

A Lumped Parameter Thermal Network Model for Dual-Stator Mover-PM Linear-Rotary Generators

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Abstract – Mover-permanent magnet linear-rotary generators (MPMLRGs) with dual-stator topology and two degrees of freedom (2-DOF) have attracted growing interest due to their high integration and compactness, enabling coupled linear-rotary motion for direct-drive energy conversion and multifunctional actuation systems. However, the coupled linear-rotary motion and unsymmetric dual-stator topology lead to complex thermal behaviors, and the associated thermal characteristics remain insufficiently investigated compared to electromagnetic studies. In this paper, a lumped-parameter thermal network (LPTN) model is developed for a dual-stator MPMLRG to accurately capture its asymmetric heat transfer and temperature rise under helical motion. A novel approach is proposed to determine the air-gap convective heat transfer coefficients under coupled linear-rotary operation, along with a time-weighted correction method to account for the partial axial overlap of stator-mover surfaces. The thermal model is validated against 3-D thermal finite element analysis (FEA), showing good agreement while significantly reducing computational effort. The results provide useful insights into the thermal management of the dual-stator MPMLRG and establish a foundation for future design optimization and cooling enhancement.

Index Terms – Dual-stator topology, lumped-parameter thermal network, permanent magnet linear-rotary generator, thermal analysis.

I. INTRODUCTION

Permanent-magnet linear-rotary generators (PMLRGs) have attracted increasing attention in compact directdrive electromechanical conversion systems, such as wind-wave combined energy conversion (WWCEC), advanced manufacturing, and robotic actuation [1–5]. Linear-rotary integration enables high torque/thrust density with reduced transmission

complexity, which is particularly attractive for marine energy applications requiring high integration and reliability [6]. However, offshore environments are typically characterized by strong reciprocating loads, high humidity and salinity, and limited cooling capability, where excessive temperature rise may cause irreversible permanent-magnet demagnetization and accelerated insulation aging, thus threatening long-term operational reliability [7, 8]. Therefore, thermal modeling and temperature prediction are critical for mover-permanent magnet linear-rotary generators (MPMLRGs) design and safe operation.

Although the electromagnetic performance of PMLRGs has been extensively investigated, studies on their thermal behaviors remain relatively limited. Representative configurations, such as dual-stator and dual-rotor PMLRGs employing alternately arranged toroidal and arc-shaped permanent magnets, have been reported in [9, 10], demonstrating the feasibility of compact linear-rotary integration. However, for MPMLRGs, the coupled helical motion and asymmetric structure lead to time-varying air-gap convection and nonuniform heat-transfer paths, posing significant challenges to accurate temperature prediction and thermal design.

Thermal analysis of electrical machines is commonly performed using computational fluid dynamics (CFD), the finite element method (FEM), and lumped-parameter thermal network (LPTN) models [11–13]. CFD is often used to evaluate convective heat transfer coefficients, while FEM provides accurate temperature distributions but is computationally expensive for rapid design iterations and optimization studies [14]. In contrast, LPTN offers a practical balance between accuracy and efficiency and has been widely used for fast thermal evaluation. Most existing LPTN studies, however, focus on conventional single degree-of-freedom (1-DOF) rotary or linear machines, and their extension to two degrees of freedom (2-DOF) linear-rotary generators (LRGs) has rarely been addressed. For instance,

equivalent thermal networks have been established for various machine types, including flux-switching permanent-magnet double-rotor machines [15], brushless doubly-fed generators [16], surface-mounted and interior permanent magnet synchronous motors [17], oil-cooled segmented-stator machines [18], and dual-radial-ventilated permanent magnet synchronous generators [19]. Despite these advances, an LPTN framework that explicitly captures motion-coupled air-gap convection and asymmetric overlap-induced heat-transfer variation for dual-stator MPMLRG is still lacking. Different from conventional air-gap thermal models that assume pure rotational motion and full axial overlap, the proposed LPTN explicitly incorporates the coupled linear-rotary kinematics and the timevarying effective air-gap exposure unique to dual-stator MPMLRG.

To bridge this gap, this paper develops an LPTN model for a dual-stator MPMLRG that explicitly accounts for coupled helical motion and asymmetric air-gap geometry. In particular, the proposed model improves temperature prediction accuracy by incorporating motion-coupled air-gap convection and the effective heat-transfer variation induced by partial axial overlap between the mover and stator surfaces. The main contributions of this work are summarized as follows:

1. A method is proposed to determine air-gap convective heat transfer coefficients under coupled helical motion by extending classical empirical correlations to the 2-DOF operating condition.
2. A time-weighted correction approach is introduced to account for the partial axial overlap between the mover and stator surfaces caused by structural asymmetry.
3. The proposed LPTN model is validated against 3-D transient thermal finite element analysis (FEA), demonstrating close agreement with significantly reduced computation time.

The remainder of this paper is organized as follows. Section II introduces the machine structure and material data. Section III presents the loss models. Section IV describes the LPTN modeling process, including the proposed air-gap heat transfer coefficient calculation and timeweighted correction method. Section V presents the LPTN results and the 3-D FEA verification. Section VI concludes the paper and discusses potential future work.

