

Article

Coupled Design of In-Plane Heat Spreading and Boundary-Tailored Nonuniform Discrete-Heater Geometry for Composite Electrothermal Ice-Protection Systems

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Abstract

Discrete heating and spatially varying convection jointly determine temperature distributions in composite electrothermal ice-protection structures. A two-dimensional transient multilayer model is used to examine the coupling between in-plane heat spreading and discrete heater geometry at fixed total-heat input. Matched-geometry material comparisons, analyses of coverage, strip count, and film thickness, and comparisons of nonuniform layouts under two flight-derived convective boundaries are used to evaluate interface temperature range, an edge thermal-mismatch metric, and thermal response. A 0.1-mm equivalent graphite-sheet spreader reduces the temperature difference between strips and gaps, with the effect depending on heater geometry and convective distribution. Increasing strip count produces different responses in temperature uniformity and the cumulative edge metric. In the 50-mm domain under the two flight boundaries, seven- and ten-strip spreader layouts reduce interface temperature range by 1.352–2.730 K and 2.607–2.930 K, respectively, relative to uniform layouts with the same strip count. For the ten-strip layout without the spreader, the comparison changes from –0.230 K to 1.327 K between the lower- and higher-Reynolds-number boundaries. For the lower-Reynolds-number ten-strip spreader case, domain extension from 50 to 100 mm reduces the benefit from 2.607 to 1.089 K, demonstrating strong dependence on lateral thermal connections.

Keywords: aircraft electrothermal ice protection; discrete heaters; active heating coverage; passive in-plane heat spreading; temperature uniformity; heater-layout design

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1. Introduction

Aircraft icing alters local pressure distributions and aerodynamic loads, thereby increasing drag, reducing lift, and compromising flight safety [1,2]. Electrothermal ice-protection systems transfer heat to the protected surface through heaters installed within or beneath the skin. Embedding heaters in a composite structure places the heat source close to the protected surface. However, discrete strip heaters introduce spatial nonuniformity: local hot spots form above the strips, while the unheated gaps may not reach the required thermal state [3,4]. Layout performance therefore depends on the spatial temperature distribution.

Models of aircraft electrothermal heat transfer have evolved from analyses with prescribed surface heat fluxes to transient descriptions of laminated skins, embedded heat sources, and temperature-dependent phase-change regions. Two- and three-dimensional conduction models can resolve the coupling between through-thickness thermal resistance and in-plane heat redistribution in composites [5,6]. Enthalpy and apparent-heat-capacity methods provide practical treatments of latent heat [7].

Existing studies of electrothermal ice protection address heater performance, power-distribution optimization, and heater arrangement. Uranai et al. numerically analyzed electric-heater anti-icing performance on a NACA 0012 airfoil [8]. Hann et al. experimentally measured heat loads during electrothermal anti-icing and de-icing on UAVs [9]. Pourbagian and Habashi used a surrogate model to optimize an electrothermal wing anti-icing system [10], while Guo et al. optimized power distribution using a differential-evolution algorithm [11]. Bennani et al. numerically simulated anti-icing and de-icing modes of an electrothermal ice-protection system [12], whereas Yousuf et al. studied a gridded heater arrangement [13]. For designs that consider temperature uniformity, thermal response, and active heater area together, multi-objective methods can describe trade-offs among the objectives; NSGA-II, proposed by Deb et al., is one such method [14]. The combined effect of passive in-plane heat spreading and discrete heating in composite structures, particularly the selection of strip count and strip-gap widths at fixed macroscopic heat input and active coverage, still requires analysis under consistent evaluation conditions.

Research on in-plane heat-spreading materials includes thermal management with large-area graphene paper [15], measurements of the c-axis thermal conductivity of nanometer-thick graphite films [16], anisotropic thermal conductivity in reduced-graphene-oxide films [17], scalable production of thick graphene films [18], and the study by Sun et al. [19] on highly conductive graphite films produced by graphitizing chemically imidized polyimide films. Other work has examined electrothermal de-icing parameters of graphene-nanoplatelet films [20], while a review discusses integrated anti-icing and de-icing functions and structures for aircraft [21]. These studies provide material- and device-level context, and motivate an assessment of the temperature response when heat spreaders and discrete heaters are combined in a composite. This study therefore compares the coupled heat-spreader and heater geometry within a common multilayer heat-transfer framework and examines that coupling under spatially varying convective boundaries.

Recent UAV icing studies address the effects of icing at the vehicle, propeller, and forward-flight levels. Ercan and Dalkın conducted full-vehicle propeller-icing experiments on a hovering quadcopter [22]. Müller, Villeneuve, and Hann experimentally characterized performance changes caused by ice accretion on a propeller [23]. Samad, Dhulipalla, Bowers, and Hu studied ice-accretion characteristics on a multirotor propeller in forward flight [24].

Near the stagnation region of an aircraft leading edge, the local heat-transfer coefficient varies markedly along the surface. Spatially varying convection represents this non-uniform cooling [25,26]. Here, the heat-transfer coefficient (HTC) denotes the parameter in the Robin convective-boundary condition applied at the ice/air outer surface. To examine the response of heater layouts to nonuniform cooling, the spatial HTC distributions measured in flight tests are converted into prescribed convective boundary conditions at the ice outer surface and used to design nonuniform strip layouts.

Research on electrical heaters for polymer structures provides material and system context for electrothermal ice protection [27]. The redistribution of strip and gap widths under fixed macroscopic heat input and active coverage remains insufficiently examined. Assigning strip widths algebraically from the local HTC cannot capture lateral conduction coupling within the composite, phase-change layer, and heat spreader. A passive heat

spreader may change the propagation length scale of temperature disturbances and reduce the need for dense strip subdivision. A common multilayer model is therefore used to analyze material configuration, strip count, and nonuniform strip-gap geometry [28,29].

This study examines how in-plane heat spreading and discrete heater geometry jointly affect interfacial temperature distributions and an edge thermal-mismatch metric at prescribed macroscopic heat input. Matched-geometry material comparisons first examine heat spreading. Design-domain statistics and controlled geometry branches then assess the trade-offs associated with coverage, strip count, and film thickness. Finally, selected seven- and ten-strip nonuniform layouts are paired with same-count uniform layouts across material configurations and two flight-derived convective boundaries.

2. Physical Model and Numerical Methods

2.1. Physical Model and Heater Layout

The protected region is represented by a two-dimensional multilayer cross-section with a computational width of $L = 50$ mm. From the outer surface inward, the model contains a fixed-thickness representative ice layer, an upper GFRP layer, a functional replacement layer, a middle GFRP layer, discrete heater regions, and a lower GFRP layer. The 2-mm phase-change layer denotes a representative pre-existing ice thickness at the start of the calculation. It is treated within a fixed reference domain using an apparent-heat-capacity formulation; meltwater drainage, transport, and subsequent refreezing are excluded, as are ice shedding and thickness evolution. The interface-temperature threshold is used as the thermal criterion in the present calculation.

The initial and ambient temperatures are $T_0 = T_\infty = 253.15$ K. The phase-change temperature is $T_m = 273.15$ K, and the apparent phase-change interval is $\Delta T_{pc} = 1$ K, corresponding to 273.15–274.15 K. The latent heat is $L_{latent} = 333500$ J/kg, and the reference density is 917 kg/m³. The solid- and liquid-phase heat capacities are $c_s = 2090$ J/(kg·K) and $c_l = 4218$ J/(kg·K), respectively, and the conductivity assigned to the solid and transition regions is 2.22 W/(m·K). The apparent heat capacity in the transition interval is calculated as $c_s + \frac{L_{latent}}{\Delta T_{pc}}$. The reference density remains constant across phases for mass and

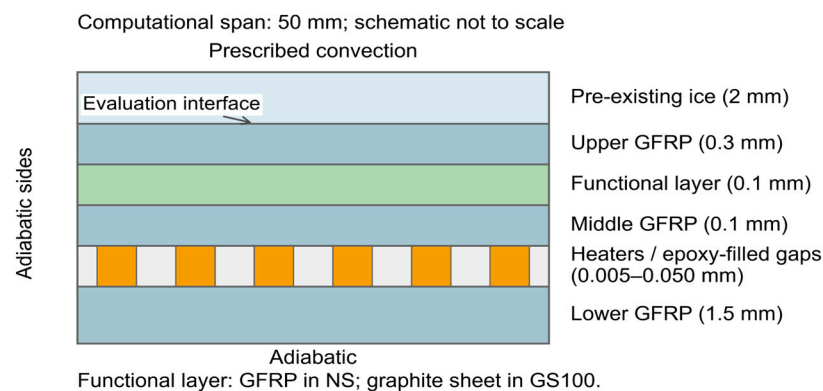
energy accounting in the fixed reference domain. In the fully liquid region, thermal conductivity follows the implemented temperature-dependent water correlation over 273.15–383.15 K.

Table 1 summarizes the thickness, density, thermal conductivities, and specific heats assigned to all layers. Inactive gaps in the heater-film layer are filled with ordinary epoxy resin and receive no heat source; its parameters follow [30]. The functional replacement layer in the no-spreader configuration NS uses GFRP properties, and GFRP is treated as an equivalent isotropic material in the present model. The spreader is represented by equivalent orthotropic properties. GS100 is the name used here for the 0.100-mm equivalent graphite-sheet configuration. Its parameters follow the 100- μ m PGS information from Panasonic Corporation (Kadoma, Osaka, Japan) [31]. For GS100, the in-plane conductivity of 700 W/(m·K) and through-thickness conductivity of 26 W/(m·K) are taken from the 100- μ m column of the material-characteristics table. The manufacturer's X–Y and Z directions correspond to the present in-plane and through-thickness directions, respectively. The numerical values for the heater and all material layers are listed in Table 1.

Table 1. Material properties used in the model.

