
| pe  | #iii                                         | 52.20 (±0.68%)                          |
| pe  | #iv                                          | 52.07 (±0.24%)                          |
| pe  | Measured tool end-winding temperature (°C)   | 52.10 (±1.22%)                          |

| Tool   | Thermal network with end-winding methodology | Measured tool end-winding temperature (°C) |
|--------|---------------------------------------------|-----------------------------------------|
| pe  | #i                                           | 71.79 (±4.33%)                          |
| pe  | #ii                                          | 71.93 (±4.57%)                          |
| pe  | #iii                                         | 72.00 (±4.87%)                          |
| pe  | #iv                                          | 72.05 (±5.12%)                          |
| pe  | Measured tool end-winding temperature (°C)   | 72.10 (±4.18%)                          |

The thermal network with end-winding methodology “#ii” (—), where the HTC is repeating periodically over four slots as shown in Table III, follows the experimental data pattern, with some differences which can be rooted to the local phenomena that are not modeled in this case, such as the gravity effects. As shown in Table VI, which tabulates key comparative data, the average temperature of the end winding is slightly higher compared to that predicted with the thermal network with end-winding methodology “#i,” as the convection area becomes smaller with the area modification factor $β_0$. The contact area between the end winding and the oil is modified in the thermal network with end-winding methodology “#ii.” The simulation results plotted with (—) provide a more accurate temperature profile, compared to the thermal network with end-winding methodology “#i.” However, it should be borne in mind that with the repeating “average $h$” used, the impingement effects from oil spraying on HTC are captured only based on the physical distance of slot to the spray nozzles. Effects of oil film flow on the end winding are not included, hence why the simulation pattern plotted by (—) follows the experimental data, though with larger deviations, compared to thermal network with end-winding simulation “#iv.”
slots within the 10 o’clock to the 2 o’clock region where cooling performance is weaker due to oil gravity effects.

Similar to the previous cases, for case III, both thermal networks with end-winding methodologies “#iii” and “#iv” provide a more accurate temperature profile, compared to the thermal networks with end-winding methodologies “#i” and “#ii.” Finally, for experimental case IV, the 12 full-cone nozzles in case I are replaced by hollow-cone nozzles. Large temperature variations are observed in Fig. 14 at nozzles locations from 10 to 2 o’clock, similar to all the previous cases. However, there are no clear patterns of the temperature distribution with this type of nozzle. In fact, only the thermal network with end-winding methodology “#iv,” which captures both the oil film thickness as well as the oil impinging effects, matches well with the measured test values for all the four experimental cases studied. Thus, developing thermal networks with end-winding methodology “#iv” can be applied to general hairpin-stator oil-spray cooling arrangements.

Table VII lists the convection area coefficient “β” value ranges calculated from the measured points. It can be seen that “β” range (0.4–1) is similar between cases “I,” “II,” and “IV,” where the nozzle number (12) and nozzles’ locations are the same. With case “III” having a lower nozzle number (six nozzles), compared to cases “I” and “II” (12 nozzles), it can be seen that spray cooling is more effective for higher nozzle numbers as the convection area coefficient is larger. From Table VII, it can be summarized that the convection area coefficient is more sensitive to the number of nozzles and their locations rather than the nozzle type or flow rate.

V. CONCLUSION

Oil-spraying and hairpin windings meld well the manufacturing and thermal management aspects to simultaneously improve the motor’s power density and cost performance. This is placing an immediate need for reference guidelines for the design and analysis of such systems. In this article, a 3-D LPTN approach is presented and experimentally validated. While the fundamentals of the LPTN bear similarities to traditional models for random-wound stators with water-jacket
cooling, some important particularities need to be considered when modeling oil-sprayed hairpin stators, and specifically, the following conditions hold.

1) Requirement to model the whole machine (360°) rather than using symmetries with half-slot models, in order to capture the nonuniform temperature distribution due to characteristics of sprayed cooling which cause hotspots in the 10–2 o’clock positions.

2) For consistently accurate modeling on sprayed cooling with different flow rates, nozzle types, and so on, the convection area for heat transfer and the local HTC calculation are proposed in this article, which considers the oil-film thickness and impingement effects. For HTC values derivation, an experimental study is the most appropriate methodology available particularly for oil-spray cooling on hairpin windings, due to the geometrical complexities and complex spray mechanisms. Tests can be conducted on representative segments at the motor design stage in order to achieve the HTCs before manufacturing the full machine prototype.