II. TOPOLOGY OF THE DUAL-STATOR MPMLRG

A. Structure

The configuration of the dual-stator MPMLRG is shown in Fig. 1. The machine consists of an outer stator, a shared mover, and an inner stator, enabling integrated

rotary and linear energy conversion in a compact structure. Permanent magnets are mounted on both the outer and inner surfaces of the mover, classifying the machine as a moverpermanent-magnet topology.

The outer stator and mover outer surface form the rotary generation unit (RGU), while the inner stator and mover inner surface constitute the linear generation unit (LGU). Both units adopt a 15-slot/20-pole configuration and are mechanically coupled through the common mover yoke. Concentrated windings are employed in the outer stator for rotary generation, whereas ring-shaped windings are arranged in the inner stator for linear generation. Correspondingly, arc-shaped and ring-shaped permanent magnets are radially magnetized with alternating polarities on the mover surfaces.

This dual-armature, dual-winding architecture provides improved integration, enhanced fault tolerance, and flexible control capability while maintaining structural compactness. The key geometric and electrical parameters are summarized in Table 1.

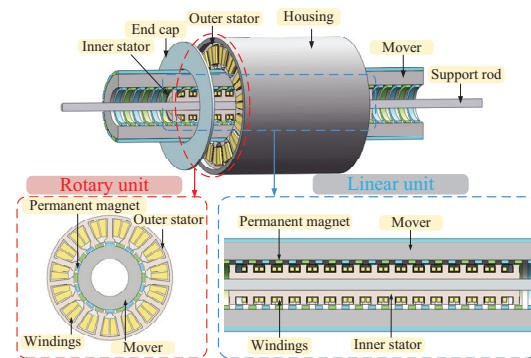


Fig. 1. Structure of the MPMLRG.

B. Materials

The material properties of the MPMLRG significantly affect both electromagnetic and thermal behaviors. The permanent magnets are N48, and the stator and mover cores are made of laminated silicon steel 50W470. Both the RGU and LGU stator cores employ axially laminated silicon steel sheets. Although axial lamination may not be optimal for all magnetic flux paths in tubular linear machines, it is adopted in this study, and its influence on the thermal model is reflected through the anisotropic thermal conductivities assigned to the laminated cores. A discussion on the influence of lamination orientation in tubular linear machines can be found in [20].

Thermal modeling of the stator slots is nontrivial due to the composite winding region. Therefore, an equivalent thermal conductivity approach is adopted to represent the anisotropic heat conduction of the slot

Table 1: Key parameters of the MPMLRG

	Parameter	Symbol	Value
RGU	Outer diameter of outer stator	D_{so1}	150 mm
	Inner diameter of outer stator	D_{si1}	86 mm
	Outer diameter of mover	D_{mo}	84 mm
	Length of outer stator	l_{os}	150 mm
	Outer stator pole height	h_{sp1}	22.5 mm
	Outer stator yoke height	h_{sy1}	3.7 mm
	Air-gap thickness	$l_{air-gap 1}$	1 mm
	Rated speed	ω	300 rpm
	Number of turns per phase	N_{ph1}	175($d = 2.0$ mm)
	Resistance of phase winding	R_{ph1}	0.468 Ω
LGU	Outer diameter of inner stator	D_{so2}	36 mm
	Inner diameter of inner stator	D_{si2}	10 mm
	Inner diameter of mover	D_{mi}	39 mm
	Length of inner stator	l_{is}	262.5 mm
	Inner stator pole height	h_{sp2}	6 mm
	Inner stator yoke height	h_{sy2}	22 mm
	Air-gap thickness	$l_{air-gap2}$	1.5 mm
	Rated speed	v	1 m/s
	Number of turns per phase	N_{ph2}	125($d = 1.5$ mm)
	Resistance of phase winding	R_{ph2}	0.167 Ω

Table 2: Material thermal data

Model Region	Thermal Conductivity (W/(m·K))			Density (kg/m ³)	Specific Heat (J/(kg·K))
	Direction X	Direction Y	Direction Z		
Stator slots of RGU	0.23	0.23	276.06	5514.34	405.9
Stator slots of LGU	273.05	273.05	0.19	6411.22	403
Permanent magnet	9	9	7	7500	400
Laminated core	35	35	2	7650	450
Support rod	47	47	47	7850	470
Housing	90	90	90	2700	900
End cap	0.3	0.3	0.3	920	1900

Note: X, Y, and Z denote the local radial, circumferential, and axial directions, respectively.

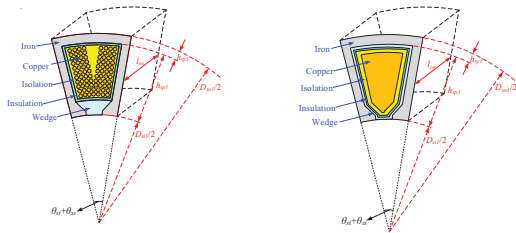


Fig. 2. Dimensions of the RGU stator slot: (a) individual slot geometry and (b) equivalent slot model.