Component	Thickness (mm)	ρ (kg·m ⁻³)	k_x (W/(m·K))	k_y (W/(m·K))	c_p (J/(kg·K))
Phase-change layer	2.0	917.0	2.22	2.22	2090
Upper/mid/lower GFRP layers	0.3/0.1/1.5	1850.0	0.30	0.30	1200
Functional replacement layer in NS	0.1	1850.0	0.30	0.30	1200
Graphite-sheet equivalent layer in GS100 [31]	0.1	850.0	700.0	26.00	850
Equivalent heater film	0.005–0.050	2486.7	150.00	150.00	2000
Inactive-region epoxy [30]	Same as heater-film thickness	1200	0.20	0.20	1150

The baseline model sets the thermal contact resistance at adjacent interfaces to zero. Using the thermal-impedance value of 3M 8810 tape (3M Company, St. Paul, MN, USA), GS100-R applies one area-normalized resistance on each side of the spreader; each resistance is $5.8 \times 10^{-4} \text{ m}^2\cdot\text{K/W}$ [32]. Two-dimensional transient conduction is solved in each layer using the following equation. Figure 1 and Table 1 specify the geometry and material properties.

**Figure 1.** Fixed-span multilayer computational domain showing the material layers, heater strips, gaps, and principal design dimensions.

$$\rho c_{\text{eff}}(T) \frac{\partial T}{\partial t} = \frac{\partial}{\partial x} \left(k_x \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(k_y \frac{\partial T}{\partial y} \right) + \dot{q} \quad (1)$$

Here, ρ is the material density, $c_{\text{eff}}(T)$ is the temperature-dependent effective specific heat capacity, k_x and k_y are the in-plane and through-thickness thermal conductivities, respectively. The term $\dot{q}$ denotes the volumetric heat source. The volumetric heat source is nonzero only within the active heater regions. The latent heat of phase change is treated using the apparent-heat-capacity method. The effective specific heat capacity is defined as:

$$c_{\text{eff}}(T) = \begin{cases} c_i, & T < T_m \\ c_i + \frac{L_{\text{latent}}}{\Delta T_{\text{pc}}}, & T_m \leq T < T_m + \Delta T_{\text{pc}} \\ c_l, & T \geq T_m + \Delta T_{\text{pc}} \end{cases} \quad (2)$$

The solid- and liquid-phase heat capacities, latent heat, phase-change temperature and apparent transition interval in Equation (2) use the prescribed phase-change parameters.

The initial temperature condition is $T(x, y, 0) = T_0$. A Robin convective boundary condition is applied at the ice/air outer surface $-k_y \frac{\partial T}{\partial n} = h_{\text{top}}(T - T_{\infty})$. The lower and lateral boundaries of the computational domain are adiabatic. For uniform periodic layouts, the lateral boundaries also represent mirror-symmetry planes between adjacent periodic cells. In the zero-contact-resistance baseline, temperature and normal heat flux are continuous across interfaces. With finite contact resistance, normal heat flux is continuous and the temperature jump equals the product of contact resistance and normal heat flux. The effects of the external flow are represented by the prescribed heat-transfer coefficient and ambient temperature.

For each prescribed macroscopic heat-input condition, the total heat input over the computational span is held constant. For an active heating coverage ratio of η_h , the local surface heat flux and volumetric heat source within the active regions are defined as follows:

$$Q_{\text{macro}} \in \{10,000, 15,000, 20,000\} \text{ W/m}^2, \quad q''_{\text{local}} = \frac{Q_{\text{macro}}}{\eta_h}, \quad \dot{q} = \frac{Q_{\text{macro}}}{\eta_h d_{\text{film}}} \quad (3)$$

The active heating coverage ratio η_h is the geometric fraction of the computational span that is heated directly. Changing η_h changes the concentration and spatial distribution of heat generation within the active regions.

The layout variables are the material configuration c , active heating coverage ratio η_h , number of equal-width heater strips N_w , and heater-film thickness d_{film} . For an equal-width periodic layout, the width of each heater strip w and the width of each complete gap g are:

$$w = \frac{\eta_h L}{N_w}, \quad g = \frac{(1 - \eta_h)L}{N_w} \quad (4)$$

A half-gap is placed at each end of the computational domain, with $N_w - 1$ complete gaps between the heater strips. Therefore, both w and g are determined by η_h and N_w . A discrete, non-full-factorial design space is used. Intermediate levels are added in selected ranges to increase the resolution of the local design space.

The baseline configurations without and with the spreader have identical layer dimensions, heater geometry, total heat input, external boundaries, and numerical settings. Only the properties assigned to the functional replacement layer differ. This comparison evaluates the effect of the prescribed material configuration on the temperature field.

2.2. Performance Metrics and Constraints

Temperature extrema, temperature range, and the edge metric are evaluated at the final simulation time; response time is defined separately from the transient history, and

the evaluation interface is the ice/composite interface. The interfacial temperature is interpolated from adjacent cell-center temperatures using the corresponding half-cell thermal resistances. The first metric is the interfacial peak-to-valley temperature difference:

$$T_{\text{peak}} = \max_x T_{\text{eval}}(x), \quad T_{\text{valley}} = \min_x T_{\text{eval}}(x), \quad F_1 = T_{\text{peak}} - T_{\text{valley}} \quad (5)$$

Here, T_{peak} and T_{valley} are the maximum and minimum temperatures at the evaluation interface, respectively. F_1 is the interfacial peak-to-valley temperature difference. A lower F_1 indicates a more uniform temperature distribution at the ice/composite interface.

The second objective is the cumulative edge thermal-mismatch comparison metric. The $F_{2,\text{edge,total}}$ is calculated from the heater-film/substrate interfacial temperature $T_{\text{hs}}(x)$ at the final simulation time. It represents the cumulative local thermal-mismatch response at the heater-strip edges. Its formulation is based on the analytical concepts developed by Suhir for thermal mismatch in slender bonded bimaterial assemblies with compliant adhesive layers [33,34]:

$$F_{2,\text{edge,total}} = \sum_{e=1}^{2N_w} \int_{\Omega_e} \tau_{\text{sl}}^2(s) ds \quad (6)$$

Here, τ_{sl} is the interfacial slip-response comparison quantity in the reduced-order model. Ω_e is the integration window associated with the e -th strip edge. Each heater strip has two edges. Equation (6) therefore accumulates the contributions from $2N_w$ edge windows.

The free thermal-strain mismatch between the heater film and substrate is defined as:

$$\Delta\epsilon(x) = (\alpha_h - \alpha_s) [T_{\text{hs}}(x) - T_{\text{ref}}] \quad (7)$$

The reduced-order parameters are defined as:

$$\bar{E}_j = \frac{E_j}{1-\nu_j^2}, \quad A_j = \bar{E}_j h_j, \quad K_b = \frac{G_b}{t_b}, \quad \lambda = \left[K_b \left(\frac{1}{A_h} + \frac{1}{A_s} \right) \right]^{1/2} \quad (8)$$

The reduced-order interfacial slip field $\phi(x)$ then satisfies:

$$\frac{d^2\phi}{dx^2} - \lambda^2\phi = \frac{d\Delta\epsilon}{dx}, \quad \phi(0) = \phi(L) = 0, \quad \tau_{\text{sl}}(x) = K_b\phi(x) \quad (9)$$

The heater-film/substrate interfacial temperature is first linearly resampled onto a uniform grid containing 16,001 nodes. Equation (9) is then solved using a second-order central-difference scheme. An integration window is centered at each heater-strip edge. Its total length is defined as:

$$\ell_{\text{edge}} = \min \left(\frac{2}{\lambda}, \frac{g}{2}, 1.0 \text{ mm} \right) \quad (10)$$

Values at the endpoints of each integration window are obtained by linear interpolation. Overlapping windows are integrated separately and retained in Equation (6). This treatment preserves the contribution of the number of heater-strip edges to the cumulative metric.

The equivalent heater-film properties are $E_h = 5.30 \times 10^9$ Pa, $\nu_h = 0.34$, and $\alpha_h = 1.60 \times 10^{-5}$ K⁻¹. These values are consistent with the Kapton EN technical data from DuPont (Wilmington, DE, USA) [35]. The equivalent properties of the G10/GFRP substrate

are $E_s = 2.90 \times 10^{10}$ Pa, $\nu_s = 0.14$, and $\alpha_s = 1.20 \times 10^{-5}$ K⁻¹. The adhesive-layer shear modulus is set to $G_b = 4.14 \times 10^8$ Pa based on the AF163 technical data from 3M Company (St. Paul, MN, USA) [36]. The comparison thickness is set to $t_b = 1.00 \times 10^{-4}$ m and is treated as a common assumed parameter in the reduced-order model. All layouts use the same equivalent material properties and numerical procedure.

Under this common parameter set, a lower $F_{2,\text{edge},\text{total}}$ indicates a lower cumulative local thermal-mismatch response at the heater-strip edges. This metric combines the thermal field with the reduced-order slip model and is used only to compare the relative edge responses of different heater layouts. The Pareto analysis uses the cumulative metric. Selection with the mean per-edge metric has not been repeated, so the reported representatives apply to the stated cumulative objective. An auxiliary mean response per edge is calculated in the strip-count comparisons:

$$\bar{F}_{2,\text{edge}} = F_{2,\text{edge},\text{total}} / (2N_w) \quad (11)$$

The third objective is the active heating coverage ratio $F_3 = \eta_h = \frac{N_w w}{L}$. The three objectives are to minimize the interfacial peak-to-valley temperature difference F_1 , the cumulative edge thermal-mismatch comparison metric $F_{2,\text{edge},\text{total}}$, and the active heating coverage ratio η_h . Only layouts that satisfy all geometric constraints, final-time temperature requirements, and transient-response requirements in Table 2 are included in the subsequent Pareto screening. The response time t_{resp} is the first time at which the minimum temperature over the entire evaluation interface reaches the phase-change temperature T_m . A feasible layout must satisfy $t_{\text{resp}} \leq 60$ s. Within the design space considered, every layout that satisfies the other final-time thermal requirements also satisfies the response-time constraint.

Table 2. Design variables, performance objectives, and feasibility criteria.

