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ORIGINAL RESEARCH OPEN ACCESS

Comparative Study of Self-Cooling PM Machines With Different Winding Structures

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ABSTRACT

This paper compares several self-cooled permanent magnet machines having different winding configurations, including concentrated single-layer and double-layer windings as well as distributed winding. The self-cooling capability is achieved by integrating fan blades into the rotor hub, specifically, the space between the rotor core and the shaft. As rotor rotates, the fan blades draw air in through the inlets and expel it through the outlets, thereby facilitating the removal of heat generated in the windings and rotor-mounted permanent magnets. The study focuses on evaluating the maximum temperatures at key machine components, such as the magnets and end windings, under varying rotational speeds. Two scenarios have been considered in this study. One assumes that all the machines have the same copper loss density while with the core losses and magnet eddy current loss being neglected, and the other uses real losses obtained by finite element models. By conducting thermal analyses under these conditions, the self-cooling capability of permanent magnet machines with different winding structures can be comprehensively assessed. Thermal analyses based on computational fluid dynamics modelling are performed under these conditions to comprehensively assess the self-cooling capability of permanent magnet machines with different winding structures. The results demonstrate that concentrated windings gain greater benefit from the proposed self-cooling scheme than distributed windings, with the double-layer configuration showing the best overall performance. The computational fluid dynamics models were subsequently validated through a series of experiments.

1 | Introduction

Permanent magnet (PM) machines have been widely adopted in various applications such as electric vehicles, aerospace, and renewable energy, due to their notable advantages including high power density and high efficiency. PM machines have a wide range of winding structures, generally categorised into concentrated windings and distributed windings, as shown in Figure 1. In concentrated windings, each coil is wound around a single tooth. This winding structure is simple and compact, making it highly suitable for mass production. Concentrated windings can be further divided into single-layer and double-layer windings. The single-layer structure is the simplest, but it tends to produce

higher winding magnetomotive force (MMF) harmonics and has a slightly lower slot fill factor [1, 2]. The double-layer winding, on the other hand, increases slightly the slot fill factor and reduces armature magnetomotive force (MMF) harmonic contents compared to the single-layer design [3]. Different from the concentrated winding designs, distributed windings spread each coil across multiple slots. As the coil pitch of the distributed winding is often the same as the pole pitch, it can achieve higher winding factor than concentrated windings. In addition, its armature MMF contains much less harmonics, therefore, its iron loss and PM eddy current loss are generally lower. However, its overlapping winding layout is much more complex and often

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results in longer end-windings (EW). This leads to longer overall machine length, higher manufacturing costs and more complicated manufacturing procedures [4].

Regardless of the winding configuration, for almost all PM machines, thermal management remains a critical concern. During operation, PM machines will generate copper losses in the windings, eddy current losses in the magnets, and iron losses in the stator and rotor cores (mainly concentrated in the stator) [5]. These losses are heat sources, contributing to temperature rise in various machine components. Amongst all the machine components, the EWs and the PMs are the two most critical cooling targets [6]. The EW are located outside the stator slots and have poor thermal conduction paths, making them highly prone to overheating. Most insulation materials used in the windings of electrical machines degrade when exposed to elevated temperatures above their rated class, typically 180°C for Class H, potentially resulting in turn-to-turn short circuits due to reduced dielectric strength. The common PMs in high power density PM machines are often made of rare-earth alloys such as NdFeB and SmCo. They are susceptible to irreversible demagnetisation at elevated temperatures, a phenomenon that can severely affect the machine's electromagnetic performance [7, 8]. NdFeB is vulnerable to temperatures above 120°C while SmCo can tolerate higher temperature, up to around 350°C. However, the excessive heat can lead to irreversible magnetic degradation in both materials, compromising torque output and overall machine reliability [7, 9]. This issue is even more pressing in modern machine designs, where increasing demands on power density continue to push the thermal limits. Therefore, it is essential to implement an effective and appropriate cooling strategy for these PM machines.

To address these thermal challenges, a wide range of cooling methods have been developed, including natural convection, forced air cooling (including self-cooling, which will be discussed separately), liquid cooling, oil spray cooling, etc. [6, 10]. In the forced air-cooling systems, cool air is actively directed towards internal heat sources by rotor mounted fans or external air pumps, thereby enhancing heat transfer. But this comes at the cost of increased system complexity and weight. Liquid

cooling offers excellent performance but demands high levels of sealing and additional pumping systems [11, 12]. Oil or spray cooling is typically reserved for high power density applications, such as those in aerospace industries, and although highly effective, it introduces concerns about fluid compatibility and corrosion [13, 14].

In contrast, the self-cooling scheme, one example can be seen in Figure 2, stands out for its simplicity and efficiency, as it does not require external cooling systems (air pumps). In addition, compared to a standard fan cooled machine, where the rotor mounted fan is typically outside of the rotor core and mounted on the shaft, the self-cooling machine integrates the fan into the rotor hub. This will reduce the machine overall axial length, making them more compact. In a self-cooling machine, the airflow induced by the rotor's rotation will carry away internally generated heat, leading to effective cooling for various machine components such as stator end-windings and rotor PMs [15, 16]. Recent studies have further explored self-ventilated and air-cooled PM machines using CFD-based thermal analysis, with particular attention to internal airflow behaviour, end-winding cooling performance, and thermal optimization under high-speed operation [17–19]. However, to maximise its effectiveness, it must be paired with an optimised machine design, particularly in terms of winding configuration and inlet/outlet positioning [20]. This places significant demands on the geometrical and structural designs of the machine, especially the winding layout [21, 22]. However, while most existing studies focus on the design and optimization of self-cooling rotor structures, the influence of winding topology on the effectiveness of such self-cooling schemes has received much less attention [17].

Effort has been made to investigate the self-cooling rotor structures [22], in which fan blades are embedded between the rotor shaft and the inner wall to induce airflow during rotor rotation. These studies demonstrate the feasibility of self-induced convective cooling, but also reveal several limitations related to mechanical robustness and the cooling effectiveness of permanent magnets.

In this study, a blade-structured rotor hub similar to those reported in the literature is adopted as a baseline self-cooling configuration. It should be emphasised that the objective of this paper is not to propose or further optimise a new self-cooling rotor design, but to use an identical and representative self-cooling structure as a controlled platform to investigate how different winding topologies affect airflow behaviour and thermal performance. By fixing the rotor cooling concept, the impact of winding-dependent geometry on the self-cooling

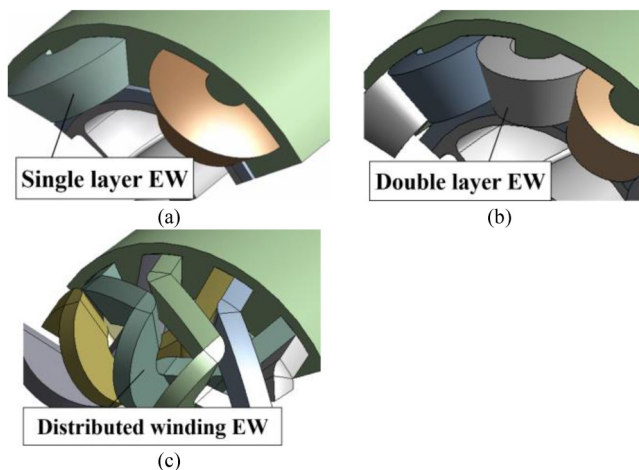


FIGURE 1 | Schematic EW structures of (a) single layer, (b) double layer and (c) distributed windings. 1/3 model is used in this figure.

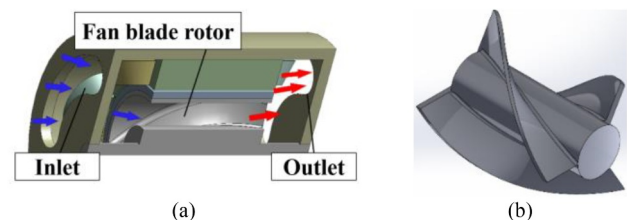


FIGURE 2 | (a) Cut away view of the SPM machine with blade rotor and (b) rotor hub design with 3 blades for self-cooling.

effectiveness can be isolated and systematically analysed. From a thermal–fluid perspective, winding topology not only affects electromagnetic performance, but also fundamentally alters end-winding geometry, airflow paths, and flow resistance inside the machine.