3) Use of an individual slot resistance for each slot component, with the number of nodes inside the slot dependent on the hairpin layer number. This is different to traditional random-winding systems where an equivalent thermal conductivity is computed based on the material (copper/enamel/resin) composition within the slot.

4) Importance of maintaining high axial thermal conductivity, while the thermal conductivity of the impregnation resin is less critical than in traditional systems.

5) In building the LPTN and end-winding thermal resistances for the hairpin winding with sprayed-oil cooling, it is important to understand and follow the same pattern as the electrical connection diagram.

The presented guidelines, modeling approach, together with the experimental setup developed should serve as a useful reference for other researchers and practicing engineers modeling this type of motor configuration for the next-generation transport applications.

APPENDIX
NOMENCLATURE

| Symbol | Description |
|--------|-------------|
| L      | Stator length. |
| M      | Axial sections number. |
| N      | Slot number. |
| r_1    | Stator inner radius. |
| r_2    | Lower layer radius. |
| r_3    | Upper layer radius. |
| r_4    | Slot bottom radius. |
| r_5    | Stator back-iron radius. |
| r_6    | Stator outer radius. |
| r_7    | Frame radius. |
| a_1    | Pin dimension-hori. |
| a_2    | Slot width-hori. |
| b_2    | Slot depth–vert. |
| b_1    | Pin dimension–vert. |
| δ_0    | Copper insulation thickness. |

δ_1 Slot liner thickness.
δ_2 Impregnation thickness-thin.
δ_3 Impregnation thickness-thick.
δ_4 Equivalent air-gap thickness between frame and core.
c Stator back-iron width-average.
λ_0 Copper insulation thermal conductivity.
λ_1 Copper thermal conductivity.
λ_2 Impregnation thermal conductivity.
λ_3 Slot liner thermal conductivity.
λ_4 Lamination radial thermal conductivity.
λ_5 Lamination axial thermal conductivity.
λ_6 Frame thermal conductivity.
λ_7 Air thermal conductivity.
h_1 Natural convection heat transfer coefficient.
h_2 Heat transfer coefficient on the spray-side for the studied slot.
h_3 Heat transfer coefficient on the thermally-insulated side.
β Convection area coefficient for the studied slot.
H End-winding height on the sprayed side.
K End-winding height on the thermally-insulated side.
d Slots number the end-winding crossing.

A. Radial Thermal Resistances

There are two types of radial thermal resistance calculation formulas in this section.

1) It consists of different components with various thermal conductivities: \( R_1 \) calculation is applied in this case with thermal resistance in copper, wire insulation, impregnation resin, slot liner, and tooth.

2) It is single component thermal resistance calculation, such as \( R_2 \) with only tooth thermal resistance

\[
R_1 = \frac{a_1}{\lambda_1} + \frac{\delta_0}{\lambda_0} + \frac{\delta_2}{\lambda_2} + \frac{\delta_3}{\lambda_3} + \frac{\pi r_1^2 - a_1^2}{\lambda_4 \left( \frac{r_5 - r_2}{r_5} \right)} \text{ (20)}
\]

\[
R_2 = \frac{r_6 - r_5}{\lambda_4 \left( \frac{r_5 - r_2}{r_5} \right)} - a_2 \text{ (21)}
\]

\[
R_8 = \frac{r_6 - r_5}{\lambda_4 \left( \frac{r_5 - r_2}{r_5} \right)} - a_2 + \frac{1}{\lambda_7 \left( \frac{r_2 - r_6}{r_2} \right)} + \frac{1}{\lambda_6 \left( \frac{2a_2}{a_2} - a_2 \right)} \text{ (22)}
\]

In Fig. 15, thermal resistance \( R_3, R_4, R_5, \) and \( R_6 \) calculation process is similar to \( R_1 \), \( R_7 \) calculation is similar to \( R_2 \), and \( R_9 \) calculation is similar to \( R_8 \).