Symbol	Definition	Level or Criterion	Role
Q_{macro}	Macroscopic heat input	10, 15, 20 kW/m ²	Independent operating condition
η_h	Active heating coverage ratio	0.15–0.90	Sampled design variable and minimization objective
N_w	Number of equal-width heater strips	1–6	Integer design variable
d_{film}	Heater-film thickness	0.005, 0.010, 0.015, 0.020, 0.050 mm	Discrete design variable
F_1	Interfacial peak-to-valley temperature difference	Unit: K	Minimized
$F_{2,\text{edge},\text{total}}$	Sum of the edge-window integrals of τ_{sl}^2	Unit: Pa ² ·m	Minimized
feasible	Final-time temperature and response-time constraints	$T_{\text{peak}} \leq 373.15$ K $T_{\text{valley}} \geq 273.65$ K $t_{\text{resp}} \leq 60$ s	Layout screening

2.3. Numerical Methods and Verification

All configurations use a cell-centered finite-volume discretization on an orthogonal grid aligned with the material interfaces. Interfacial effective conductivities are calculated

from adjacent-cell thermal resistances. Time integration uses an alternating-direction implicit method, with tridiagonal systems solved by the Thomas algorithm. Temperature-dependent properties are updated with the temperature field. The heat-transfer calculations use an in-house C++17 solver compiled with MinGW-w64 GCC 13.1.0. Layout generation and post-processing use Python 3.12.

The main analysis uses a mesh-scale factor of $s_{\text{mesh}} = 2$. The base nominal in-plane spacing is 0.10 mm, giving 0.050 mm at the main mesh-scale factor of 2, and each geometric segment contains at least two control volumes. The base through-thickness counts are 30, 15, 5, 5, 2, and 20 cells; the main mesh uses 60, 30, 10, 10, 4, and 40 cells, respectively. The main analysis uses a time step of $\Delta t = 0.00125$ s and a final simulation time of $t_{\text{final}} = 60$ s.

Numerical verification includes comparison with an analytical solution, an energy-conservation assessment, and mesh- and time-step-sensitivity analyses. The conduction benchmark compares a 0.10-m numerical domain with the semi-infinite analytical solution over 0–50 mm, using the same step-surface temperature and an initial-temperature far boundary. At $t = 100$ s, this interval is not appreciably affected by the finite far boundary of the numerical domain. The L_2 norm characterizes the overall spatial error over the comparison interval, whereas the L_∞ norm characterizes the maximum local error. The energy-conservation assessment compares the cumulative heat input with the sum of the sensible-energy increase, latent-energy increase, and convective heat loss through the boundary. The mesh- and time-step-sensitivity analyses compare changes in the peak-to-valley temperature difference at the ice/composite interface across different discretization levels. These tests verify the numerical implementation and discretization. The complete multilayer ice-protection system has not been validated against experimental temperature measurements; the results therefore describe model-based comparisons under the prescribed material properties and boundary conditions.

2.4. Configuration Comparisons and Multi-Objective Analysis

Three types of analysis are conducted to distinguish the effect of material configuration from that of heater-geometry reselection. These analyses comprise matched-geometry comparisons, performance-preserving coverage matching, and three-objective nondominance analysis. At each macroscopic heat input of 10, 15, and 20 kW/m², separate scenarios are defined for configurations without and with the prescribed heat-spreading layer. Thermal-feasibility screening and three-objective nondominance evaluation are performed independently for each scenario within the same sampled design domain. The matched-geometry comparison considers only pairs with identical Q_{macro} , η_h , N_w , and d_{film} but different material configurations. It isolates the effect of the functional replacement-layer material. Performance-preserving coverage matching allows the strip count and heater-film thickness to be reselected. Both F_1 and $F_{2,\text{edge,total}}$ of the configuration with the heat-spreading layer must not exceed those of the corresponding configuration without the layer. This comparison evaluates changes in active heating coverage while preserving both thermal-performance metrics.

A feasible configuration belongs to the sampled Pareto set if no other feasible configuration in the same comparison set is no worse in all three objectives and strictly better in at least one objective. A representative configuration is selected for each material configuration using min–max normalization within its Pareto set and the distance from the ideal point.

$$\tilde{F}_{j,i} = \frac{F_{j,i} - \min_{k \in \mathcal{P}_c} F_{j,k}}{\max_{k \in \mathcal{P}_c} F_{j,k} - \min_{k \in \mathcal{P}_c} F_{j,k}} i^* = \arg \min_{i \in \mathcal{P}_c} \left(\tilde{F}_{1,i}^2 + \tilde{F}_{2,i}^2 + \tilde{\eta}_{h,i}^2 \right)^{1/2} \quad (12)$$

Here, $\mathcal{P}_c$ denotes the sampled Pareto set for material configuration c . The quantities $\tilde{F}_{1,i}$, $\tilde{F}_{2,i}$, and $\tilde{\eta}_{h,i}$ are the normalized values of the three objectives, respectively. The configuration selected using Equation (12) represents a characteristic compromise among the three objectives within the sampled design domain.

The active-coverage substitution analysis is performed separately after Pareto screening for each macroscopic heat-input scenario. For each Pareto configuration s without the heat-spreading layer, the set of configurations with the layer that satisfy the performance-preserving conditions is defined as:

$$\mathcal{A}(s) = \left\{ y \in \mathcal{D}_{\text{sp,feas}} : F_1(y) \leq F_1(s), F_{2,\text{edge,total}}(y) \leq F_{2,\text{edge,total}}(s) \right\} \quad (13)$$

The matched configuration is the member of $\mathcal{A}(s)$ with the lowest active heating coverage ratio, η_h . If several configurations have the same coverage ratio, they are ranked sequentially by the normalized distance from the configuration without the heat-spreading layer in the two thermal metrics, F_1 , $F_{2,\text{edge,total}}$, and the configuration identifier. This comparison evaluates the change in active heating coverage required by the configuration with the heat-spreading layer while maintaining interfacial temperature uniformity and cumulative edge thermal-mismatch performance. Figure 2 summarizes the workflow for the baseline active–passive coupled design and the heater-layout design under flight-derived convective boundaries.

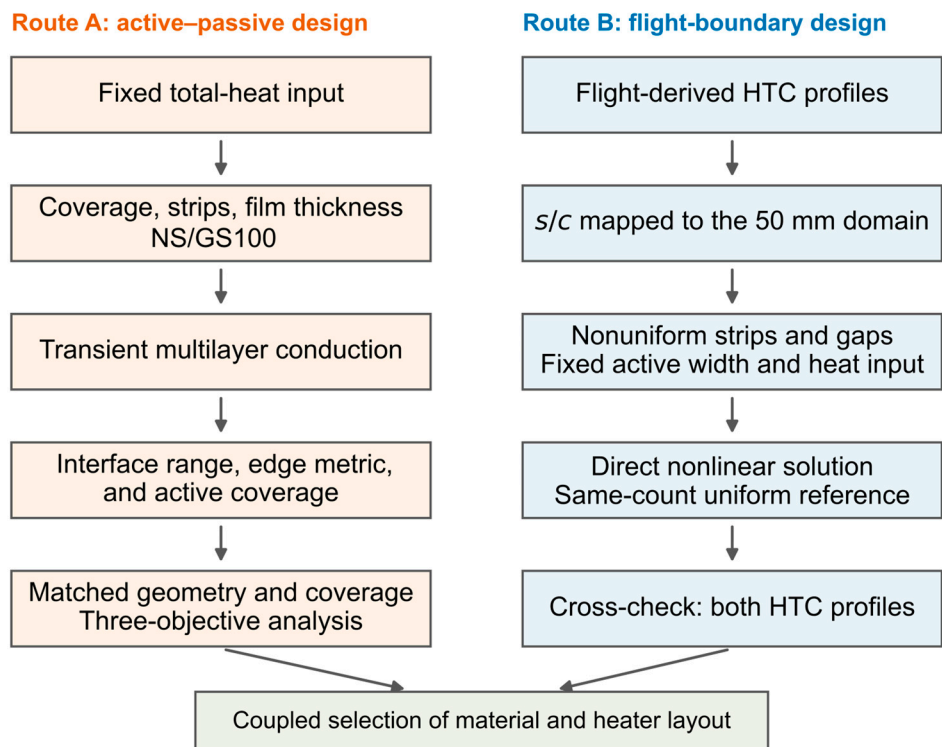


Figure 2. Workflow for heater-layout design and evaluation.

2.5. Heater-Layout Design Under Flight-Derived Convective Boundaries

Two zero-angle-of-attack convective heat-transfer distributions are represented by digitized nodes from NASA TM-102448/AIAA-90-0199 [25]. The retained normalized surface-coordinate interval is -0.024 to 0.083 and is mapped to the 50-mm computational domain using Equation (14).

$$\frac{s}{c} = -0.024 + 0.107 \frac{x}{L}, \quad x = \frac{L \left(\frac{s}{c} + 0.024 \right)}{0.107} \quad (14)$$

The mapped stagnation location is 11.215 mm from the left boundary. Piecewise-linear interpolation between the nodes provides the spatial coefficient in the ice-surface convective condition in Equation (15).

$$-k_y \frac{\partial T}{\partial n} = h(x) [T(x, y_{\text{outer}}, t) - T_{\infty}] \quad (15)$$

The uniform design-domain scan uses $N_w = 1-6$, whereas nonuniform layout generation uses $N_w = 4-10$. The latter fixes the domain width at 50 mm, total strip, and gap widths at 25 mm each, coverage at $\eta_h = 0.50$, film thickness at 0.020 mm, and heat input at 15 kW/m². A layout contains N_w strips and $N_w + 1$ gaps, including both end gaps. Each width is at least 1 mm, and adjacent widths of the same type have a ratio no greater than 3. The widths satisfy:

$$\sum_{i=1}^{N_w} w_i = 25 \text{ mm}, \quad \sum_{j=0}^{N_w} g_j = 25 \text{ mm} \quad (16)$$

Strip and gap widths are parameterized separately. For each set, an individual width is $l_i = 1 + (25 - m) \text{softmax}(z)_i$ in millimeters. Here, m is the number of widths in the set. Layouts are checked for total width, coordinate continuity, minimum width, and adjacent-width ratios. The original generation procedure uses continuous-density proposals, projection to discrete strips and gaps, surrogate-assisted proposals, and direct transient thermal evaluation. The density is represented by a cubic B-spline with five coefficients and two updates from each of three starts. Projection uses eight starts and L-BFGS-B, with at most 250 iterations and $\text{ftol} = 10^{-11}$; the density and projection grids contain 501 and 101 points, respectively.