Different winding structures, such as the above-mentioned concentrated single-layer and double-layer windings, and distributed windings can result in distinct flow patterns and velocities of cooling air across the EW region. In this study, the terms “single-layer” and “double-layer” refer to concentrated tooth-coil windings, while the distributed winding denotes an overlapping winding configuration with coils spanning multiple slots. As a result, the self-cooling effectiveness would vary with the winding configuration, and understanding this relationship is crucial for thermally optimised machine design. To address this, a comparative study of the self-cooling performance of PM machines with different winding structures is conducted in this paper, with particular focus on the winding-dependent airflow characteristics, heat transfer behaviour, and temperature distribution under an identical self-cooling scheme. Rather than introducing additional cooling hardware, this study aims to provide design-oriented insights into how winding topology alone influences the effectiveness and limitations of self-cooling in compact PM machines.

2 | CFD Modelling and Feature of the Machines

2.1 | CFD Modelling

Because fluid flow inside the machine is predominantly turbulent, iterative solutions using computational fluid dynamics (CFD) models involve simulation methods such as Direct Numerical Simulation (DNS), Large Eddy Simulation (LES), and Reynolds Averaged Navier–Stokes (RANS), all of which are governed by a set of nonlinear partial differential equations from (1) to (4). Amongst these methods, the RANS approach is employed in this study because it requires significantly fewer computational resources compared to the others, making it more commonly used in simulation calculations [23, 24].

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho u) = 0 \quad (1)$$

where ρ is the fluid density (kg/m^3), t is the time (s), ρu is the mass flux vector, and ∇ is the divergence operator.

$$\frac{\partial (\rho u)}{\partial t} + \nabla \cdot (\rho u \otimes u) = -\nabla p + \nabla \cdot \tau + \rho f \quad (2)$$

where u is the velocity vector (m/s), p is the pressure (Pa), τ is the viscous stress tensor and f is the body force vector per unit mass, for example, gravity.

$$\frac{\partial (\rho E)}{\partial t} + \nabla \cdot (u(\rho E + p)) = \nabla \cdot (k \nabla T) + \phi \quad (3)$$

with

$$E = e + \frac{1}{2} u \cdot u \quad (4)$$

where e is the internal energy per unit mass (J/kg), T is the temperature (K), k is the thermal conductivity [W/(m·K)], and

ϕ is the Viscous dissipation term, representing the conversion of mechanical energy into thermal energy through shear.

In addition, the shear stress transport (SST) k - ω turbulence model is adopted to maintain numerical stability in viscous regions during simulation, thereby improving the accuracy of the fluid flow prediction [25]. Furthermore, this study also utilises the Moving Reference Frame (MRF) method. Compared to conventional simulation techniques, the MRF method only requires fine mesh generation around the rotating body to form the MRF region, making it more efficient and less demanding in terms of computational resources [26].

2.2 | Features of the Machines With Self-Cooling Capability

An SPM machine is employed to investigate a novel rotor structure resembling a propeller, as illustrated in Figure 2b. As mentioned before, rotor rotation can draw cold air in from inlets and pump it out through outlets. Both inlets and outlets are in the endplates. This type of self-cooling scheme has been proven to be highly effective [22]. By utilising a propeller-like structure, it draws in a large volume of cool air during rotor rotation and directs it through an airflow path from the inlet → through the machine interior → to the outlet, effectively removing the heat generated inside the machine, particularly from the rotor mounted PM and stator EW regions. Unlike the self-cooling strategy adopted in [22], the design proposed in this paper utilises thickened rotor blades. This structural enhancement not only allows the cooling system to be effectively applied to machines with extended stack lengths but also provides greater mechanical integrity. Additionally, the robust blades serve a dual purpose: while facilitating the intake of cooling air, their enlarged surface area also acts as a direct heat dissipation interface, like cooling fins. Since most of this heat originates from the PM, the proposed design significantly enhances the cooling effectiveness in the PM regions.

In this study, at a rotor speed of 3000 rpm, the application of a rotor with a 3-blade design [as shown in Figure 2b] can generate a cooling airflow of 91.2 L/min within the machine. Note that other numbers of blades such as 2, 4 and 5 have also been investigated, but the 3-blade design offers the best cooling efficiency, so it has been chosen for this study [19]. The generated airflow passes over the PM and EW regions, carrying away the heat. Using single layer as example, this results in a maximum PM temperature of 90.8°C and a maximum EW temperature of 85.4°C. In contrast, under the same operating conditions with a conventional rotor (without the cooling blades), the maximum temperatures can reach as high as 180°C at the PM and 161°C at the EW, demonstrating the remarkable cooling performance of this scheme.

The detailed specifications of the machine are provided in Table 1. Table 2 presents the thermal properties of various machine components [19]. As shown in Figure 1, three different CFD models of the EW were developed, corresponding to the single-layer, double-layer, and distributed winding configurations. Since this study is based on a 12-slot/10-pole SPM machine, a 1/3 symmetric model was adopted to reduce computational cost. In the single-layer winding model, each slot

TABLE 1 | Parameters of SPM machines.

Slot number	12	Stack length (mm)	50
Pole number	10	DC voltage (V)	72
Stator outer radius (mm)	50	Phase RMS current (A)	7.07
Rotor outer radius (mm)	24.5	N ^o of turns per phase	132
Airgap length (mm)	1	Rated speed (rpm)	3000
Magnet thickness (mm)	3		

TABLE 2 | Properties of machine solid components.

	Housing	Rotor/Stator	Magnet	Shaft
ρ (kg/m ³)	2790	7650	7500	7800
λ (W/m/K)	168	30	7.6	52
c (J/kg/K)	833	460	7500	460

Note: ρ , λ and c denote the density, thermal conductivity, and specific heat capacity, respectively.

contains one side of a coil, whereas in the double-layer and distributed winding models, two coil sides share a single slot. In addition, the volume of the distributed winding model (131659.2 mm³) is 4–6 times bigger than that of the single layer (37834.8 mm³) and double layer (28627.2 mm³) due to the complicated overlapping structure.

The reported volume ratio (4–6 times) refers to the EW envelope volume (both end sides) extracted from the CAD geometry, rather than the copper volume in the slots. The EW geometries for different winding configurations were generated under the same end-space constraints and consistent routing/clearance assumptions. The distributed winding naturally results in a larger EW envelope due to the longer coil pitch and end-turn overlapping in the end region, while the concentrated windings have compact tooth-coil end turns. It is worth noting that no dedicated EW layout optimization was carried out for the distributed winding in this study; the geometry represents a practical, unoptimised end-turn envelope.

2.3 | Boundary Conditions in the CFD Models

In the CFD model, all wall boundary conditions were defined as no-slip, implying zero relative velocity between the fluid and the wall surfaces. Nonetheless, since the machine's housing and end cap are in thermal contact with the ambient environment (maintained at room temperature, 25°C), it is essential to account for heat dissipation through these external surfaces. This necessitates the application of a heat transfer coefficient (HTC) on the outer walls. A widely adopted approach for estimating the convective HTC involves the use of the Nusselt number (Nu), a dimensionless quantity that characterises the ratio of convective to conductive heat transfer at a boundary [27], as described by

$$h = \frac{Nu \cdot k}{L} \quad (5)$$

where h is the convection coefficient [W/(m²·K)], k is the thermal conductivity of the fluid [W/(m·K)] and L is the characteristic length (m). Nu can be determined using empirical

correlations that depend on the flow conditions. The flow type (laminar or turbulent) can be determined by calculating the Reynolds number (Re):

$$Re = \frac{\rho v L}{\mu} \quad (6)$$

where v is velocity of the fluid (m/s), μ is dynamic viscosity of the fluid (Pa·s).