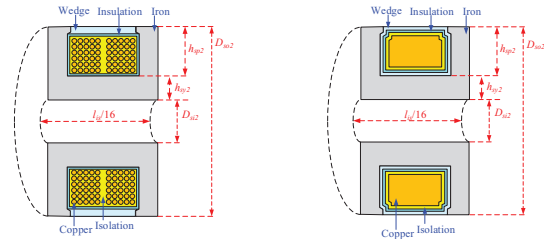


Fig. 3. Dimensions of the LGU stator slot: (a) individual slot geometry and (b) equivalent slot model.

region. As shown in Figs. 2 and 3, the slot region is homogenized by combining the copper conductors, insulation, impregnation materials, wedges, and

slot fillers into an equivalent anisotropic medium. The homogenized slot regions correspond to the equivalent stator-slot materials used in the thermal model. Owing

to the different winding configurations, the RGU and LGU stator slots exhibit different principal thermal conductivity directions. The RGU winding is mainly arranged along the axial direction, whereas the annular winding of the LGU provides higher equivalent thermal conductivity within the plane of the winding layers. The equivalent conductivities are determined from the slot geometry, material fractions, and a standard series model. Consequently, the detailed material distribution inside each slot is represented by an equivalent homogeneous anisotropic region in the thermal model.

The thermal conductivities, densities, and specific heats used in the transient thermal analysis are summarized in Table 2.

III. LOSS MODELS

Accurate loss estimation is essential for thermal analysis because the losses act as heat sources. In this section, the losses of the MPMLRG under different operating conditions are analyzed using both FEM and analytical methods.

A. Copper loss

The MPMLRG operates at relatively low electrical frequencies (50 Hz for RGU and 41.67 Hz for LGU under rated conditions). Therefore, only DC resistance is considered, and copper loss is calculated as follows:

$$P = mI_{ph}^2 R_{ph}, \quad (1)$$

where m is phase number, I_{ph} and R_{ph} are phase current (RMS) and phase resistance, respectively.

B. Core loss

According to the Steinmetz formula, the core loss can be expressed as [21]:

$$P_{FeH} = \int_{V_c} \left(\sum_{n=1}^N C_H (n\omega_s) B_n^2 \right) dV, \quad (2)$$

$$P_{FeE} = \int_{V_c} \left(\sum_{n=1}^N C_E (n\omega_s)^2 B_n^2 \right) dV, \quad (3)$$

where P_{FeH} and P_{FeE} represent the hysteresis and eddy-current losses (W/kg), V_c is the volume of the laminated iron core, B_n is the amplitude of the n th harmonic flux density, N is the total number of harmonics, ω_s denotes the fundamental electrical angular frequency (rad/s), the angular frequency of the n th harmonic is $n\omega_s$, and C_H and C_E are the hysteresis and eddy-current loss coefficients. In this study, $C_H = 183.674$ and $C_E = 1.224$, which were obtained by nonlinear fitting of the loss data for 50W470 silicon steel.

C. Permanent magnet eddy current loss

The magnetic field in the MPMLRG can be divided into two components: the main field, generated by the mover-mounted permanent magnets, which rotates and translates synchronously with the mover and is therefore approximately stationary in the permanent magnet reference frame, contributing negligibly to the permanent magnet eddy-current loss; and the harmonic field, which lacks such synchronism and consequently induces electromotive force (EMF), eddy currents, and associated losses in the permanent magnets of both the RGU and the LGU. The eddy-current loss in the mover-permanent magnets, resulting from harmonic components of the air-gap magnetic field, is quantified by (4) over one alternating-magnetization period, as described in [22]:

$$P_e = \frac{1}{T_e} \int_0^{T_e} \sum_{e=1}^{N_e} J_e^2 \frac{1}{\sigma_r} \Delta_e l_a dt \quad (4)$$

where P_e denotes the rotor eddy current loss [W], J_e is the current density in each element [A/m²], Δ_e is the element area [m²], l_a is the mover axial length [m], σ_r is the conductivity of the eddy current zone [S/m], and T_e is the time corresponding to an alternating magnetization period in each element.

D. Heat generation

To evaluate the thermal behavior of the 2-DOF MPMLRG under helical motion, five representative operating cases are considered by simultaneously varying the rotational and translational speeds of the mover around the rated condition. The volumetric heat generation rates of major components are given in Table 3. Case 1 corresponds to the rated speeds of 300 rpm and 1.0 m/s, while in Cases 2–5 the rotational speed is increased in steps of 50 rpm and the linear speed in steps of 0.2 m/s. For all cases, the RGU and LGU load resistances are set to 5 Ω and 1 Ω , respectively. The resulting losses are converted into volumetric heat generation rates and applied as heat sources in the LPTN and FEA models.

Table 3: Volumetric heat generation rates of major components

Motor Part	Heat Generation (W/m ³)				
	Case 1	Case 2	Case 3	Case 4	Case 5
Stator of RGU	10513.53	12085.01	13861.75	15418.34	16588.97
Winding of RGU	165496.42	201620.49	241031.21	266156.82	290465.48
PM of RGU	2437.40	3335.63	4595.64	5772.04	6974.32
Mover	42.49	59.29	80.92	100.64	108.76
PM of LGU	670.24	961.91	1028.13	1060.38	1066.33
Stator of LGU	207225.95	302137.07	415043.91	560906.21	692365.80
Winding of LGU	39518.48	55733.89	75263.31	99711.47	124060.68

IV. LUMPED PARAMETER THERMAL NETWORK MODEL

To facilitate numerical analysis, the motor structure is discretized into thermal nodes based on material properties and geometry. Thermal resistances between nodes are calculated according to their spatial arrangements and heat pathways. Losses from components are proportionally distributed to corresponding nodes. Steady-state and transient heat transfer equations are formulated and solved using MATLAB to obtain the temperature at each node.