The search starts with 20 records for each material-boundary combination: 28 uniform records are reused and 52 layouts are evaluated directly across the four combinations. Gaussian-process models of interface range, surface range, and cumulative F_2 propose candidates. Each round directly evaluates five candidates per active combination, selected from at most 700 admissible proposals within 7000 generation attempts. Up to three rounds are allowed. A combination stops after the second round when improvement in its best interface range is below 0.034610 K in both preceding rounds.

The original screening examined four source candidates under both boundaries. Each candidate was required to satisfy the thermal and response criteria in Table 2, improve the interface range by more than 0.069220 K relative to both its same-count uniform reference and the best uniform reference over $N_w = 4-10$, maintain a non-worsening surface range, and give a finite F_2 . Candidates are ranked by worst-boundary interface range, scenario-normalized interface range, surface range, the change in F_2 relative to $N_w = 6$, strip count, and geometric smoothness. The seven- and ten-strip geometries selected by this original procedure are retained for the revised material comparisons. Each retained

geometry is reevaluated using the prescribed layer properties and apparent-heat-capacity formulation against its same-count uniform reference under both prescribed boundaries.

The retained source-coordinate interval corresponds to 57.031 mm for a chord of 0.533 m and is mapped to the 50-mm domain. A separate source-scale comparison multiplies node distances from stagnation by 1.14062, retains the node HTC values and piecewise-linear interpolation, and samples the resulting distribution within the domain. This comparison changes the spatial scale, sampled interval and mean cooling intensity together.

The exact strip and gap widths of the retained seven- and ten-strip nonuniform layouts are listed in Table A1. The strips and gaps are ordered from left to right, and the gap widths include both end gaps.

3. Results and Discussion

3.1. Numerical Accuracy and Time-Step-Sensitivity Analyses

To assess the accuracy of the numerical method and the effect of discretization on configuration comparisons, a one-dimensional analytical-conduction comparison, an energy-accounting check, and spatial- and temporal-discretization checks were performed. Figure 3a compares the profiles at $t = 100$ s. The analytical and numerical solutions are compared over the 0–50-mm overlap interval. The L_2 and L_∞ errors are 0.015 K and 0.041 K, respectively, indicating that the numerical solution accurately resolves the temperature distribution in this interval. The maximum difference over the full domain is approximately 2.028 K and is concentrated near the far boundary. This difference results from the different far-boundary conditions of the finite numerical domain and the semi-infinite solid.

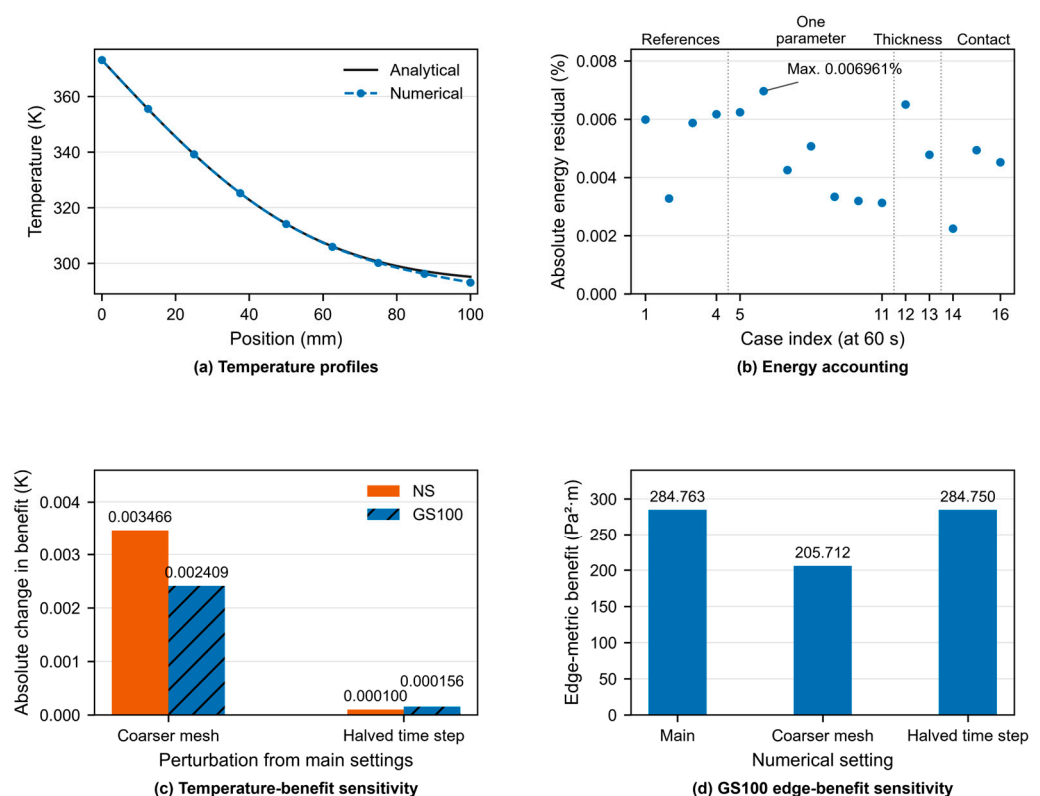


Figure 3. Numerical checks. (a) Temperature-profile comparison; (b) final-time energy residuals; (c) changes in temperature-range benefit; (d) GS100 edge-metric benefit.

The energy-accounting residual is defined as cumulative input heat minus the sum of internal-energy increase and convective boundary loss, normalized by cumulative input heat. The internal-energy increase includes sensible and latent contributions. Figure 3b includes all 16 cases in the material and contact-parameter diagnostic set. Each case uses a six-strip uniform layout, an active heating coverage of 0.5, a heat input of 15 kW/m², a uniform heat-transfer coefficient of 15 W/(m²·K), and a duration of 60 s. In the plotted order, cases 1–4 comprise the no-spreader baseline, the historical equivalent-material reference, and two material parameter combinations; cases 5–11 vary in in-plane conductivity, through-thickness conductivity, density, or specific heat individually; cases 12–13 vary in spreader thickness; and cases 14–16 introduce finite contact resistance. The set covers changes in conduction, heat storage, thickness, and interface contact-testing energy accounting across the prescribed material diagnostics. The largest absolute normalized residual at the final time is 0.006961%, indicating agreement between cumulative input heat and stored and dissipated heat under these conditions.

The discretization sensitivity analysis uses ten-strip uniform and nonuniform layouts under the lower-Reynolds-number boundary. This pair has the smallest absolute layout benefit without a spreader and is therefore used to examine the sensitivity of a small benefit to discretization; the same layouts are also compared with GS100. Layout benefit is defined as uniform-layout F_1 minus nonuniform-layout F_1 . The nominal in-plane spacings of the main and coarser meshes are 0.050 and 0.0667 mm, respectively. The main time step is 0.00125 s, and additional calculations use 0.000625 s. Figure 3c shows the absolute changes in benefit relative to the main settings for the coarser mesh and halved time step. Under the main settings, the benefits are −0.230 K for NS and 2.607 K for GS100. The mesh change alters the benefits by 0.00347 and 0.00241 K, respectively, while halving the time step changes either benefit by no more than 0.000156 K. The NS benefit remains negative and the GS100 benefit positive in all three settings. These changes are small relative to the benefit magnitudes and preserve the comparison direction, supporting the subsequent material-response analysis of the ten-strip layouts under the lower- Re boundary.

The cumulative edge metric shows a stronger response to spatial discretization. Figure 3d presents the edge-metric benefit for the same GS100 lower- Re ten-strip pair under three numerical settings, calculated as the uniform-layout cumulative edge metric minus its nonuniform-layout counterpart. The benefits on the main and coarser meshes are approximately 285 and 206 Pa²·m, respectively, corresponding to a 27.8% decrease relative to the main mesh; halving the time step changes the benefit by approximately 0.013 Pa²·m. All comparisons use 16,001 post-processing nodes. The benefit direction is consistent for this pair, and its magnitude is more sensitive to the mesh than to the time step. The mesh dependence of the benefit magnitude limits the quantitative interpretation of small edge-metric improvements, even where the comparison direction is preserved for the tested pair.

3.2. Lateral Heat-Spreading Mechanism in Matched-Geometry Layouts

Matched-geometry NS and GS100 layouts are compared to examine temperature redistribution and lateral heat transport. The heat input, active heating coverage, strip count, heater-film thickness, boundary conditions and numerical settings are identical; only the material parameters assigned to the functional replacement layer are changed. The comparison uses a macroscopic heat input of 15 kW/m², a uniform HTC of 15 W/(m²·K), a coverage of 0.50, six equal-width strips, and $d_{\text{film}} = 0.020\text{ mm}$, evaluated at 60 s. Figure 4a,b uses the same spatial grid and color scale. As shown in Figure 4c and Table 3a, the maximum and minimum interface temperatures are 314.635 and 277.828 K for NS, giving $F_1 = 36.807\text{ K}$, and 300.233 and 299.646 K for GS100, giving $F_1 = 0.587\text{ K}$. The

spreader lowers the peak by 14.402 K and raises the valley by 21.818 K, reducing the interface temperature range by approximately 98.4%. The cumulative edge metric decreases from 2.161×10^6 to 1.829×10^5 Pa²·m, a reduction of approximately 91.5%. Temperature peaks occur mainly above the heater strips and valleys above the gaps. The layer transfers heat from the heated regions toward adjacent gaps, producing a more uniform interface temperature distribution. Heat passes through the multilayer structure before reaching the ice/composite interface, so the interface temperature extrema may be displaced from the heater-strip edges.