Following are common correlations between Nu and Re :

$$Nu = \begin{cases} 0.664 \cdot Re^{\frac{1}{2}} \cdot Pr^{\frac{1}{3}}, & Re < 2000 \\ 0.037 \cdot Re^{0.8} \cdot Pr^{\frac{1}{3}}, & 2000 < Re < 4000 \\ 0.023 \cdot Re^{0.8} \cdot Pr^{0.3}, & Re > 4000 \end{cases} \quad (7)$$

with

$$Pr = \frac{C \cdot \mu}{k} \quad (8)$$

where C is specific heat capacity [J/(kg·K)] at constant pressure, μ is dynamic viscosity of the fluid (Pa·s) [27].

Based on the calculations, the HTC for the outer surface of the end cap is determined to be 10.7 W/(m²·K), while that of the housing's outer surface is 12 W/(m²·K). To replicate the ambient environment more accurately, an additional surrounding air domain was incorporated into the model. The boundary conditions for this air region were defined with a fixed temperature of 25°C (room temperature) and an opening boundary set to standard atmospheric pressure (≈ 101325 Pa).

The present study focuses on steady-state thermal behaviour under fixed operating conditions. The numerical results are obtained based on specific machine geometry, ambient conditions, and boundary settings, and therefore the quantitative values may vary for different designs. Nevertheless, the comparative trends and winding-dependent cooling mechanisms identified in this work remain valid within the investigated operating range.

In the thermal model, both the active winding and end winding regions are represented as solid copper to simplify the CFD conjugate heat transfer (CHT) model. The copper loss is applied as a volumetric heat source within the winding domains. All solid–solid interfaces are treated as perfect thermal contact (bonded contact) in the CFD model to reduce modelling complexity and computational cost. In this case, the influence of slot insulation between the windings and the stator core is accounted for by introducing an

equivalent insulation thermal resistance at the winding–core interface (implemented as an equivalent insulation layer/effective thermal conductivity), so that the reduced heat conduction across the slot liner is captured without explicitly modelling detailed insulation geometry. These simplifications mainly affect the absolute temperature levels, while the comparative trends reported in this work are obtained under consistent modelling settings across all winding configurations.

2.4 | Copper Losses, Core Losses, Magnet Eddy Current Losses

The total losses in PM machines can be approximated by adding the copper losses— P_{cu} , stator and rotor core losses— P_{core} , magnet eddy current loss— P_{eddy} , and mechanical losses— P_{mech} such as

$$P_{total} = P_{cu} + P_{core} + P_{eddy} + P_{mech} \quad (9)$$

with

$$\begin{cases} P_{cu} = I^2 R \\ P_{core} = K_h B^2 f + K_c (Bf)^2 + K_e (Bf)^{3/2} \\ P_{eddy} = k_m (B_m f d)^2 \\ P_{mech} = k_{mech} \cdot \omega^2 \end{cases} \quad (10)$$

where I is the current flowing through the winding, R is the resistance of the windings. It is worth noting that in this study, only the DC copper loss has been considered, any AC effects such as skin effect, proximity effect and circulating current, etc have been neglected. K_h , K_c , and K_e are the coefficients of hysteresis loss, classical eddy current loss, and excess eddy current loss, respectively, and B is the peak flux density, f is the frequency. k_m is a lumped coefficient that accounts for the PM electrical conductivity/resistivity and geometry-related factors (e.g., magnet aspect ratio and segmentation), while d denotes the magnet thickness, which directly affects the induced eddy-current path length and therefore the loss magnitude [28]. k_{mech} is a mechanical loss constant and ω is the angular velocity. In this work, the PM eddy current losses used in the CFD simulations are obtained from electromagnetic finite-element analysis (FEA) and applied as spatially non-uniform volumetric heat sources. Equation (10) is provided to describe the loss components and their scaling with key parameters. The PM eddy current loss term is written in a limited resistance form. Under this assumption, the loss scales with $(Bfd)^2$, indicating a quadratic dependence on the magnet thickness d when the skin depth is larger than the magnet thickness [28, 29].

The iron losses are calculated using a standard loss separation approach including hysteresis, classical eddy current, and excess loss components, while mechanical losses such as bearing losses are not included as they do not directly contribute to internal heat generation within the active cooling domain. Figure 3 illustrates the variation of eddy current loss and core loss with rotational speed obtained by finite element analyses for the three investigated machines. For all three winding structures, the losses increase with speed, consistent with theoretical predictions. In addition, the loss magnitudes of the three winding structures follow the order: single-layer winding > double-layer winding > distributed winding. This is mainly because the

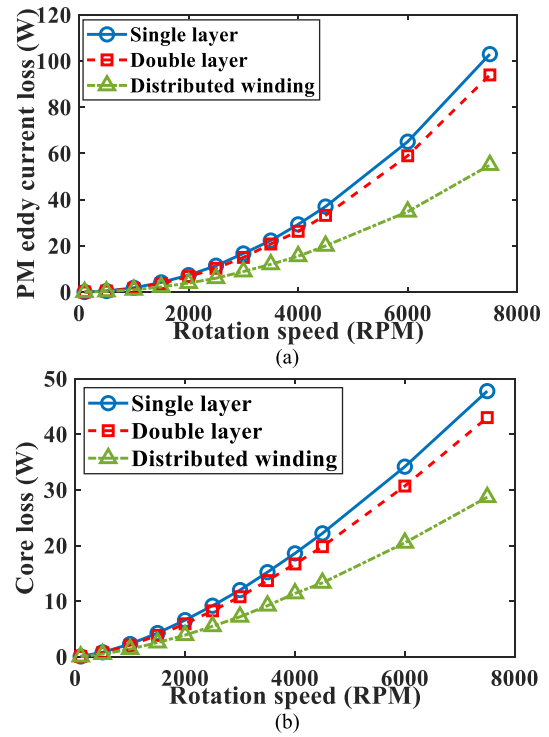


FIGURE 3 | (a) PM eddy current losses and (b) core loss versus rotation speed.

concentrated windings generally have higher harmonic components in the armature MMF than the distributed windings, with the single-layer winding being the highest. In addition, in this study, the copper losses used in the CFD simulations are based on the same copper loss density (W/mm^3). Correspondingly, the total copper loss for each winding structure is as follows: 22.14 W for the single-layer winding, 19.2 W for the double-layer winding, and 50.3 W for the distributed winding.

It is also worth noting that the windage loss generated by fan blade rotor structures was automatically included in the MRF method so no need to add this manually in the CFD modelling. However, the MRF area is around the rotor and shaft (EW are not included), the pressure drop from inlet and outlet should be monitored since the windage loss still happens when air flows around the EW area. Moreover, the PM eddy current losses are obtained from electromagnetic finite element simulations and applied to the CFD model as spatially non-uniform volumetric heat sources, thereby capturing local loss concentration and hot-spot effects.

3 | Results and Discussion

Given the use of a forced air-cooling system, variations in air properties, particularly the density and dynamic viscosity, can be negligible. Furthermore, the compressibility of air is assumed to have minimal impact. A steady, fully developed airflow with a mass flow rate of 0.48 kg/s and an inlet temperature of 25°C is supplied to the SPM machines under investigation. In the context of forced convection cooling, gravitational effect has not been considered. Due to the circumferential symmetry of the machine structure, both the flow and heat transfer exhibit circumferential symmetry as well. As a result, only a 120° (one-third) section of the machine is modelled for 3D CFD analysis.

Note that the operating conditions considered in this study are intended to investigate the steady-state thermal behaviour of self-cooled PM machines, rather than to reproduce a specific application drive cycle. At low rotational speeds, the airflow induced by the self-cooling rotor is inherently weak and the thermal behaviour is dominated by copper losses, whereas at higher speeds enhanced convection is partially offset by increasing iron and magnet losses. A comprehensive evaluation under realistic drive cycles would require transient thermal modelling with time-varying loss profiles, which is beyond the scope of the present study.

To better assess the impact of self-cooling across different winding structures, this study examines both real-case loss distribution and a simplified case where the same copper loss density is used for all windings and the magnet eddy current and core losses effect are excluded. This simplification eliminates the influence of varying losses on temperature distribution, allowing the analysis to focus primarily on the different airflow paths created by different EW structures.