To simplify the three-dimensional temperature field while maintaining sufficient accuracy, the following assumptions are adopted: (i) circumferential temperature symmetry is assumed; (ii) heat transfer is dominated by conduction, whereas radiation and skin effects are neglected; and (iii) the short end windings of the RGU are ignored.

Based on these assumptions, the machine is partitioned into representative thermal regions, each modeled as a thermal node located at its geometric center. Heat exchange between nodes is represented by thermal resistances. The resulting LPTN is illustrated in Fig. 4, where nodes are assigned to the stator yoke, stator slots, stator teeth, permanent magnets, mover yoke, air gaps, support rod, housing, end caps, and internal air regions of both the RGU and LGU.

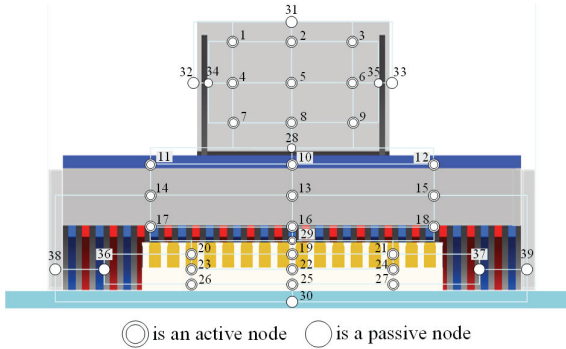


Fig. 4. Key thermal nodes of the MPMLRG cross-section.

A. Steady-state and transient thermal network method

The steady-state thermal behavior of the motor is modeled by formulating a linear algebraic system derived from the nodal energy balance equations. In this framework, the motor is discretized into multiple thermal nodes, each corresponding to a specific structural region. The inter-nodal heat conduction is captured by a global thermal conductance matrix G_s [23], constructed as the inverse of the thermal resistances between physically connected nodes. The internal heat generation due to electromagnetic losses is represented

by a source vector Q_s , uniformly distributed across the corresponding nodes. The steady-state temperature distribution T_s is then obtained by solving:

$$G_s \cdot T_s = Q_s, \quad (5)$$

where T_s denotes the vector of steady-state nodal temperatures. Owing to the localized heat-transfer paths, G_s is symmetric and sparse, and its conductance topology (Fig. 5) provides an efficient representation of the steady-state thermal behavior of the MPMLRG.

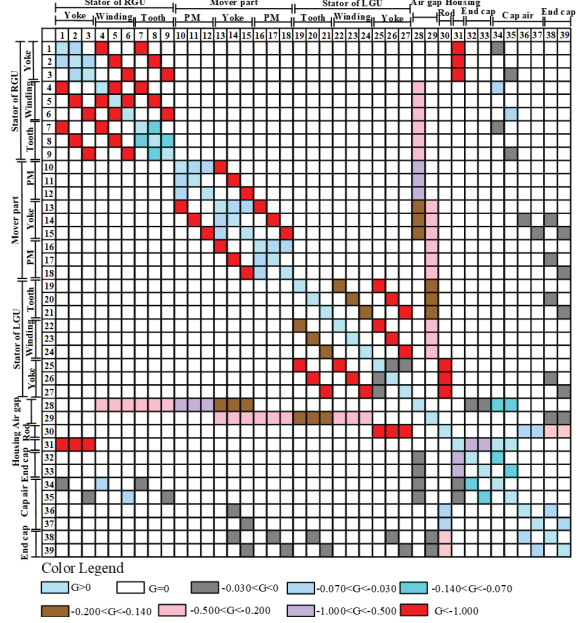


Fig. 5. Thermal conductance network between different components.

The transient thermal model describes the time-dependent temperature evolution by incorporating both internodal thermal conductance and thermal capacitance, the latter determined by material density, specific heat, and volume. The resulting dynamic thermal behavior is governed by the following first-order differential equation:

$$C_t \cdot \frac{dT_t}{dt} + G_t \cdot T_t = Q_t(t), \quad (6)$$

where C_t is the diagonal thermal capacitance matrix, G_t is the thermal conductance matrix, $T_t(t)$ represents the instantaneous nodal temperature vector, and $Q_t(t)$ denotes the time-dependent heat generation vector. This formulation captures both heat diffusion across components and energy accumulation due to thermal inertia.

B. Thermal resistance and thermal capacitance

Thermal resistance in electric machines can be categorized into two types: conduction thermal resistance

and convection thermal resistance. Conduction thermal resistance occurs within the solid components of the machine. As shown in Fig. 6, based on the fundamental law of heat conduction, the thermal conduction resistances of the two models are derived as follows [24].

The thermal resistance of the flat plate conduction model is given by:

$$R = \frac{\Delta T}{Q} = \frac{L}{\lambda S}. \quad (7)$$

The thermal resistance of the cylindrical conduction model is given by:

$$R = \frac{\Delta T}{Q} = \int_{r_1}^{r_2} \frac{dr}{2\lambda\pi rL} = \frac{1}{2\pi\lambda L} \ln \frac{r_2}{r_1}, \quad (8)$$

where L is the length along the direction of heat flow [m], λ denotes the thermal conductivity of the material [W/(m·K)], and S represents the cross-sectional area for heat conduction [m²].