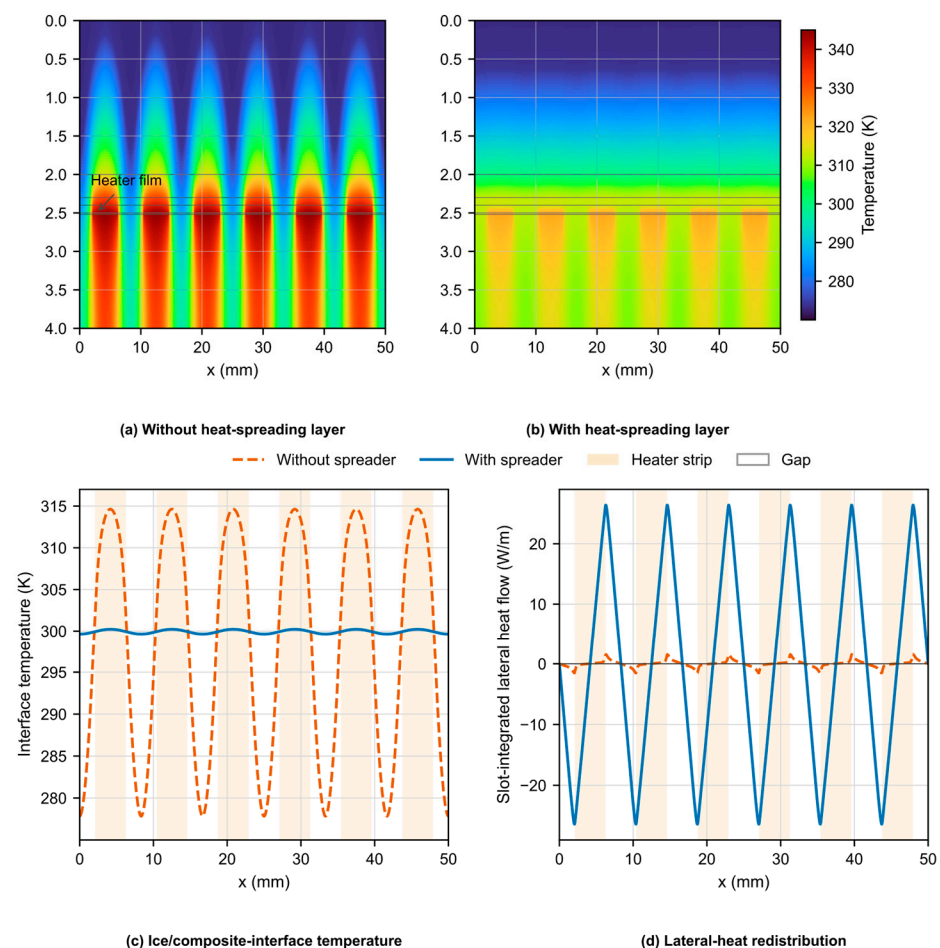


Figure 4. Matched-geometry comparison at a macroscopic heat input of 15 kW/m². (a,b) NS and GS100 temperature fields; (c) interface temperature; (d) integrated lateral heat flow.

Table 3. Matched-geometry comparison and parameter responses.

(a) Matched-geometry comparison.				
Configuration	T_{peak} (K)	T_{valley} (K)	F_1 (K)	$F_{2,\text{edge,total}}$ (Pa ² ·m)
NS	314.635	277.828	36.807	2.161×10^6
GS100	300.233	299.646	0.587	1.829×10^5
GS100 minus NS	−14.402	21.818	−36.220	$−1.979 \times 10^6$
(b) Thickness effects.				
Thickness (mm)	Temperature range (K)	Min temperature (K)	Response time (s)	
0.050	0.670	299.497	11.110	
0.070	0.396	299.510	11.170	
0.100	0.190	299.426	11.289	
(c) Contact-resistance effects.				

Resistance per face (m ² ·K/W)	Temperature range (K)	Min temperature (K)	Response time (s)
0	0.587	299.646	11.001
3.2 × 10 ^{−4}	0.493	298.522	12.067
5.8 × 10 ^{−4}	0.434	297.594	12.886

To further characterize lateral heat transport, the through-thickness-integrated lateral heat flow in the functional replacement layer is defined as:

$$J_x(x) = \int_{y_1}^{y_2} q_x(x, y) dy \quad (17)$$

Here, y_1 and y_2 denote the lower and upper boundaries of the functional replacement layer, respectively.

As shown in Figure 4d, the magnitude of the integrated lateral heat flow increases from approximately 1.532 W/m for NS to 26.400 W/m for GS100; positive and negative values indicate heat transport in the two in-plane directions. The larger lateral flow accompanies lower temperature peaks and higher gap temperatures, indicating enhanced heat exchange between the strips and gaps. This heat-transfer path explains the reduced strip-scale temperature variation in Figure 4c.

To distinguish the effect of in-plane conductivity from those of the other properties, in-plane conductivity is varied for the same six-strip uniform layout while through-thickness conductivity, density, specific heat and spreader thickness are fixed at 1 W/(m·K), 1900 kg/m³, 850 J/(kg·K), and 0.100 mm, respectively. The fixed value of 1 W/(m·K) is used only as an idealized through-thickness conductivity in the controlled parameter scans. Nanometer-thick graphite films can exhibit cross-plane conductivities of this order [16], but this does not establish the value for the 100 µm PGS sheet used in GS100. Increasing in-plane conductivity from 0.3 to 100 W/(m·K) reduces the interface temperature range from 37.810 to 3.436 K. Further increases to 700 and 1000 W/(m·K) yield ranges of 0.335 and 0.190 K, respectively. The smaller absolute reductions at high conductivity indicate diminishing improvement in temperature uniformity.

Thickness changes further affect heat transport and storage in the spreader. Table 3b presents the interface temperature range, minimum temperature, and response time for the thickness cases. For the in-plane-conductivity set of 1000 W/(m·K), increasing spreader thickness from 0.050 to 0.070 and 0.100 mm reduces the interface temperature range from 0.670 to 0.396 and 0.190 K, respectively. The three GFRP thicknesses remain fixed, so the total structural thickness varies with spreader thickness. Increasing thickness raises both in-plane sheet conductance and areal heat capacity, while increasing through-thickness thermal resistance. Table 3b also shows a slight increase in response time with thickness, indicating that thickness selection should consider both temperature uniformity and heating response.

Interface contact resistance affects both temperature uniformity and heating response. Table 3c lists the contact-resistance comparisons for GS100. Increasing the contact resistance on each face from zero to 5.8 × 10^{−4} m²·K/W reduces the interface temperature range from 0.587 to 0.434 K and the minimum temperature from 299.646 to 297.594 K, while increasing the response time from 11.001 to 12.886 s. The smaller temperature range accompanies a lower minimum temperature and delayed heating, indicating that heat spreading should be assessed together with temperature level and transient response.

3.3. Consistency of Heat-Spreading Effects Across the Sampled Design Domain

To examine whether heat redistribution between strips and gaps persists across the sampled design domain, matched-geometry comparisons are performed at three macroscopic heat-input levels. Each pair has identical heat input, active heating coverage, strip count, heater-film thickness, and boundary conditions; only the material parameters of the functional replacement layer are changed. Of 279 candidate pairs, 236 have complete final states at 60 s, comprising 92, 80, and 64 pairs at 10, 15, and 20 kW/m², respectively. The NS calculations in the remaining pairs reach the temperature limit of the liquid-conductivity correlation before 60 s. Figure 5 uses the complete pairs, with each change defined as the GS100 result minus the NS result. Medians describe the typical response, while interquartile intervals describe variation among configurations.

In Figure 5a,b, the median maximum-temperature change is negative and the median minimum-temperature change is positive, with their respective interquartile intervals below and above zero. The typical effect of the spreader is therefore to lower temperature peaks and raise temperature valleys, through lateral heat transport between strips and gaps. In Figure 5c,d, both the medians and interquartile intervals of the changes in interface temperature range and cumulative edge metric lie below zero. The individual paired results also show reductions in both metrics throughout the complete paired set.

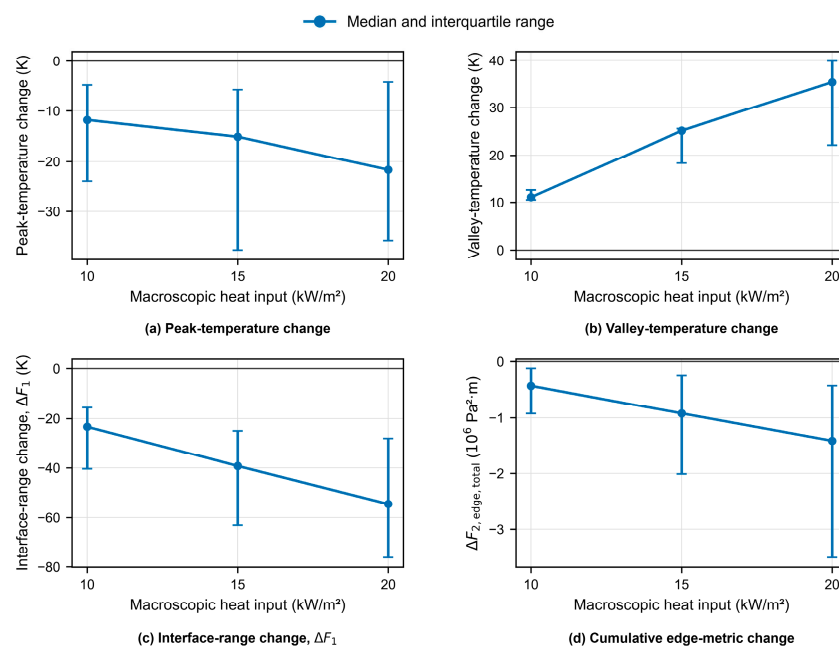


Figure 5. Performance changes induced by the heat-spreading layer for strictly matched-geometry configurations at three macroscopic heat-input levels.