It is worth noting that the results in Sections III.A, B and C are obtained using the real-loss case, where FE-derived copper loss, iron loss and PM eddy-current loss are all applied to the CFD model as heat sources. However, in Section III.D, only copper loss is applied to isolate the heat sources and show the cooling effectiveness of the self-cooling design without the impact of core and PM eddy current losses.

3.1 | Fixed Rotor Speed

Figure 4 shows the temperature contour comparisons at 3000 rpm for the three winding structures. In all plots, the bottom-left corner corresponds to the outlet region, and the top-right corner corresponds to the inlet region. For both the single-layer and double-layer configurations shown in Figure a,b respectively, the maximum temperature occurs at the EW on the outlet side, with values of 61.9°C for the single-layer and 49.5°C for the double-layer. The minimum temperatures are located at the EW on the inlet side, with 44.1°C for the single-layer and 46.9°C for the double-layer.

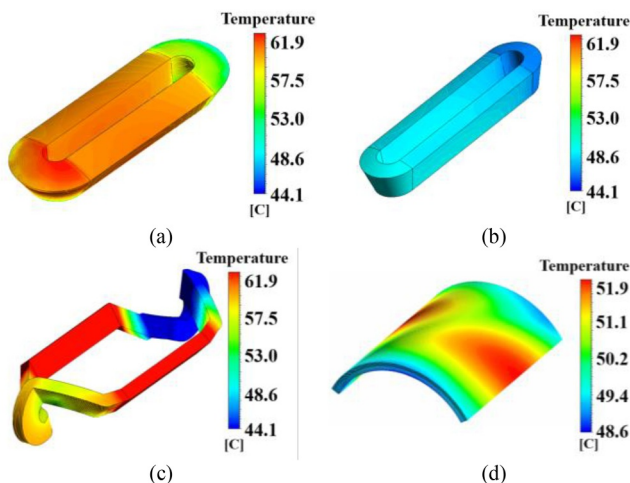


FIGURE 4 | Temperature contour of (a) single layer, (b) double layer, (c) distributed windings and (d) PM (of double layer machine).

Figure 4c illustrates the winding temperature distribution for the distributed winding structure, where the maximum temperature reaches 58.7°C, located at the active winding, and the minimum temperature is 33.1°C, found at the EW on the inlet side. This indicates that, at a relatively high speed of 3000 rpm, the self-cooling scheme is more effective for the EW region in the distributed winding, resulting in a lower temperature at the EW compared to the active winding. In contrast, for both the single-layer and double-layer configurations, the EW remains hotter than the active winding.

Figure 4d presents the temperature distribution of the PMs for the double-layer machine. Since the PM temperature distributions are similar across all three winding types, only one case is shown here for simplicity. The maximum temperature reaches 51.9°C in the middle part of the PM, while the minimum temperature is 48.6°C, located at two ends of the PM.

For better comparison, Table 3 summarises the key temperature metrics of the EW and PM regions for different winding configurations under the reference operating condition.

Figure 5 shows the air streamline distribution at a rotational speed of 3000 rpm for three different winding configurations. For the single-layer and double-layer windings, the airflow patterns are relatively similar. The air enters through the right-side inlet, reaches its peak velocity around the rotor blades, and then exits from the left-side outlet. In contrast, with the distributed winding, air follows a more complex path: after entering the inlet, it passes through the voluminous EW region, moves into the rotor hub, and exits through the opposite EW region. This layout increases the contact area between the cooling air and the EW, shaft, and PM regions, thereby enhancing the local heat exchange, particularly in the PM region.

However, the distributed winding also introduces more flow resistance due to the larger EW volume, which limits the maximum airflow velocity. At 3000 rpm, the peak velocity is reduced to 11.9 m/s, compared to 12.9 m/s for the single-layer and 12.4 m/s for the double-layer configurations. This velocity reduction becomes more pronounced at lower rotational speeds, suggesting that the self-cooling efficiency for distributed windings may degrade more significantly under low-speed conditions.

3.2 | Variable Rotor Speed

The previous results are based on a fixed rotor speed. However, changes in rotor speed will alter the airflow as the airflow rate drawn into the rotor is directly proportional to the rotor speed, which in turn affects the cooling efficiency. As shown in

TABLE 3 | Maximum temperature of three different winding structures.

	Single layer	Double layer	Distributed winding
EW maximum temperature (°C)	61.9	49.5	58.7
PM minimum temperature (°C)	54.0	51.9	52.4

Figure 6, the slopes of the curves for the single-layer and double-layer windings are very similar. This is because, in the concentrated windings, the EW volume is relatively small and does not significantly obstruct the airflow path into the machine. In contrast, the distributed winding has a much larger volume, which affects the airflow and reduces the velocity of the incoming air.

Figure 7a illustrates the variation in EW temperature across a speed range from 100 to 7500 rpm for three different winding configurations. Overall, the EW temperature for all three configurations initially decreases with increasing speed and then gradually increases. This trend is primarily due to the dual

effects of speed: on one hand, higher speed enhances airflow through the machine, improving convective cooling; on the other hand, increased speed also increases iron losses and PM eddy loss, part of which conduct heat to the EW region, thereby weakening or even offsetting the benefits of self-cooling. This explains why the EW temperature slightly increases at high speeds even when copper losses remain constant. For single- and double-layer windings, the cooling performance of the self-cooling system is most effective between 2000 and 4000 rpm, whereas for the distributed winding, the optimal cooling range shifts to between 4000 and 6000 rpm.

The cooling performance of the self-cooling strategy varies significantly across the different winding types. The single-layer winding consistently exhibits the least effective cooling performance across the entire speed range. Its temperature curve is the flattest among the three, with the highest minimum temperature (60.71°C at 3000 rpm) and the lowest starting temperature at low speed (134.59°C at 100 rpm). The double-layer winding follows a similar trend but demonstrates stronger cooling response, particularly at low speeds (below 1500 rpm), where its EW temperature drops more sharply and remains lower than that of the single-layer winding. This is because the double-layer winding has a higher surface-area-to-volume ratio in the winding region, allowing more efficient convective cooling by the incoming air. Additionally, it typically experiences lower iron and eddy current losses than the single-layer winding, resulting in lower EW temperatures at high speeds. Notably, at very low speeds (< 500 rpm), the EW temperature of the double-layer winding is even higher than that of the single-layer winding. This is attributed to the absence of forced airflow, which causes heat to accumulate locally due to poor convection, creating thermal barriers. Under such conditions, the more

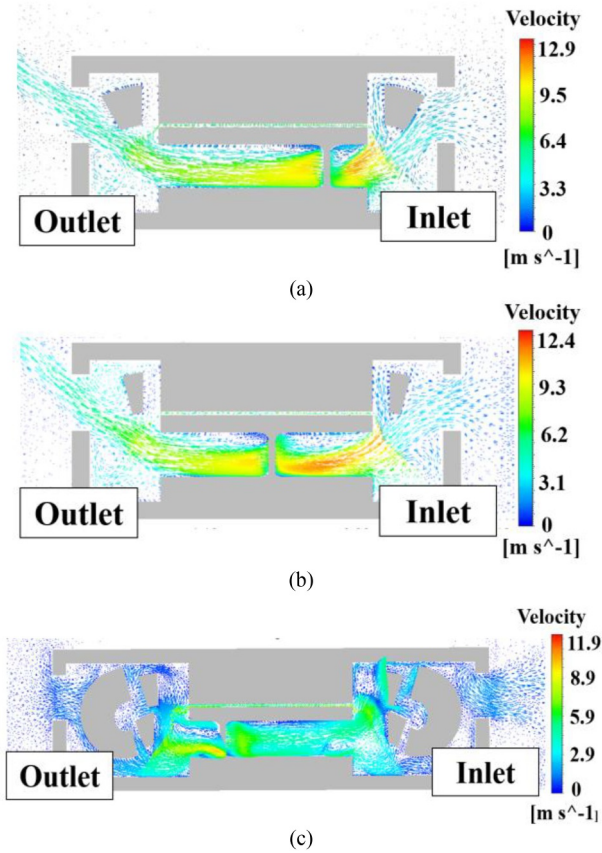


FIGURE 5 | Air velocity at 3000 rpm for (a) single layer, (b) double layer and (c) distributed windings.