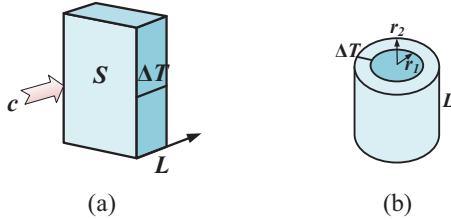


Fig. 6. Two types of conduction resistance models.

Thermal capacitance C can be given as:

$$C = mc_p, \quad (9)$$

where m denotes body mass and c_p is specific heat capacity.

As shown in Fig. 7, the nodes denote different components of the MPMLRG interconnected along the actual heat-transfer paths to establish the simplified LPTN. Based on well-established lumped-parameter thermal modeling theory, the thermal resistances and thermal capacitances of each node are determined from the corresponding geometry and material properties, and only the general formulation is presented here for conciseness.

C. Air gap heat transfer coefficient

The convective heat transfer within the air gap is evaluated through the following approach. For unified expression, the outer and inner boundaries of each cylindrical air gap are denoted as r_1 and r_2 . Initially, the air-gap Taylor number Ta_g , defined using the mean radius $r_g = (r_1 + r_2)/2$, and the geometry factor F_g [25] are calculated by:

$$Ta_g = \omega r_g^{0.5} (r_1 - r_2)^{1.5} / \nu_{air}, \quad (10)$$

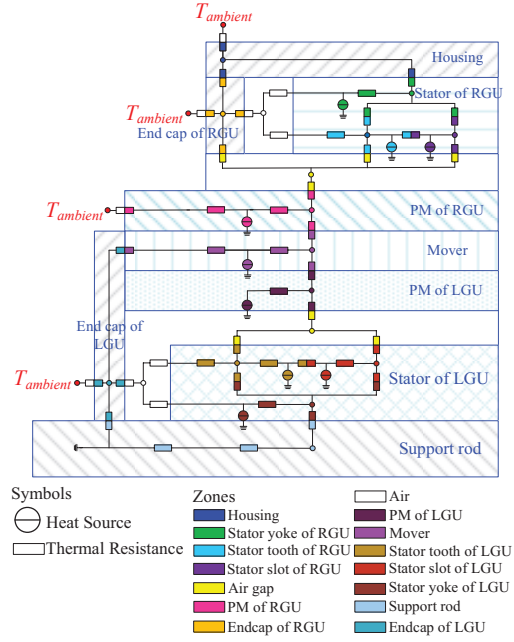


Fig. 7. Simplified LPTN for the MPMLRG.

and

$$F_g = \frac{\pi^2}{41.19\sqrt{S}} \left(1 - \frac{r_1 - r_2}{2r_g} \right)^{-1}, \quad (11)$$

where ω denotes the mover's angular velocity and ν_{air} is the kinematic viscosity of air. The structural parameter S is given by:

$$S = 0.0571 \times \left[1 - 0.652 \times \frac{(r_1 - r_2)/r_g}{1 - (r_1 - r_2)/(2r_g)} \right] + 0.00056 \times \left[1 - 0.652 \times \frac{(r_1 - r_2)/r_g}{1 - (r_1 - r_2)/(2r_g)} \right]^{-1}. \quad (12)$$

Subsequently, the air-gap Nusselt number Nu_g is obtained as a function of $(Ta_g/F_g)^2$, as shown in (13). When $(Ta_g/F_g)^2 < 1700$, thermal convection is negligible, and heat transfer across the air gap is dominated by conduction. For $1700 < (Ta_g/F_g)^2 < 10^4$, the flow in the gap becomes laminar with vortex formation, while for $10^4 < (Ta_g/F_g)^2 < 10^7$, turbulent flow governs the convective heat exchange.

$$Nu_g = \begin{cases} \frac{2(r_1 - r_2)/r_{ro}}{\ln[1 + (r_1 - r_2)/r_{ro}]}, & \text{for } \frac{Ta_g^2}{F_g^2} < 1700 \\ 0.128(Ta_g^2/F_g^2)^{0.367}, & \text{for } 1700 < \frac{Ta_g^2}{F_g^2} < 10^4 \\ 0.409(Ta_g^2/F_g^2)^{0.241}, & \text{for } 10^4 < \frac{Ta_g^2}{F_g^2} < 10^7. \end{cases} \quad (13)$$

Based on the calculated Nusselt number, the air-gap convective heat transfer coefficient h_g can be derived accordingly as:

$$h_g = k_{air} \frac{Nu_g}{2g}. \quad (14)$$

Referring to (11) and the dimensional parameters listed in Table 1, the geometry factor of outer and inner airgap for the analyzed MPMLRG at rated speed is calculated to be $F_g^{(RGU)} = 1.01190 \approx 1$ and $F_g^{(LGU)} = 1.041667 \approx 1$, respectively. The air's kinematic viscosity is taken as $\nu_{air} = 1.5 \times 10^{-5}$ and the air-gap thermal conductivity is $k_{air} = 0.023 \text{ W/(m} \cdot \text{K)}$. According to (10), the corresponding air-gap Taylor numbers under the operating condition of a mover speed of $n = 300 \text{ rpm}$ are calculated as: $Ta_g^{(RGU)} = 13.65382$, $Ta_g^{(LGU)} = 16.66085$.