B. Axial Thermal Resistances and the End-Winding Region

1) Axial Thermal Resistances: Axial thermal resistance in the core part only consists of one component with single thermal conductivity. The following formula is presented as
The convection thermal resistance calculation formula used as an example to calculate the thermal resistance $R_{16}$ between two axial planes. $R_{10}, R_{11}, R_{12}, R_{17},$ and $R_{18}$ use a similar formula

$$R_{16} = \frac{L}{\lambda_1(a_1b_1)}.$$  \hfill(23)

Axial thermal resistance $R_{20}$ between the core machine and the end-winding region is calculated with the following equation, which can also be used to calculate $R_{21}$:

$$R_{20} = \frac{L}{\lambda_1(a_1b_1)} + \frac{h}{\lambda_1(a_1b_1)}.$$  \hfill(24)

2) Convection Thermal Resistances in the End-Winding Region: The convection thermal resistance calculation formula between the stator and the fluid in the end-winding region consists of two parts, axial conduction thermal resistance, and convection thermal resistance. The following formulas are used as an example to calculate the thermal resistance $R_{19}$ between the tooth and the fluid in the end-winding region. Similar formulas can be extracted easily to calculate $R_{13}, R_{14},$ and $R_{15}$

$$S_{19} = \frac{\pi((r_2 + \delta_0 + \delta_1 + b_1/2)^2 - r_1^2)}{N} - \frac{a_2b_2}{2},$$ \hfill(25)

$$R_{19} = \frac{\frac{2M}{\lambda_5S_{19}}} + \frac{1}{h_2S_{19}}.$$ \hfill(26)

The convection thermal resistance calculation between the winding and the fluid is complicated as there are four surfaces in contact with the fluid. The following formulas are used to calculate $R_{22}$ in Fig. 16. $R_{23}, R_{24},$ and $R_{25}$ can be calculated with similar formulas

$$R_{22-1} = \frac{\frac{a_2}{\lambda_1(H + 3c)b_1} + \frac{\delta_0}{\lambda_0(H + 3c)b_1}}{2}.$$ \hfill(27)

$$R_{22-2} = \frac{\frac{b_1}{\lambda_1(H + 3c)a_1} + \frac{\delta_0}{\lambda_0(H + 3c)a_1}}{2}.$$ \hfill(28)

$$R_{22} = \frac{R_{22-1}R_{22-2}}{R_{22-1} + R_{22-2}} + \frac{1}{h_2\beta(H + 3c)(2a_1 + 2b_1)}.$$ \hfill(29)

C. Connection Thermal Resistance Between Different Slot

As mentioned in Section II-A, a thermal model for the full stator is built, with thermal resistance connecting different slots. The following formula is used to calculate the axial thermal resistance between two slots (crossing “d” slots) on the oil-sprayed side, as shown in Fig. 17. A similar formula can be obtained on the nonsprayed side

$$R_{26} = \frac{H + cd}{\lambda_1(a_1b_1)}.$$ \hfill(30)

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Fengyu Zhang received the B.E. degree in thermal engineering from the Huazhong University of Science and Technology, Wuhan, China, in 2014, and the Ph.D. degree in electrical machines from the University of Nottingham, Nottingham, U.K., in 2019. She is currently a Research Fellow in the area of thermal management on electrical machines with the PEMC Group, University of Nottingham. Her main research interests include high-performance motors for transport applications and their multidomain optimization.

David Gerada (Senior Member, IEEE) received the Ph.D. degree in high-speed electrical machines from the University of Nottingham, Nottingham, U.K., in 2012. From 2007 to 2016, he was with the Research and Development Department, Cummins, Stamford, U.K., first as an Electromagnetic Design Engineer from 2007 to 2012 and then as a Senior Electromagnetic Design Engineer and an Innovation Leader from 2012 to 2016. At Cummins, he pioneered the design and development of high-speed electrical machines, transforming a challenging technology into a reliable one suitable for the transportation market while establishing industry-wide-used metrics for such machinery. In 2016, he joined the University of Nottingham, where he is currently a Principal Research Fellow, responsible for developing state-of-the-art electrical machines for future transportation which push existing technology boundaries while propelling the new technologies to higher technology readiness levels. Dr. Gerada is a Chartered Engineer in U.K. and a member of the Institution of Engineering and Technology.
Zeyuan Xu received the Ph.D. degree in mechanical engineering from The University of Manchester, Manchester, U.K., in 2002.