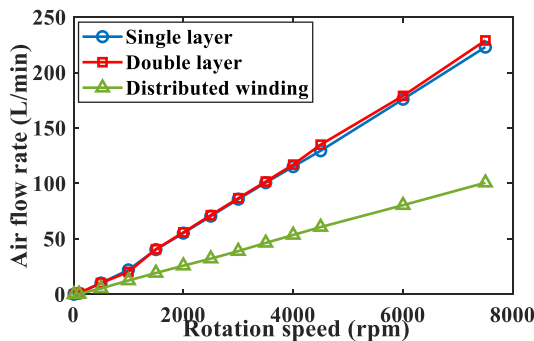


FIGURE 6 | Flow rate inside the machines with three different winding structures.

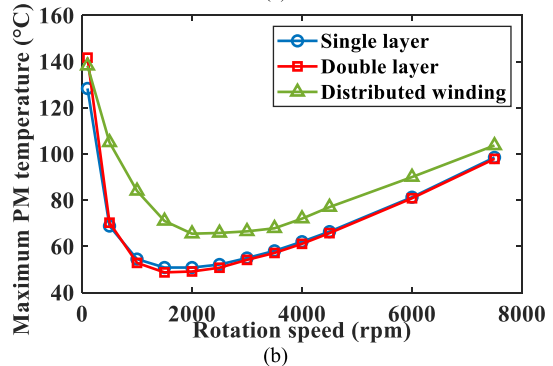
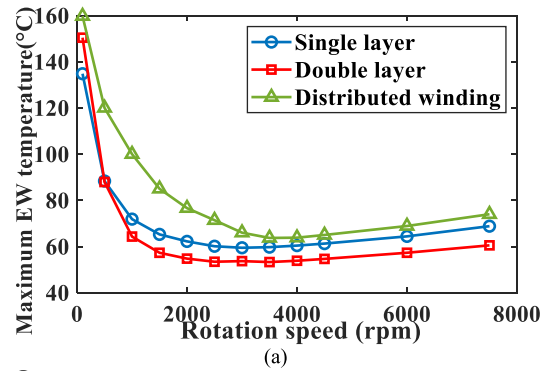


FIGURE 7 | Maximum temperatures versus rotation speed for (a) EW and (b) PM.

compact structure of the double-layer winding hinders natural cooling, resulting in higher temperatures.

In contrast, the distributed winding has a significantly larger EW volume, which leads to higher EW temperatures at low speeds due to greater thermal mass and limited airflow penetration. Moreover, its bulky EW occupies a larger portion of the inlet path, reducing the intake of cold air and diminishing the overall cooling effectiveness. This is reflected by a delayed arrival of the minimum EW temperature, which occurs at 4500 rpm (57.13°C).

In summary, under relatively low-speed conditions (< 3500 rpm), the self-cooling system provides the best EW cooling performance for double-layer windings, followed by single-layer and then distributed windings. In the medium-speed range (3500 rpm < speed < 7000 rpm), the cooling effectiveness ranks as double-layer > distributed > single-layer. At relatively high speeds (> 7000 rpm), the distributed winding shows the best cooling performance, followed by double-layer and then single-layer windings.

The variation of PM temperature with rotational speed for the three winding configurations is shown in Figure 7b. At relatively low speeds (< 3000 rpm), the temperature trend is similar to that of the EW, as PM eddy current losses and iron losses are relatively minor compared to copper losses. However, as the speed increases, PM and iron losses gradually become the dominant heat sources, with a significant portion of the generated heat concentrated in the PM region, resulting in a continuous rise in PM temperature. When comparing the three winding types at speeds above 3000 rpm, the single-layer winding exhibits the highest PM temperature due to a higher level of MMF harmonic content. In contrast, the double-layer winding shows a lower PM temperature, indicating reduced harmonic components and a larger surface-area-to-volume ratio, which enhances its heat dissipation capability. The distributed winding demonstrates the best thermal performance. This is attributed not only to its ability to reduce magnet eddy current losses through reduced armature MMF harmonics—enabled by its winding configuration—but also to the unique structure of its EW, which contributes to more effective heat management.

3.3 | Heat Transfer Coefficient (HTC) Analyses

As shown in Figure 8, the plots illustrate how the HTC of the EW and PM regions vary with rotational speed for the three different winding structures. The HTC of the EW and PM region increases continuously with rising speed. However, the slope of the curve gradually decreases. This is because although higher rotational speed enhances the airflow velocity, the improvement in cooling efficiency exhibits a diminishing return as the velocity increases.

In Figure 8a, the HTC of the double layer structure is consistently higher than that of the single layer, while the distributed winding has the lowest HTC among the three. This is because the double layer has a larger surface area-to-volume ratio, resulting in better heat dissipation performance. Although the distributed winding also has a relatively large surface area-to-volume ratio, its complex EW structure reduces

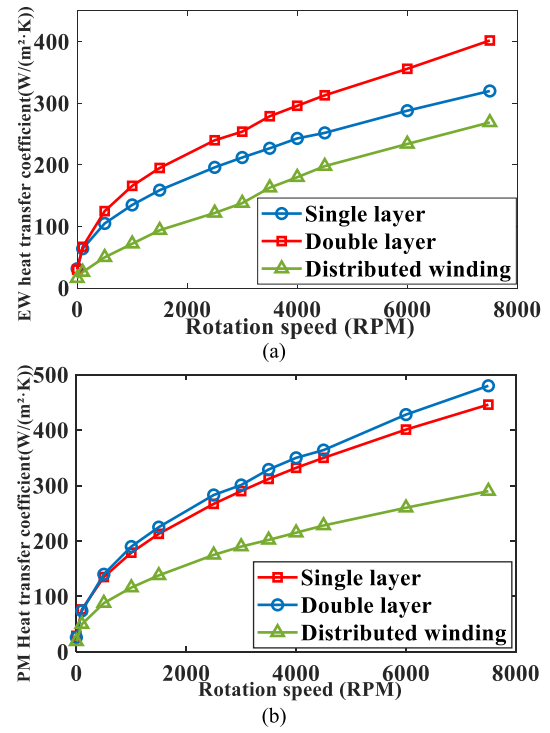


FIGURE 8 | HTC of three winding structures versus rotor speed for (a) EW and (b) PM.

the airflow velocity, leading to a lower HTC. This aligns with the temperature-based conclusions discussed earlier. As for the PM HTC, it primarily depends on the airflow velocity entering the machine. Comparatively, the airflow velocity in the double layer case is slightly higher than in the single layer, while it is the lowest in the distributed winding. Therefore, the HTC follows the trend: double layer > single layer > distributed winding. The HTC values reported here are extracted from the CFD results and are area-averaged over the EW/PM surfaces. Specifically, the local HTC is evaluated from the surface heat flux and the temperature difference, such as

$$h = q'' / (T_{\text{wall}} - T_{\text{ref}}) \quad (11)$$

where q'' is the local wall heat flux [W/m^2], T_{wall} is the local wall temperature, and T_{ref} is the reference air temperature (taken as the bulk/ambient air temperature in the present simulations). The local HTC is then averaged over the corresponding surface. The empirical relationship between HTC and rotational speed can be established as follows.

- For EW

$$\text{HTC}_{\text{EW}} = \begin{cases} 32.5 \cdot \omega^{0.314} & \text{(Single layer)} \\ 37.2 \cdot \omega^{0.329} & \text{(Double layer)} \\ 17.6 \cdot \omega^{0.377} & \text{(Distributed winding)} \end{cases} \quad (12)$$

- for PM

$$\text{HTC}_{\text{PM}} = \begin{cases} 41.2 \cdot \omega^{0.336} & \text{(Single layer)} \\ 41.5 \cdot \omega^{0.349} & \text{(Double layer)} \\ 27.2 \cdot \omega^{0.357} & \text{(Distributed winding)} \end{cases} \quad (13)$$

where ω is the rotor speed (rpm).