Since $(Ta_g^{(RGU)})^2 / (F_g^{(RGU)})^2 \approx 180 < 1700$, and $(Ta_g^{(LGU)})^2 / (F_g^{(LGU)})^2 \approx 243 < 1700$, heat transfer within the air gap is dominated by thermal conduction rather than convection. As a result, the air-gap convective heat transfer coefficient is calculated as: $h_g^{(RGU)} = 46.54556 \text{ W/(m}^2 \cdot \text{K)}$, $h_g^{(LGU)} = 31.92737 \text{ W/(m}^2 \cdot \text{K)}$. In this regime, $Nu_g \approx 1$, and the resulting h_g represents the conduction-dominated equivalent heat-transfer coefficient across the air gap (i.e., $h_g \approx k_{air}/\delta$). It should be noted that this formulation represents an equivalent heat-transfer coefficient across the air gap, which is commonly adopted in electrical machine thermal modeling when the air flow remains weak.

By substituting the thresholds $(Ta_g^{(RGU)})^2 / (F_g^{(RGU)})^2$ and $(Ta_g^{(LGU)})^2 / (F_g^{(LGU)})^2 = 1700, 10^4$, and 10^7 into (13), the corresponding boundary mover speeds for the RGU and LGU are determined to be 916, 2223, and 70308 rpm, and 793, 1923, and 60828 rpm, respectively, which indicates that the analyzed operating range is well within the conduction-dominated regime.

However, in the case of MPMLRG, the mover undergoes both rotational and translational motions simultaneously, resulting in a helical trajectory. To more accurately represent the convective flow driven by such compound motion, an effective air-gap velocity can be defined as:

$$u_{eff} = \sqrt{u_\theta^2 + u_z^2}, \quad (15)$$

where $u_\theta = \omega r_{gap}$ is the tangential velocity and u_z is the linear velocity of the mover. According to forced convection theory, the convective heat transfer coefficient is proportional to velocity raised to a power n , typically $n = 0.8$ for turbulent forced convection of gases:

$$h \propto u^n. \quad (16)$$

Consequently, the original convective coefficient can be corrected as:

$$h_{g,new} = h_g \cdot \left(\frac{u_{eff}}{u_\theta} \right)^{0.8}. \quad (17)$$

This correction allows the model to incorporate the enhancement in heat transfer caused by the superimposed translational motion of the mover.

To further explore the influence of motion parameters on air-gap heat transfer performance, the same five operating conditions (Cases 1–5) introduced in section III part D are adopted. These include the rated operating condition (Case 1) and four additional cases with increased rotational and linear speeds. In each case, the corresponding Taylor number and Nusselt number are calculated, and the convective heat transfer coefficients of both the air gaps of RGU and LGU are evaluated accordingly.

As shown in Fig. 8, the convective heat transfer coefficients of both air gaps increase with speed due to enhanced forced convection.

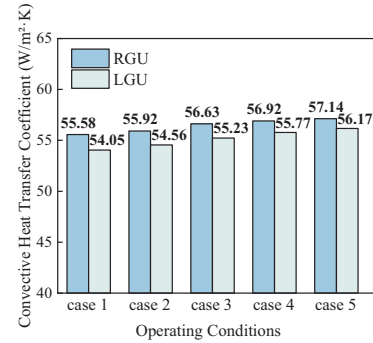


Fig. 8. Air-gap convective heat transfer coefficients.

Due to the dual-stator configuration of the MPMLRG, the outer and inner stators have axial lengths of 150 mm and 262.5 mm, respectively, whereas the mover spans a total of 400 mm. During operation, the mover undergoes a compound motion consisting of rotation and linear translation, resulting in a helical trajectory along the axial direction. Because of the mismatch in axial dimensions, not all portions of the mover surface are continuously enclosed by the stator cores. This partial air-gap exposure is illustrated schematically in Fig. 9, where the helical motion path and bidirectional travel of the mover are shown. Specifically, only a segment of the mover's outer surface forms an active air gap with the outer stator, while the rest is exposed to ambient air. A similar condition applies to the inner air gap.

To accurately represent this intermittent contact during motion, the convective heat transfer coefficients

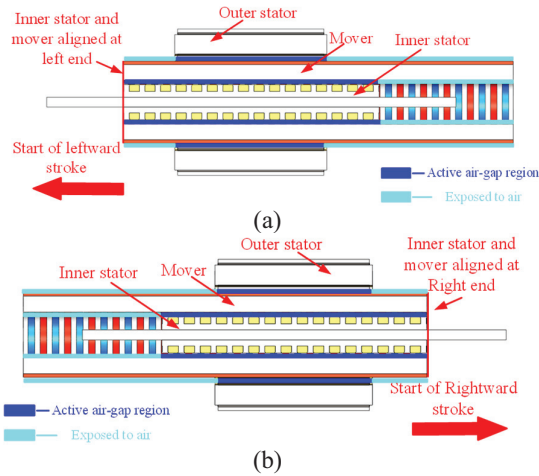


Fig. 9. Relative positions between the mover and stator during reciprocating motion: (a) left stroke limit and (b) right stroke limit.

of the inner and outer air gaps are modified using a time-weighted approach, expressed as:

$$\bar{h}_g = \eta \cdot h_g + (1 - \eta) \cdot h_{air}, \quad (18)$$

where η is the effective contact ratio determined by the axial coverage of the stator relative to the mover and h_{air} is the natural convection heat transfer coefficient for the exposed surfaces. This method ensures the thermal model reflects realistic boundary conditions during coupled motion.