He worked as a Research Fellow with UMIST, Manchester, Brunel University, Uxbridge, U.K., and the University of Nottingham, Nottingham, U.K. He is currently a Senior Research Fellow of the Thermo-Mechanical Design of High-Speed Electrical Machines with the PEMC Group, University of Nottingham. His main research interests include turbulent thermo-fluid flow, heat transfer enhancement, and thermal management of advanced electrical machines and power electronics.

Chuan Liu received the B.E. degree in mechanical engineering from Nanjing Agricultural University, Nanjing, China, in 2015, and the Ph.D. degree in electrical and electronic engineering from the University of Nottingham, Nottingham, U.K., in 2020.

In 2015, he was a Research Assistant of Electrical Machine Design and Manufacture with the University of Nottingham Ningbo China, Ningbo, China. Since 2020, he has been working as a Knowledge Transfer Associate for a partnership between Motor Design Ltd., Wrexham, U.K., and the University of Nottingham. The knowledge transfer partnership aims to develop fast and accurate modeling capability to simulate and design oil spray cooling systems for the next generation of electric traction motors.

He Zhang (Senior Member, IEEE) received the B.Eng. degree from Zhejiang University, Hangzhou, China, in 2002, and the M.Sc. and Ph.D. degrees in electrical machines from the University of Nottingham, Nottingham, U.K., in 2004 and 2009, respectively.

He worked as a Research Fellow with the University of Nottingham and the Director of the Best Motion Technology Centre, Nottingham. He moved to the University of Nottingham Ningbo China, Ningbo, China, as a Senior Research Fellow, in 2014, where he promoted to Principal Research Fellow in 2016 and Professor in 2020. He is currently the Director of the Nottingham Electrification Centre (NEC), Power Electronics, Machines and Control Research Group, University of Nottingham Ningbo China. His research interests include high-performance electric machines and drives for transport electrification.

Tianjie Zou (Member, IEEE) was born in Hubei, China, in 1991. He received the B.Sc. degree in electrical and electronic engineering and the Ph.D. degree in electrical engineering from the Huazhong University of Science and Technology, Wuhan, China, in 2013 and 2018, respectively.

He joined the Power Electronics, Machines and Control (PEMC) Group, University of Nottingham, Nottingham, U.K., in 2018, as a Research Fellow. In 2020, he was awarded the Nottingham Research Fellowship and started his independent research career. His main research interests include the design, analysis, and intelligent control of permanent-magnet machines.

Dr. Zou was a recipient of the Best Paper Award in the 22nd International Conference on Electrical Machines (ICEM 2016).

Yew Chuan Chong (Eddie) received the Ph.D. degree from the University of Edinburgh, Edinburgh, U.K., in 2015.

He joined Motor Design Ltd. (MDL), Ellesmere, U.K., as a Senior Research Engineer, in January 2014. He is currently the Technical Lead (Asia) of the MDL Office, Shanghai, China, managing both technical and commercial functions in the Asia market and leading thermal aspects of research consultancy projects and Motor-CAD software development. His major research area is improving algorithms for thermo-fluid modeling in electrical machines and effectively make use of lumped parameter thermal network, flow network, finite element, and computational fluid dynamic (CFD) methods for thermal management of electrical machine and to improve the cooling of electrical machines.

Chris Gerada (Senior Member, IEEE) received the Ph.D. degree in numerical modeling of electrical machines from the University of Nottingham, Nottingham, U.K., in 2005.

He worked as a Researcher with the University of Nottingham, on high-performance electrical drives and the design and modeling of electromagnetic actuators for aerospace applications. In 2008, he was appointed as a Lecturer of Electrical Machines, in 2011, as an Associate Professor, and in 2013, as a Professor with the University of Nottingham. He is currently an Associate Pro-Vice-Chancellor of Industrial Strategy and Impact and a Professor of Electrical Machines with the University of Nottingham. He has secured over sterling 20M of funding through major industrial, European, and U.K. grants and authored more than 350 referred publications. His principal research interest lies in electromagnetic energy conversion in electrical machines and drives, focusing mainly on transport electrification.

Prof. Gerada was awarded a Research Chair from the Royal Academy of Engineering in 2013. He served as an Associate Editor for the IEEE TRANSACTIONS ON INDUSTRY APPLICATIONS and is the past Chair of the IEEE IES Electrical Machines Committee.