With increasing heat input, the medians in Figure 5c,d become more negative, indicating larger absolute reductions. The median reduction in interface temperature range increases from 23.408 K at 10 kW/m² to 54.425 K at 20 kW/m². Comparing the 64 geometries common to the three groups gives the same median trends for both metrics, supporting the influence of heat input on the magnitude of heat spreading. This is consistent with the lateral heat-transfer mechanism: redistribution of local heat produces greater absolute improvement at higher heat input. The interquartile intervals in Figure 5c,d also widen, indicating greater differences in absolute improvement among configurations. Coverage and strip count alter strip widths, gaps and lateral conduction distances, while film thickness affects local temperature response. Spreader selection must therefore account for heat

input and heater geometry when comparing temperature uniformity and the cumulative edge metric.

3.4. Heat-Input Response and Changes in Representative Configurations

Two comparisons are used to examine how heat input and geometry selection affect temperature distribution. The first holds coverage, strip count, and heater-film thickness constant and varies only heat input. The second selects representative configurations separately for each combination of heat input and material to examine the resulting design trade-offs.

The upper row of Figure 6 compares heat-input responses at fixed geometry. The geometric parameters are set to $\eta_h = 0.50$, $N_w = 6$, and $d_{\text{film}} = 0.020\text{mm}$, and the results at 60 s are evaluated for heat inputs of 10, 15, and 20 kW/m^2 . Figure 6a shows that both maximum and minimum temperatures rise with heat input for the two materials, with a smaller difference between the GS100 extrema. In Figure 6b, the NS interface temperature range increases from 21.184 to 50.411 K, whereas the GS100 range increases from 0.421 to 0.772 K. The temperature levels and changes in F_1 in Figure 6a,b indicate that increasing heat input raises the overall temperature, while lateral heat transport through the spreader reduces strip-scale temperature variation. Figure 6c shows that the cumulative edge metric increases with heat input for both materials and remains lower for GS100.

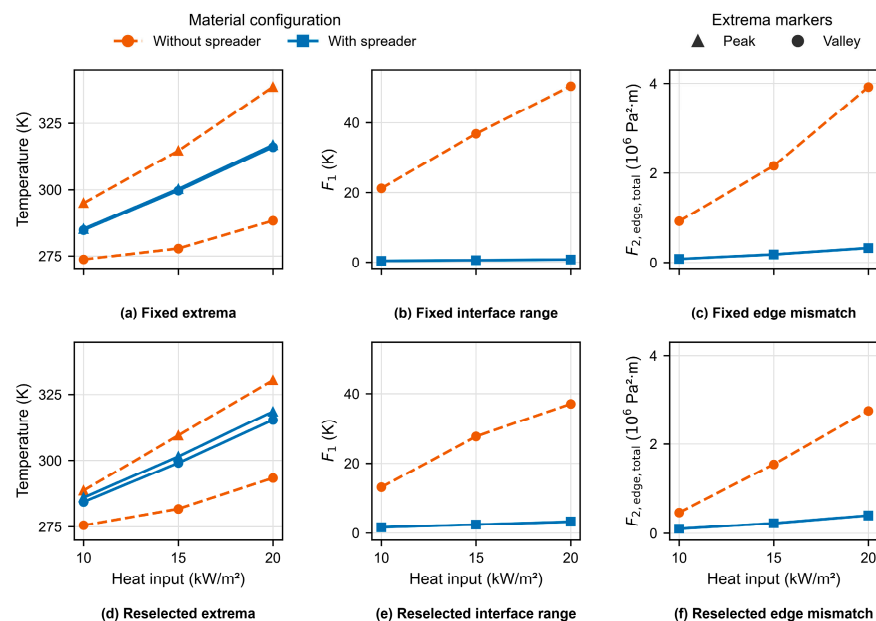


Figure 6. Thermal responses to macroscopic heat input for the fixed-geometry and representative configurations.

Table 4 and the lower row of Figure 6 present representative configurations for the six combinations of heat input and material. Each representative is the feasible Pareto configuration closest to the ideal point after min–max normalization within its own heat-input–material scenario, with equal weights for the three objectives. The NS representative uses $\eta_h = 0.70$ at 10 kW/m^2 and a coverage of 0.60 at both 15 and 20 kW/m^2 ; strip count and film thickness remain $N_w = 6$ and $d_{\text{film}} = 0.020\text{mm}$, respectively. The same GS100 geometry is selected at all three heat inputs, with $\eta_h = 0.20$, $N_w = 4$, and $d_{\text{film}} = 0.010\text{mm}$. Figure 6e shows that F_1 still increases with heat input even though the GS100 representative geometry remains unchanged.

Table 4. Representative configurations for the six macroscopic-heat-input-material-configuration scenarios.

Q_{macro} (kW/m ²)	Configuration	η_h	N_w	d_{film} (mm)	T_{peak} (K)	T_{valley} (K)	F_1 (K)	$F_{2,\text{edge,total}}$ (Pa ² ·m)
10	NS	0.70	6	0.020	288.666	275.542	13.124	463,848.302
10	GS100	0.20	4	0.010	285.991	284.359	1.632	99,667.177
15	NS	0.60	6	0.020	309.393	281.744	27.649	1,544,186.388
15	GS100	0.20	4	0.010	301.452	299.043	2.409	224,149.850
20	NS	0.60	6	0.020	330.377	293.328	37.049	2,743,397.784
20	GS100	0.20	4	0.010	318.461	315.223	3.238	398,392.001

Table 4 and Figure 6d–f show that the GS100 representatives have a coverage of 0.20 and lower interface temperature ranges and cumulative edge metrics than the NS representatives. The spreader therefore helps combine lower active heating coverage with a more uniform temperature distribution, and all selected configurations satisfy the thermal criteria. Compared with the fixed-geometry GS100 configurations, the representatives reduce coverage from 0.50 to 0.20 and also change strip count and film thickness. Their larger interface temperature ranges and cumulative edge metrics illustrate the trade-off associated with lower coverage. The NS representatives instead use higher coverage than the fixed geometry while retaining the same strip count and film thickness, achieving reductions in both metrics.

In Figure 6, the upper-row curves primarily show the effect of heat-input magnitude on a fixed geometry. In the lower row, the NS coverage changes between 10 and 15 kW/m², whereas the GS100 geometry remains unchanged across all three heat inputs. The lower-row responses thus combine heat-input effects with the NS coverage change; the upper row isolates heat-input effects.

3.5. Active Heating Coverage Substitution and Design-Variable Trade-Offs

This main analysis examines whether in-plane heat spreading can reduce the directly heated area without increasing the interface temperature range or the cumulative edge metric. Figure 7 uses the NS Pareto configurations at each heat input as references. For each reference, the lowest-coverage configuration is selected from the GS100 Pareto set at the same heat input, subject to F_1 and cumulative $F_{2,\text{edge,total}}$ being no greater than the reference values. Both configurations satisfy the final-time temperature and response-time criteria, and strip count and heater-film thickness may vary during matching. This gives 13, 20, and 23 matches at 10, 15, and 20 kW/m², respectively. All matched points in Figure 7a lie below the equal-coverage reference line. The mean coverage reductions in Figure 7b are 50.8, 42.8, and 42.6 percentage points, respectively. These results show that coordinating the spreader and heater geometry allows coverage to be reduced while maintaining these performance measures.

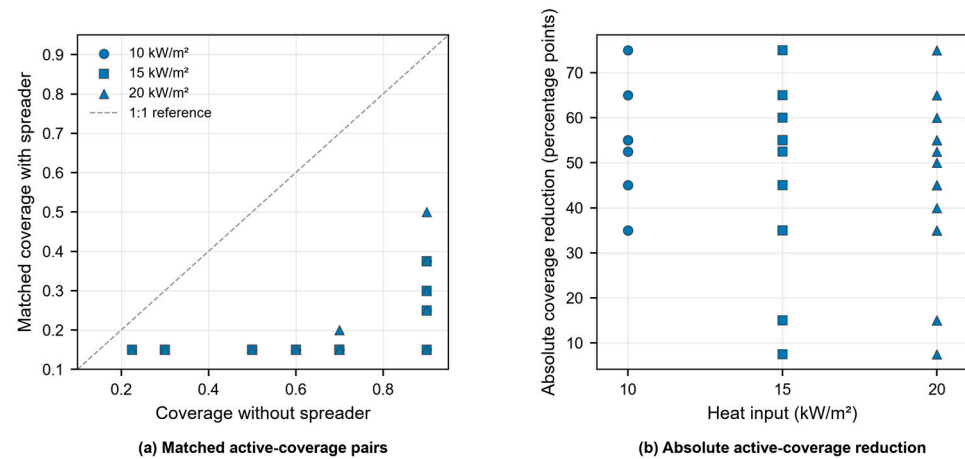


Figure 7. Active-coverage matching. (a) No-spreader and qualifying spreader coverage; (b) absolute coverage reduction.

Figure 8 presents single-variable results for GS100 configurations at 15 kW/m². Each branch varies one design variable while holding the other geometric parameters at the values given in the caption. The blue left axis shows the interface temperature range, and the purple right axis shows the cumulative edge metric.