V. LPTN MODEL SOLVING AND 3-D FEA VERIFICATION

The proposed LPTN model predicts the average temperature of lumped components rather than detailed spatial distributions. Therefore, a 3-D thermal FEA model is established to obtain the temperature field and to validate the accuracy of the LPTN predictions. The heat transfer coefficients used in the FEA are consistent with those derived from the thermal network modeling in section IV.

A. Establishment of finite element model

The temperature distributions obtained from the 3-D thermal model implemented in ANSYS Workbench are shown in Fig. 10. The ambient temperature is set to 22°C.

At the rated operating condition, thermal behaviors are observed in the RGU and LGU. In the RGU, the stator iron and windings exhibit the highest temperatures, whereas the mover and permanent magnets remain significantly cooler due to partial exposure to ambient air. Similarly, in the LGU, the stator and windings

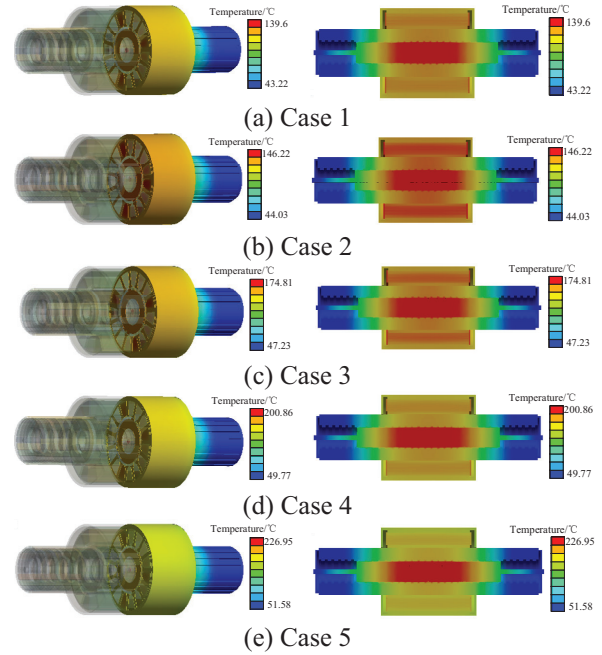


Fig. 10. 3-D temperature distribution and cross-section of MPMLRG @ five operating cases.

experience higher temperature rise, while the permanent magnets remain at relatively low temperatures. Although the LGU winding generates higher losses, its temperature is comparable to that of the RGU owing to the annular winding configuration, which enhances heat conduction to the stator core. However, the enclosure of the inner stator by the mover restricts heat dissipation, leading to a slightly higher temperature rise.

With increasing operating speed, the temperatures of both the RGU and LGU increase monotonically, indicating that natural convection alone is insufficient to offset the additional heat generation. At Case 5, severe thermal stress is observed, particularly in the stator windings, with the LGU inner winding becoming the most thermally critical component. In contrast, the mover and permanent magnets remain within a relatively safe temperature range.

It should be noted that these results are obtained under natural convection cooling only. At high operating speeds, excessive winding temperatures may compromise insulation reliability and reduce machine lifetime. Therefore, enhanced thermal management measures, such as winding impregnation or improved cooling strategies, are required for practical applications.

B. Steady-state LPTN

As shown in Fig. 11, the computing results of steady-state LPTN are compared with the results of FEA.

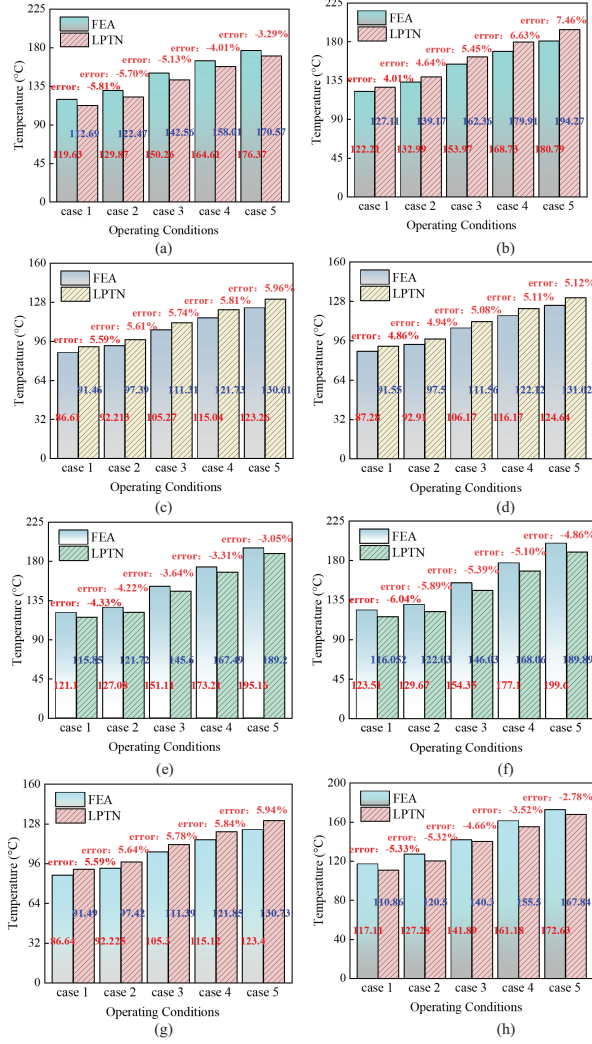


Fig. 11. Comparison of steady-state temperature of major components in the MPMLRG predicted by the LPTN and 3-D FEA: (a) RGU stator, (b) RGU winding, (c) RGU PM, (d) LGU PM, (e) LGU stator, (f) LGU winding, (g) mover, and (h) housing.