The coefficients in Equations (12) and (13) are obtained by curve fitting the CFD-extracted (area-averaged) HTC values over the investigated speed range. Both HTC equations are geometry- and boundary-condition-dependent (e.g., end-region volume, inlet/outlet arrangement, and machine dimensions such as air-gap diameter). Therefore, the fitted coefficients are specific to the investigated machine and should not be directly transferred to other geometries without appropriate assumptions and adjustments. However, the generalisable conclusion is the monotonic increase of HTC with speed with diminishing returns.

3.4 | Cooling Effectiveness Analysis—With Copper Loss Only

In the real case scenarios, although an increase in rotor speed enhances the self-cooling performance by driving greater airflow through the machine, it simultaneously leads to higher magnet eddy current losses and increased stator and rotor core losses. In this case, it will be difficult to tell how much benefit has been gained purely from the self-cooling design. To isolate the influence of the cooling scheme itself and enable a more objective comparison of its effectiveness across different winding structures, the present study imposes a controlled condition in which only copper losses (with the same copper loss density), assumed to be independent of rotational speed, are applied to the windings. By doing so, the observed temperature trends can be primarily attributed to the differences in cooling channel design and winding geometry, thereby allowing a more accurate assessment of structural influence on heat dissipation and offering guidance for further thermal optimization.

As shown in Figure 9, the maximum temperatures of the EW and PM under different winding structures vary with rotational speed when only copper loss is considered. For all three winding types, both EW and PM temperatures decrease as speed increases. A steep temperature drop is observed at lower speeds (< 1500 rpm), followed by a gradual decline in the mid-speed range (1500–5000 rpm), and a nearly flat trend at higher speeds (> 5000 rpm). This trend is primarily attributed to the enhanced self-cooling airflow driven by the rotor blades, which significantly improves convective heat transfer.

Since stator and rotor core losses and PM eddy current loss are excluded in this scenario, the temperature distribution is mainly determined by each winding structure's capacity to guide airflow and dissipate heat. Notably, in Figure 9a, the temperature difference in maximum EW temperature between the distributed winding and concentrated winding (represented by the single-layer structure) shows a non-monotonic trend: it increases initially and then decreases. For instance, at 100 rpm, the maximum temperatures of the distributed and single-layer windings are 181.4°C and 136°C respectively (a difference of 45.4°C). At 500 rpm, the gap widens to 67.6°C (151.3°C vs. 83.7°C), but by 1000 rpm, it narrows to 26.7°C (102°C vs. 75.3°C) and continues to decrease at higher speed.

This behaviour highlights the dominant role of geometric obstruction at low speeds, where the lack of effective airflow

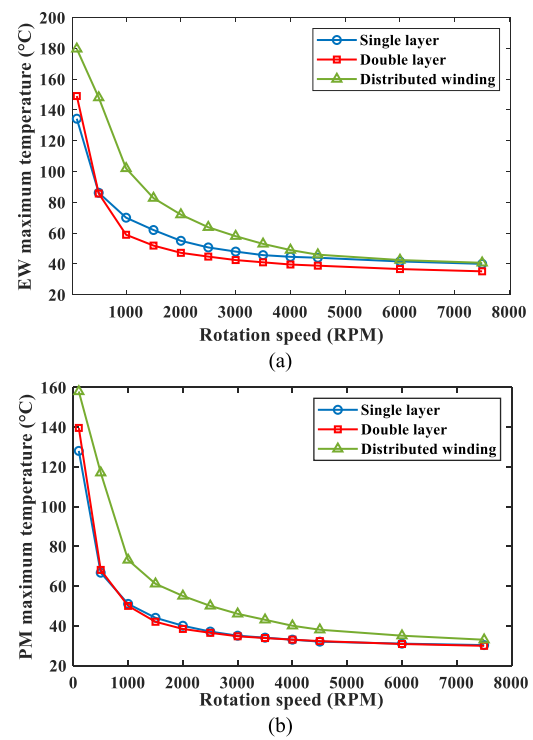


FIGURE 9 | Maximum temperatures with only copper loss as heat source. (a) EW and (b) PM.

limits cooling performance. The distributed winding, due to its large EW volume and complex geometry, significantly impedes airflow at low speed, resulting in heat accumulation and elevated temperature. As the rotational speed increases beyond 500 rpm, the self-induced airflow strengthens and gradually overcomes the flow resistance posed by the distributed winding, improving the cooling performance. In contrast, the concentrated winding exhibits a more stable temperature profile in the mid-to-high speed range, causing the temperature gap between the two to progressively shrink. Therefore, in this self-cooling system, the geometry of the EW structure has a stronger influence on airflow and heat dissipation at low-speed operating conditions.

As a result, when maintaining a constant heat source (identical copper losses), the geometric structure of the double-layer winding proves to be more compatible with the proposed self-cooling scheme. Additionally, a clear distinction emerges between concentrated and distributed windings in terms of their cooling performance under increasing airflow. Specifically, the cooling efficiency of concentrated windings improves significantly with higher airflow, whereas the enhancement for distributed windings is more gradual. However, at high rotational speeds, the maximum temperatures of all three winding structures tend to stabilise, and the differences among them become less pronounced. Therefore, for the relatively small-scale machines investigated in this paper, to fully utilise the benefits of this self-cooling design with distributed windings, relatively high speeds (above 2500 rpm) are required. In contrast, concentrated windings can achieve effective cooling at much lower speeds (above 1000 rpm), making them more suitable for low-to-medium-speed applications.

3.5 | Generalisability and Limitations of the Proposed Study

The conclusions drawn in this study are primarily based on fundamental thermal-fluid and electromagnetic mechanisms, and therefore can be extended to other PM machines employing self-cooling schemes. In particular, the observed relationships between end-winding geometry, airflow resistance, induced mass flow rate, and local heat transfer coefficient are governed by physical principles rather than by machine-specific parameters. As a result, the qualitative trends identified in this work, such as the winding-dependent variation in cooling effectiveness and the competition between enhanced convective heat transfer and increasing electromagnetic losses with speed, are generally applicable to a wide range of self-cooled PM machines. Nevertheless, it is also worth noting that certain quantitative results, including absolute temperature levels and airflow rates, depend on structural and geometric details of the machine, such as the inlet and outlet area, housing clearance, rotor hub dimensions, and blade configuration. Variations in these parameters may shift the operating points at which optimal cooling performance is achieved, but do not alter the underlying mechanisms discussed in this paper.

In addition, the present study intentionally focuses on a baseline self-cooling configuration to enable a controlled comparison of different winding topologies. While the introduction of flow-guiding structures or ducting could potentially mitigate inlet blockage and improve the cooling performance of distributed windings, such modifications would increase design complexity and fundamentally change the cooling configuration. The investigation and optimization of additional flow-guiding features are therefore considered beyond the scope of this paper and are identified as an important direction for future work.

4 | Experimental Validation

4.1 | Prototype Machine and Test Rig

For the experimental setup shown in Figure 10, an existing 12-slot stator [see Figure 11a] was adopted, with key specifications summarised in Table 1. The experimental validation in this work focuses on a relatively low-speed regime (0–1000 rpm) due to hardware limitations, while the high-speed performance is evaluated based on CFD with FE-derived speed-dependent

losses. To replicate the blade geometry, a 3D-printed plastic rotor without PMs [see Figure 11b] was first used as a substitute for the actual rotor, with dimensions adjusted to preserve the airgap length. One of the endcaps with outlets is shown in Figure 11c. Although the 3D-printed rotor does not generate heat, this setup is sufficient to validate the airflow generation (pumping) behaviour and the stator/EW thermal response under the present low-speed test conditions. It should be noted that the low-speed tests (0–1000 rpm) provide a controlled thermal loading where copper loss dominates, enabling a repeatable comparison between winding configurations and a validation of the blade-induced pumping mechanism and the stator/EW-side temperature-speed trend. The high-speed performance and full-loss cases are evaluated using CFD coupled with FE-derived speed-dependent losses and should therefore be interpreted as model-based predictions.