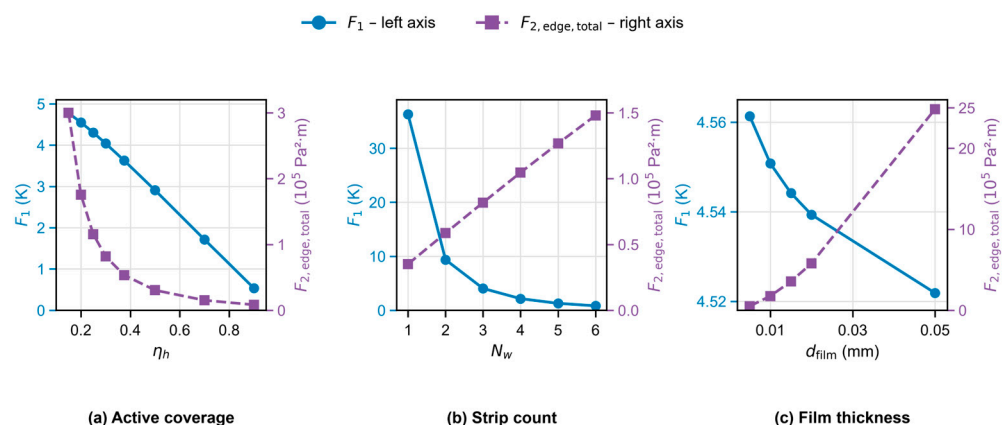


Figure 8. Single-variable responses at 15 kW/m². (a) Coverage: three strips, 0.010-mm film; (b) strip count: coverage 0.30, 0.010-mm film; (c) film thickness: coverage 0.20, three strips.

In Figure 8a, with $N_w = 3$, $d_{\text{film}} = 0.010\text{ mm}$, and macroscopic heat-input fixed, increasing η_h from 0.15 to 0.90 reduces F_1 from 4.776 to 0.535 K and $F_{2,\text{edge,total}}$ from 3.002×10^5 to $8.324 \times 10^3 \text{ Pa}^2\cdot\text{m}$. Greater coverage increases the directly heated area, reduces local heat-source concentration and shortens the unheated gaps, producing a more uniform interface temperature distribution. Coverage selection therefore requires balancing the directly heated area against these two metrics. The matched results in Figure 7 show that the spreader combined with geometry adjustment can shift this balance, allowing lower-coverage configurations to retain values of both metrics no greater than the NS references.

Once coverage is selected, strip count determines the in-plane spatial scale of heat-source distribution. In Figure 8b, with $\eta_h = 0.30$ and $d_{\text{film}} = 0.010\text{ mm}$ fixed, increasing N_w from 1 to 6 reduces F_1 from 36.312 to 0.823 K, while $F_{2,\text{edge,total}}$ increases from 3.509×10^4 to $1.481 \times 10^5 \text{ Pa}^2\cdot\text{m}$. Adding strips shortens the lateral conduction distance. The temperature range decreases rapidly at low strip counts, with progressively smaller absolute reductions as the strips are further subdivided. Meanwhile, the number of edges included

in the cumulative metric increases. The mean metric per edge decreases from 1.755×10^4 to 1.234×10^4 Pa²·m. Increasing strip count therefore requires balancing the additional reduction in temperature range against the increase in the cumulative edge metric.

Film thickness mainly affects the heating process and edge response. In Figure 8c, with $\eta_h = 0.20$ and $N_w = 3$ fixed, increasing d_{film} from 0.005 to 0.050 mm changes F_1 from 4.561 to 4.522 K, while $F_{2,\text{edge,total}}$ increases from 5.017×10^4 to 2.484×10^6 Pa²·m. The endpoint difference in F_1 is only approximately 0.0395 K, indicating little change in interface temperature range over this thickness interval. The minimum temperature decreases and the heating response time increases. Film thickness also changes the film axial stiffness, transfer parameters, and edge-integration window length. Edge count remains constant in this branch, while the width of each window increases from 0.169 to 0.513 mm. Changes in the cumulative metric are governed by both the local edge response and the integration extent. Film thickness should therefore be selected with regard to heating requirements and the cumulative edge metric.

3.6. Nonuniform Heating Configurations Under Flight-Derived Convective Boundaries

This section examines the coupled effects of the spreader and heater layout under nonuniform cooling. The two flight-derived HTC distributions are shown in Figure 9 and summarized in Table 5. Both peak near stagnation and then decrease overall along the surface; the higher-Reynolds-number boundary has a higher HTC at most input nodes. The retained seven- and ten-strip nonuniform layouts are compared with uniform layouts of the same material and strip count. All comparisons use a total active width of 25 mm, a film thickness of 0.020 mm, and a macroscopic heat input of 15 kW/m² to examine spatial heat allocation at equal directly heated areas and heat inputs. The temperature-range reduction is defined as uniform-layout F_1 minus nonuniform-layout F_1 ; a positive value indicates a smaller range for the nonuniform layout.

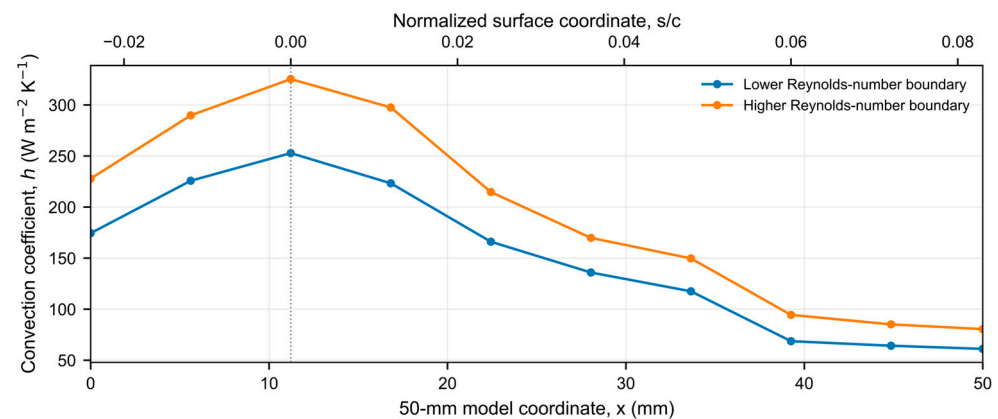


Figure 9. Flight-derived HTC distributions.

Table 5. Prescribed flight-derived convective heat-transfer boundaries.

Boundary	Re	s/c Range	$h_{\min}$	h_{mean}	$h_{\text{mean},50}$	$h_{\max}$
			W/(m ² ·K)	W/(m ² ·K)	W/(m ² ·K)	W/(m ² ·K)
Uniform reference	-	-	15.00	15.00	15.00	15.00
Lower-Re boundary	1.29×10^6	-0.024 to 0.083	61.12	148.97	153.28	252.85
Higher-Re boundary	2.48×10^6	-0.024 to 0.083	80.47	193.48	198.93	325.37

Strip widths need to be determined from the overall thermal response of the multi-layer structure. Heat supplied by a strip travels through the thickness and affects temperatures in adjacent heated regions and gaps. A layer with high in-plane conductivity also redistributes heat laterally across multiple strip-gap units. Strip widths are therefore selected from the coupled transient temperature field under width and thermal constraints.

For the ten-strip nonuniform layout without the spreader, the interface temperature ranges are 20.811 and 20.610 K under the lower- and higher-Reynolds-number boundaries, respectively, compared with 20.581 and 21.937 K for the uniform layouts. The nonuniform layout therefore increases the range by 0.230 K under the lower-Re boundary and reduces it by 1.327 K under the higher-Re boundary. The smaller range under the higher-Re boundary reflects changes in both peaks and valleys caused by the HTC and lateral heat redistribution. Temperature range thus depends on the spatial cooling profile and layout response, as well as HTC magnitude. Applying the seven-strip nonuniform layout to the no-spreader configuration gives ranges of 45.161 and 43.793 K, with minimum temperatures below the thermal criterion under both boundaries. Strip allocation therefore needs to match the material's lateral heat-transfer capability.

With the spreader, the seven-strip nonuniform layout gives interface temperature ranges of 2.229 and 1.712 K under the two boundaries, reducing the corresponding uniform-layout ranges by 1.352 and 2.730 K. The ten-strip nonuniform layout gives ranges of 0.800 and 1.368 K, with reductions of 2.607 and 2.930 K. Both layouts satisfy the thermal criteria and improve temperature uniformity under both boundaries. The seven-strip layout retains fewer strip edges, whereas the ten-strip layout achieves smaller temperature ranges but higher cumulative edge metrics. For the fixed ten-strip layouts, the material-crossed comparison shows that adding the spreader increases the temperature-range reduction from nonuniform allocation by 2.837 and 1.604 K under the lower- and higher-Re boundaries, respectively, further linking layout performance to the spreading material. At the same strip count, the cumulative edge metrics of both the seven- and ten-strip spreader nonuniform layouts are also lower than those of their respective uniform references. Edge count is unchanged within each pair, so this change reflects the local thermal-mismatch response to the redistribution of strip and gap widths.

The Reynolds numbers, retained coordinate ranges, and HTC statistics of the two prescribed boundaries are summarized in Table 5. The node mean is the arithmetic mean of the ten digitized HTC values; the spatial mean is the integral of the piecewise-linear HTC distribution divided by the 50-mm domain width.

Figure 10 compares strip positions and interface temperature profiles for the NS ten-strip and GS100 seven-strip layouts, together with their respective uniform references. Without the spreader, the profiles contain both strip-scale oscillations and longer-scale variation associated with cooling. Dense strip subdivision shortens local lateral conduction distances and reduces the temperature drop within gaps. With the spreader, lateral conduction attenuates strip-scale oscillations, while nonuniform strip and gap widths adjust heat allocation over longer scales to accommodate spatial HTC variation. Together with the material-crossed comparison at the same ten-strip geometry, these profiles show that the spreader reduces the temperature range and changes the spatial scale over which strip allocation acts.