It can be observed that the results of the LPTN calculation are in close agreement with those of the FEA, confirming the accuracy of the proposed LPTN model for the MPMLRG. The largest discrepancy is observed in the winding of the RGU, which is mainly attributed to the geometric simplifications adopted in the LPTN model. Nevertheless, the overall agreement demonstrates that the proposed model can provide reliable temperature prediction for engineering applications.

VI. TRANSIENT LPTN

As shown in Fig. 12, the computing results of transient LPTN are compared with the results of FEA.

It can be observed that the final steady-state results of both methods are in good agreement; however, the

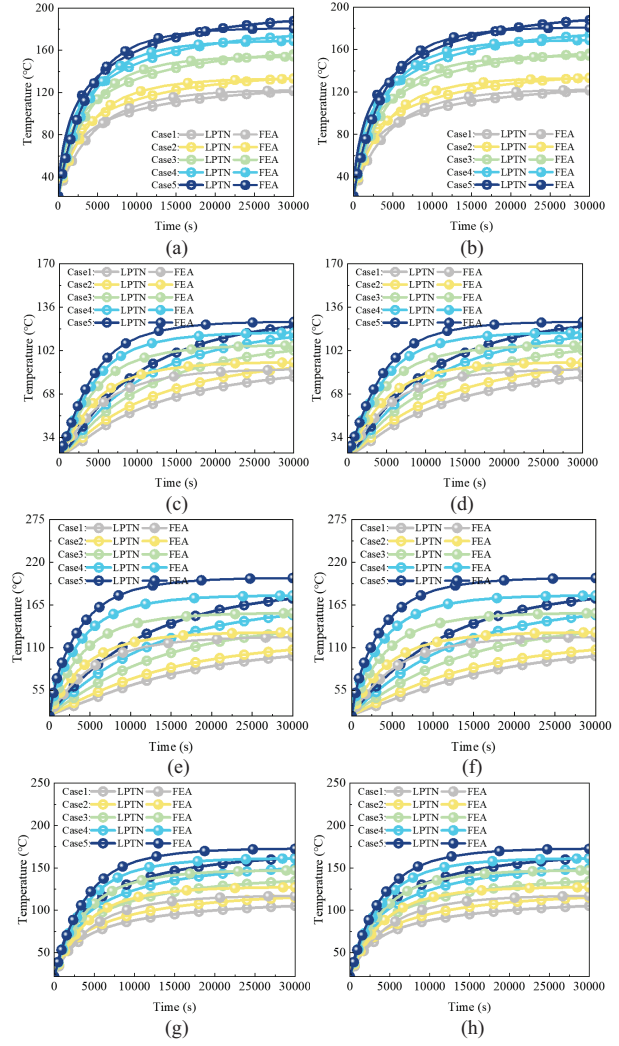


Fig. 12. Comparison of transient temperature of major components in the MPMLRG predicted by the LPTN and 3-D FEA: (a) RGU stator, (b) RGU winding, (c) RGU PM, (d) LGU PM, (e) LGU stator, (f) LGU winding, (g) mover, and (h) housing.

temperature predicted by the LPTN reaches steady-state more slowly than that of the FEA. This discrepancy mainly arises from the intrinsic modeling assumptions of the LPTN. In the LPTN, multiple local heat-transfer paths are represented by equivalent thermal resistances and thermal capacitances. The spatial averaging smooths local temperature gradients and increases the effective thermal inertia of each thermal node, thereby slowing the predicted heat diffusion process. In contrast, the FEA explicitly resolves the distributed temperature field and local heattransfer paths, allowing local regions to respond more rapidly to thermal excitation. Consequently, the LPTN exhibits a slightly slower transient response while maintaining good agreement with the FEA under steady-state conditions. This characteristic

does not affect its suitability for thermal design and steady-state temperature prediction.

VII. CONCLUSION

A lumped-parameter thermal network model for the dual-stator MPMLRG was developed and validated against 3-D FEA. Both methods reveal that stator windings are the most thermally critical components, with the inner stator winding of the LGU exhibiting the highest temperature due to limited heat dissipation. The LPTN demonstrates good agreement with FEA in steady-state prediction, though it slightly underestimates transient dynamics owing to simplified thermal inertia representation. At high operating speeds, winding overheating becomes significant, underscoring the necessity of enhanced cooling strategies. Future work will integrate advanced cooling methods and couple the LPTN with dynamic electromagnetic loss models to support reliability assessment and design optimization. Future work will include experimental validation on a prototype to further verify the proposed thermal model under practical operating conditions.

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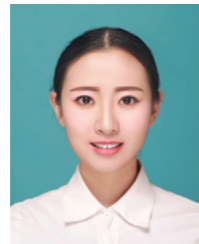
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