It is worth noting that to avoid any ambiguity regarding the EW geometry, Figure 1c is provided as a schematic illustration of the distributed winding EW arrangement, while the prototype EW geometry in the test rig is shown in Figure 11a. The EW geometry adopted in the CFD model is constructed as a representative envelope to preserve the dominant end-region blockage level and the main flow passages observed in the prototype. Therefore, although the detailed conductor paths are

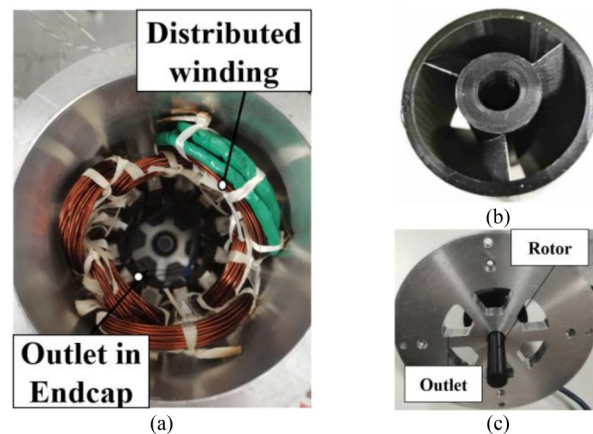


FIGURE 11 | Prototype machine with self-cooling capability. (a) Stator with housing, (b) 3D-printed 3-blade rotor without shaft, and (c) outlet side endcap.

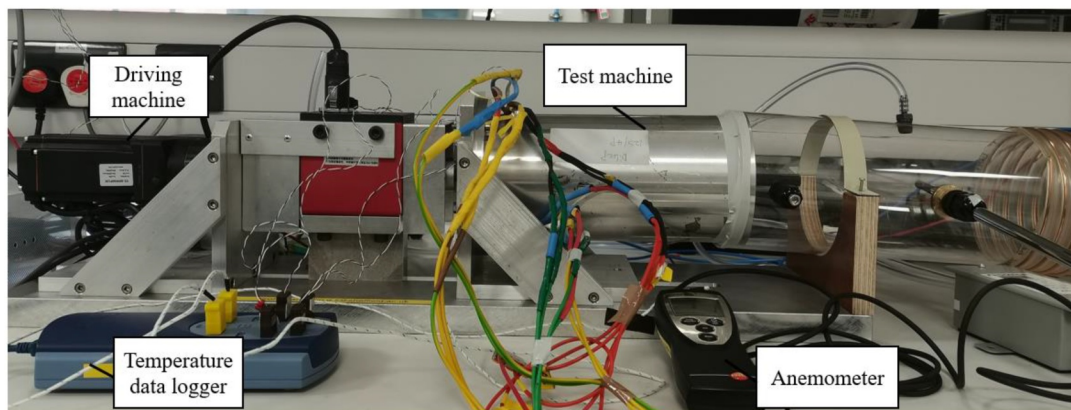


FIGURE 10 | Test rig for the prototype machine with 3D printed plastic rotor and laminated iron-core rotor.

not identical, the CFD model remains representative for validating the global pumping behaviour (outlet flow rate) reported later in Figure 12.

It is also worth noting that, in the present study, the scope of the experimental validation is intentionally focused on the distributed winding machine, with measurements conducted mainly on the EW and active winding temperatures. For the single layer configuration, the experimental data reported in authors' previous work [19] are adopted for comparison. Both sets of experiments were performed on the same 12-slot machine under identical operating and boundary conditions, including supply voltage, current, and test setup. The only distinction between the two cases lies in the winding arrangement, and therefore employing the previously obtained single layer results as a benchmark ensures consistency and enables a fair assessment of the relative cooling performance between different winding structures. In the experimental setup shown in Figure 10, the prototype machine was mechanically coupled to a dynamometer for controlled operation. To capture the thermal behaviour, six thermocouples were mounted on the stator windings, i.e., two for each phase, and three positioned on the active windings and three on the end-windings. Additional thermocouples were affixed to the machine casing and placed in the surrounding environment to record the housing and ambient temperatures, respectively. The outlet flow rate of the machine was obtained by first measuring the mean air velocity at the outlet using a hot-wire anemometer, and subsequently multiplying it by the outlet cross-sectional area. The working principle of the anemometer relies on detecting variations in the electrical resistance of a heated wire caused by the airflow. Temperature measurements of the coils were conducted using T-type thermocouples, with data acquisition performed via a logger. The specific instruments employed were a Testo 425 thermo-anemometer with flow probe for airflow measurements and T-type thermocouples with a temperature range of -270°C – 370°C for temperature monitoring.

4.2 | Measured Results Using 3D Printed Plastic Rotor

As shown in Figure 12, the measured airflow rate of the distributed winding configuration matches well with the simulated results across the entire speed range, thereby validating

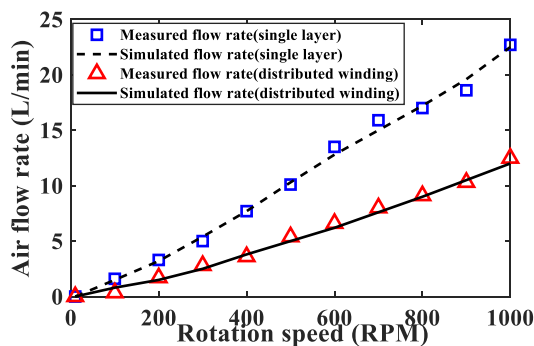


FIGURE 12 | Simulated and measured air flow rates of distributed winding machine. The single layer results in [19] are also added for comparison.

the accuracy of the CFD predictions. In addition, a clear distinction can be observed between the distributed winding and the single layer configuration: the measured airflow rate of the distributed winding is consistently lower, approximately half of that obtained with the single layer winding. This trend is in line with the CFD results presented earlier, confirming that the distributed winding structure inherently generates weaker airflow due to its geometrical characteristics.

Figure 13 compares the measured and CFD-simulated temperatures of the active windings and end windings under different rotor speeds. The overall trends are consistent: both results indicate a monotonic reduction in temperature with increasing speed, confirming that the self-cooling effect is well captured by the simulation model. The deviations between simulation and experiment are relatively small, generally within 5%, which is acceptable given the uncertainties in both numerical modelling and experimental measurement. The discrepancies can be attributed to several factors. First, in the experiment, thermocouples were positioned at discrete locations (e.g., at the inlet side of the EW), which may not perfectly represent the spatially averaged temperatures predicted in the CFD model. Second, the CFD simulation assumes ideal boundary conditions and material properties, whereas the experimental setup inevitably involves additional thermal contact resistances. Third, manufacturing tolerances of the 3D-printed rotor and small variations in airflow paths could also contribute to differences in local cooling effectiveness.

Figure 14 presents a comparison of measured temperatures of the EW and active windings between the single-layer (concentrated winding) [19] and distributed winding configurations under various rotor speeds. The overall temperature trends are consistent with those observed in the earlier case where only copper loss was considered, which is reasonable since the present experiment was also conducted under copper-loss-only conditions. It is evident that the distributed winding exhibits significantly higher temperatures than the single-layer winding across the investigated speed range (0–1000 rpm). Specifically, both the EW and active coil temperatures in the distributed winding are approximately 20°C higher than those of the single-layer winding. In terms of cooling sensitivity, the EW temperature of the distributed winding decreases from 96.3°C at standstill to 83.2°C at 1000 rpm (a reduction of 13.1°C), whereas

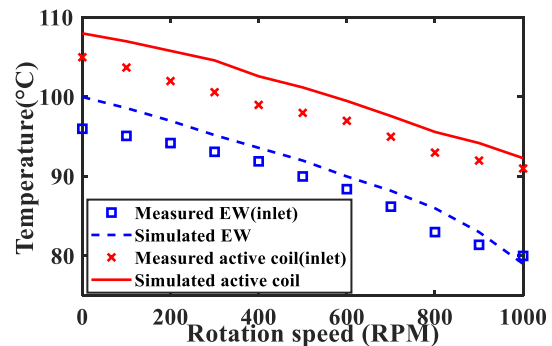


FIGURE 13 | Simulated and measured temperatures of distributed winding machine versus rotor speed. EW stands for end-winding in this figure.