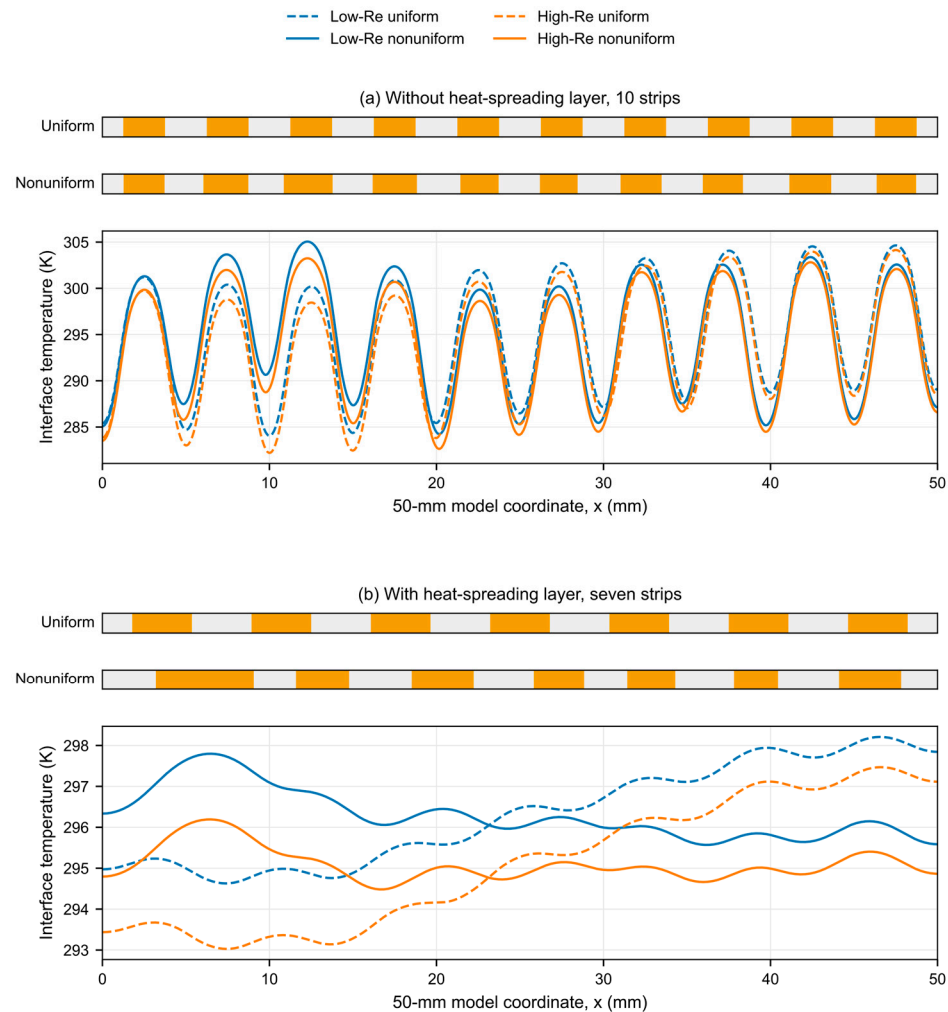


Figure 10. Strip positions and interface temperatures at 60 s. (a) No spreader, ten strips; (b) spreader, seven strips. Dashed/solid curves denote uniform/nonuniform layouts.

For GS100-R, the same seven-strip geometries and a contact resistance of $5.8 \times 10^{-4} \text{ m}^2\text{-K/W}$ on each spreader face give nonuniform-layout temperature-range reductions of 2.129 and 3.077 K under the two boundaries. Relative to the same nonuniform layout in GS100, adding contact resistance lowers the minimum temperature and delays the heating response. Layout selection with contact resistance should therefore consider the temperature-range reduction, temperature level, and heating process together.

The lateral-boundary comparison further examines the influence of surrounding unheated regions on layout performance. Adding unheated regions on both sides gives total widths of 75 and 100 mm, while the evaluated region spreader remains the central 50-mm and heater positions and total input remain fixed. The spreader extends continuously into the added regions, the peripheral HTC retains the original endpoint values, and the outer boundaries are adiabatic. Under the lower-Re boundary, the GS100-NU10 temperature-range reduction changes from 2.607 K in the 50-mm domain to 1.144 and 1.089 K in the 75- and 100-mm domains. Increasing total width from 75 to 100 mm changes F_1 and minimum temperature by up to 2.061 and 2.977 K across the examined configurations. All calculated extended NS configurations have minimum temperatures below the thermal criterion. Lateral thermal connections to surrounding regions therefore affect temperature levels and layout performance, and geometry should be evaluated with specified surrounding thermal boundaries.

The source-scale HTC comparison examines the influence of mapping on layout comparisons. For the lower-Re boundary, the domain-mean HTC changes from 153.278 to 162.309 W/(m²·K). Under the lower-Re boundary, GS100-NU10 gives temperature-range reductions of 2.607 and 2.498 K for the mapped and source-scale distributions, respectively; NS-NU10 gives −0.230 and −0.513 K. Under both mappings, the reduction remains positive for the ten-strip spreader layout and negative for the ten-strip no-spreader layout. Its magnitude changes as the sampled interval and mean cooling intensity change together.

Once the spreader attenuates local oscillations, strip and gap allocation can further compensate for temperature variations associated with the cooling distribution. Increasing strip count also requires consideration of the cumulative edge metric. Layout selection should therefore account for lateral heat-transfer capability, interface contact conditions and surrounding thermal connections. This joint material–layout design problem also connects with research on structurally integrated electrothermal anti-icing components and electrothermal anti-/de-icing materials.

4. Conclusions

The study combined active heater-layout design with a passive orthotropic in-plane heat-spreading layer under fixed macroscopic heat input. A two-dimensional transient multilayer model was used to examine material configuration, active heating coverage, strip count, heater-film thickness, and nonuniform strip-gap allocation. The principal findings are:

1. Discrete heater strips produced strip–gap-scale temperature variations at the ice/composite interface, whereas the heat-spreading layer reduced the temperature peaks and raised the valleys by enhancing lateral heat transport. In the matched-geometry comparison at $Q_{\text{macro}} = 15 \text{ kW/m}^2$, F_1 decreased from 36.807 to 0.587 K, $F_{2,\text{edge,total}}$ decreased from 2.161×10^6 to $1.829 \times 10^5 \text{ Pa}^2\cdot\text{m}$, and the integrated lateral heat-flow magnitude increased from 1.532 to 26.400 W/m.
2. Matched-geometry comparisons isolated the material effect of the heat-spreading layer, while reselection of the design variables quantified its configuration effect. At all three macroscopic heat inputs, the paired results showed a typically lower maximum temperature, higher minimum temperatures, and smaller F_1 , indicating consistent heat-spreading behavior within the sampled design domain.
3. The active heating coverage η_h determined the directly heated fraction and heat-source concentration, N_w controlled the strip–gap length scale, and d_{film} affected both the heating process and the cumulative edge thermal-mismatch metric. Under performance-preserving matching for both F_1 and $F_{2,\text{edge,total}}$, active heating coverage decreased by an average of 50.8, 42.8, and 42.6 percentage points at 10, 15, and 20 kW/m², respectively. Adding strips improved temperature uniformity while increasing edge count, requiring local edge response and the cumulative metric to be considered together.
4. Under the two flight-derived nonuniform convective boundaries, the ten-strip no-spreader nonuniform layout gave temperature-range reductions of −0.230 and 1.327 K, respectively, whereas the seven-strip spreader nonuniform layout reduced F_1 by 1.352–2.730 K relative to the uniform layouts with the same strip count. The seven-strip spreader layout also had lower cumulative $F_{2,\text{edge,total}}$ than its uniform references, and the temperature profiles show how passive spreading changes the spatial scale of variation addressed by the active layout. These quantitative improvements apply to the prescribed 50-mm domain. For the lower-Reynolds-number ten-strip

spreader case, extending the total width to 75 and 100 mm while retaining the central 50-mm evaluation region reduced the benefit from 2.607 K to 1.144 and 1.089 K, respectively. Layout benefits therefore depend strongly on the surrounding lateral thermal connections.

5. The matched-geometry and boundary-tailored results support joint selection of material configuration, active heating coverage, strip count, and strip-gap allocation under nonuniform external cooling. The model uses effective layer properties, prescribed contact resistance and phase change in a fixed ice layer. Quantitative edge-metric improvements remain sensitive to mesh resolution, and the representatives were selected using the cumulative objective without repeating Pareto selection with the per-edge metric. Further work will combine transient temperature experiments with actual laminate, bonding, and insulation properties, and examine electrical power allocation and ice–water evolution.

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Abbreviations

The following abbreviations are used in this manuscript:

AIAA	American Institute of Aeronautics and Astronautics
GFRP	Glass-fiber-reinforced polymer
HTC	Heat-transfer coefficient
NASA	National Aeronautics and Space Administration
TM	Technical Memorandum

Appendix A

Dimensions of the Retained Nonuniform Heater Layouts

Reading order: Gap 0, Strip 1, Gap 1, Strip 2, and so on, ending with Gap 7 or Gap 10. A dash denotes an element that does not exist in that layout. The widths are displayed to eight decimal places; the original full-precision layout files are retained.

Both layouts have a total active width of 25 mm and a total gap width of 25 mm over the 50-mm domain. Each strip and gap is at least 1-mm wide. Adjacent strip-width and adjacent gap-width ratios satisfy the prescribed limit of 3.

Table A1. Strip and gap widths of the retained seven- and ten-strip layouts. All dimensions are in millimeters.

Element	Seven-Strip Layout	Ten-Strip Layout
Strip 1	5.86202540	2.45250459
Strip 2	3.18365431	2.68894713
Strip 3	3.70919316	2.92450419
Strip 4	3.01151862	2.65167579
Strip 5	2.87469544	2.29108647

Strip 6	2.63670955	2.25783883
Strip 7	3.72220353	2.45991443
Strip 8	—	2.40542512
Strip 9	—	2.50667755
Strip 10	—	2.36142589
Gap 0	3.19928194	1.26788908
Gap 1	2.52586109	2.32821354
Gap 2	3.74877271	2.12052135
Gap 3	3.60153953	2.39720377
Gap 4	2.58168645	2.59864576
Gap 5	3.51632425	2.47040673
Gap 6	3.64962963	2.57012845
Gap 7	2.17690441	2.46639485
Gap 8	—	2.77867614
Gap 9	—	2.71890598
Gap 10	—	1.28301435
Total strip width	25.00000000	25.00000000
Total gap width	25.00000000	25.00000000

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