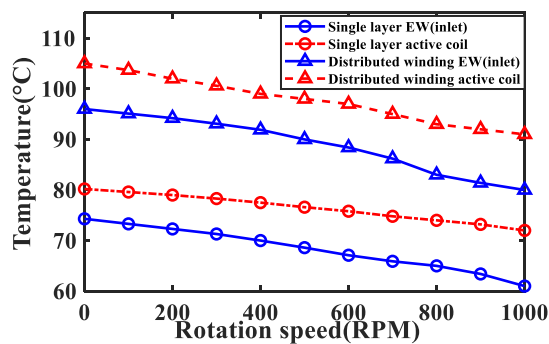


FIGURE 14 | Measured temperature comparison between single layer [19] and distributed windings.

in the single-layer winding, the EW temperature decreases from 75.2°C to 60.1°C (a reduction of 15.1°C). This indicates that the self-cooling scheme is more effective at enhancing heat dissipation in the single-layer winding configuration, particularly at relatively lower speeds. These observations are consistent with the CFD-based analyses presented earlier.

4.3 | Supplementary Validation With Laminated Iron Rotor and Installed PMs

To further consider the influence of rotor material and the presence of PMs on the experimental validation, a 12-slot/8-pole laminated iron rotor prototype was manufactured, while retaining the same rotor-hub fan-blade self-cooling concept. It is worth noting that this iron-rotor prototype adopts a 12-slot/8-pole rotor to demonstrate the applicability of the proposed self-cooling concept with a realistic metallic rotor and installed PMs; it is not intended as a strict one-to-one validation of the electromagnetic loss levels for the original study cases.

As illustrated in Figure 15a,b, 0.35 mm FeSi laminates were cut using wire eroder and stacked onto a 3D-printed assembly mould to form the designed fan blade shape rotor hub. The stack was then secured by laser welding on the outer surface to ensure mechanical integrity. The resulting laminated rotor hub with integrated fan blades and the rotor assembly with installed PMs are shown in Figure 15c,d, respectively.

Using the same test rig and measurement procedures, a series of tests were carried out and the EW and active-winding temperatures were recorded for both winding configurations. Figure 16 compares the measured winding temperatures obtained using the plastic rotor (PR) and the laminated iron rotor (IR). As can be seen, for both the single-layer and distributed windings, the temperatures decrease monotonically with increasing speed, which is consistent with the increased pumping capability of the integrated fan blades and the associated enhancement of convective heat transfer in the end region. Compared with the PR case, the IR case exhibits consistently higher temperatures over the whole speed range. This is expected because the laminated iron rotor provides a more realistic rotor-side thermal conduction path, and the installed PMs allow magnet-related losses to be included in the overall thermal balance.

The temperature difference between IR and PR is moderate and remains within a few degrees at the upper end of the tested speed

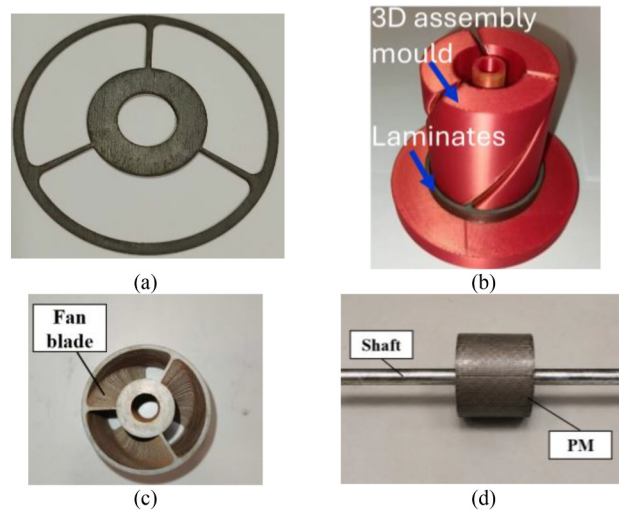


FIGURE 15 | Iron-core rotor prototyping. (a) 0.35 mm FeSi laminate, (b) 3D-printed assembly mould used to align the laminates and form the designed rotor-hub passages during laminates stacking, (c) laminated iron rotor hub with integrated fan blades, and (d) assembled rotor on the shaft with installed PMs.

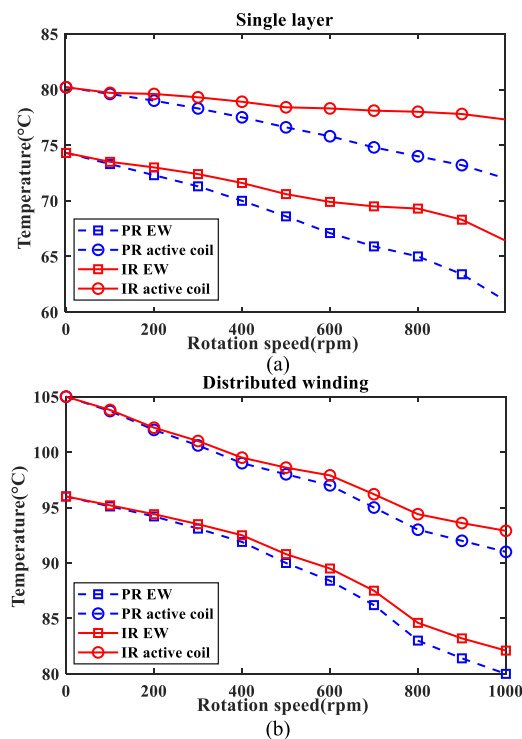


FIGURE 16 | Measured winding temperatures versus speed using plastic rotor (PR) and iron rotor (IR). (a) Single-layer winding and (b) distributed winding. EW stands for end winding and AW stands for active winding.

range (1000 rpm). For example, at 1000 rpm, the IR case increases the EW and active-winding temperatures by 5.3°C for the single-layer winding, while the corresponding increase is 1.9°C for the distributed winding. This difference is also consistent with the lower airflow rate of the distributed winding configuration, where the winding temperatures are dominated by copper loss and the relative contribution of additional losses is smaller. Overall, the

results in Figure 16 confirm that the temperature–speed trends and the relative ranking between winding configurations remain unchanged when a laminated iron-core rotor (with installed PMs) is used, thereby providing additional experimental evidence beyond the plastic-rotor validation.

5 | Conclusion

This paper compares the cooling efficiency of a self-cooling scheme, incorporating fan blades to the rotor hub region (space between rotor iron core and the shaft), across three typical winding structures such as concentrated single- and double-layer windings as well as distributed winding of SPM machines. To analyse the temperature at the primary heat sources, namely the EW and PM, CFD modelling is employed. Results such as temperature contour maps of the EW and PM, as well as air velocity vector diagrams, were analysed. The cooling performance variation of this self-cooling scheme with different winding structures was investigated as a function of rotor speed. As the rotor speed increases, the resulting higher electromagnetic losses partially offset the benefits of enhanced self-cooling, causing the temperatures of both the EW and PM to rise—initially sharply, then more gradually. Based on this analysis, the HTC was calculated, and empirical formula were derived to describe the relationship between HTC and rotor speed, revealing a nonlinear but predictable relationship. In addition to evaluating the real loss case thermal behaviour, this study also conducts comparative simulations under unified loss conditions (the same copper loss density for all windings) to isolate and better investigate the influence of structural differences.

The results demonstrate that, overall, concentrated windings benefit more from the proposed self-cooling scheme compared to distributed winding. Within the concentrated winding group, the double-layer winding exhibits superior thermal performance relative to the single-layer winding. In contrast, the distributed winding shows only a limited improvement in cooling effectiveness at low speeds. These conclusions are further supported by experimental measurements, which confirm the validity of the numerical predictions.

Rather than introducing additional cooling hardware, this study demonstrates that winding topology alone plays a critical role in determining the effectiveness and limitations of self-cooling in compact PM machines. These findings provide design-oriented insights into the thermal design of high-performance machines with self-cooling capability and highlight the importance of considering winding structure at an early design stage. The investigation of long-term reliability and transient thermal behaviour under time-varying operating conditions is beyond the scope of this study and will be addressed in future work.

Author Contributions

Kaile Li: writing – original draft. **Guang-Jin Li:** writing – review and editing.

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Conflicts of Interest

The authors declare no conflicts of interest.

Data Availability Statement

Data available on request from the authors.

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