

Article

Finite Element Analysis of Thermal Frictional Contact Characteristics of a Functionally Graded Coated Brake Disc

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Abstract

To address the issues of local high temperatures, thermal stress concentration, and the susceptibility to spalling of homogeneous ceramic coatings in disc brakes under high-frequency thermal–mechanical cyclic loading, this paper proposes a surface design scheme incorporating a functionally graded material (FGM) coating along the thickness direction. A three-dimensional thermal frictional contact model of a graded coated brake disc with continuously varying material properties (silicon carbide/gray cast iron) along the thickness direction is established by developing user subroutines on the Abaqus finite element platform. The effects of exponential, power-law, and trigonometric gradient distributions on the transient temperature and stress fields are systematically compared. The results indicate that the high thermal conductivity silicon carbide coating significantly reduces the disc surface temperature; however, a homogeneous coating induces interfacial thermal stress concentration due to a sudden stiffness mismatch. The graded design effectively mitigates the stress concentration through a smooth transition of material properties. Taking the power-law function ($n = 1.5$) as an example, this design not only significantly reduces the maximum disc surface temperature but also limits the residual equivalent stress at the end of braking to 245 MPa, which is approximately 24.8% lower than that of the homogeneous coating (325.8 MPa). The study demonstrates that the gradient function exerts a stronger regulatory effect on the stress field than on the temperature field, meaning the two cannot be simultaneously optimized. Nevertheless, exponential functions and power-law functions with small exponents can achieve a favorable balance of thermal–mechanical performance. This research reveals the mechanism by which thickness-direction gradient distributions regulate thermal–mechanical coupling behavior, providing a theoretical basis for the gradient design of thermal fatigue-resistant friction components.

Keywords: functionally graded materials; disc brake; thermo-mechanical coupling stress field; contact mechanics

Received: 29 May 2026

Revised: 25 June 2026

Accepted: 29 June 2026

Published: 30 June 2026

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1. Introduction

The brake is a critical component for vehicle safety, subjected to severe thermal–mechanical cyclic loads during service. The braking process converts the vehicle’s kinetic energy into frictional heat, and the transient injection of high-density heat flux causes

rapid temperature rise, thermal expansion, and significant thermal stress in the brake disc. Uneven heat distribution can lead to localized hot spots and non-uniform expansion, resulting in stress concentration and the initiation of fatigue cracks, which seriously threatens the reliability of the braking system [1]. Therefore, enhancing the thermodynamic service performance of brake discs through surface modification has become a research hotspot.

The brake disc must simultaneously meet requirements such as structural strength, stiffness, a stable friction coefficient, good thermal conductivity, and a low coefficient of thermal expansion [2–4]. SiC ceramic is an ideal friction coating material due to its high rigidity, high temperature resistance, wear resistance, and high thermal conductivity [5–8]. However, the elastic modulus of a homogeneous ceramic coating is significantly higher than that of the metal substrate, which can cause an abrupt change in surface contact stiffness, leading to severe stress concentration and interfacial delamination [9]. Won et al. [10] established a multi-scale friction model revealing the influence of surface micro-topography on frictional heat generation and contact stress, indicating that rational surface and interfacial design is crucial for regulating braking performance.

With the advancement of materials science and surface engineering, metal matrix composites, bimetallic composite brake discs, and laser-clad coatings have been widely utilized to enhance the service reliability of brake discs under extreme thermal-mechanical conditions [11–14]. Functionally graded materials (FGMs) eliminate macroscopic physical interfaces by continuously varying the volume fraction of constituent materials along a specific spatial direction, thereby significantly alleviating stress concentration caused by material property mismatches [15,16]. In the field of braking systems, Hernik and Ganczarski [17] were among the first to propose the use of FGMs for brake discs; Gupta and Talha [18] reviewed the mechanical behavior of FGM structures in thermal environments, noting that gradient index optimization is a key design factor. Yevtushenko et al. [19,20] derived partial analytical solutions for frictional heat generation in FGMs. Peng et al. [21] proposed an integral equation method to solve the thermoelastic problem of FGM rotating disks with arbitrary radial gradients and variable thickness. Attia and Shanab [22] established a nonlinear analysis model to reveal the coupled regulation mechanism of thickness distribution on the thermal stress field. Bayat et al. and Shahzamanian et al. [23,24] investigated the contact response and steady-state thermal behavior of brake discs with radially varying material parameters using the finite element method. Madan and Bhowmick [25] further fabricated functionally graded disks with a radially varying volume fraction of reinforcing particles using powder metallurgy.

However, existing studies have predominantly focused on the radial gradient variation in material properties in brake discs. During actual braking, the temperature difference and thermal stress gradient along the thickness direction (i.e., the normal depth direction) are much steeper and more critical to failure. Gradient structural design along the thickness direction holds significant potential for regulating heat dissipation and alleviating surface shear stress, yet its underlying regulation mechanism remains insufficiently investigated. To address this gap, this paper establishes a full-scale three-dimensional thermomechanical coupled finite element model based on the ABAQUS software (ABAQUS2019). This is achieved by developing user subroutines that define key parameters such as elastic modulus, coefficient of thermal expansion, and thermal conductivity as functions of the thickness-direction coordinate. The influence of different gradient functions on the thermomechanical coupling behavior of brake discs is systematically studied, revealing the synergistic regulation mechanism of gradient distribution on temperature and stress.

2. Material Model and Theoretical Basis

2.1. Theoretical Governing Equations

As shown in Figure 1, the composite brake disc studied in this study consists of gray cast iron and a functionally graded (FGM) surface coating with a thickness of h . During vehicle braking, the brake disc rotates with the wheels at an initial angular velocity and is subjected to a symmetrically applied axial normal contact pressure $p(r, t)$ from the brake pads on both sides. To emphasize the influence of the thickness-direction gradient on the multi-field coupling response and to simplify computational efforts, a three-dimensional axisymmetric thermal friction contact model is established. It is assumed that the brake disc decelerates at a constant angular deceleration until it completely stops, and the centrifugal inertia effect is neglected. The kinetic energy of the system is converted into thermal energy through friction at the rough interface. According to tribological contact theory, the transient heat flux generated by frictional work on the coating surface ($z = 0$) can be defined by the following equation:

$$q(r, t) = \eta f p(r, t) r \omega(t) \quad (1)$$

where η is the heat partition coefficient, f is the interface friction coefficient, $p(r, t)$ is the normal contact pressure, $\omega(t)$ is the time-decaying instantaneous angular velocity, and r is the radial coordinate on the disc surface. In the present model, the heat partition coefficient is set to 0.25. This value is determined based on the significant thermal capacity and conductivity differences between the high-conductivity disc (metallic substrate with SiC coating) and the organic-based brake pads, which assigns 25% of the generated frictional heat to the pads, with the remaining 75% entering the disc [26].

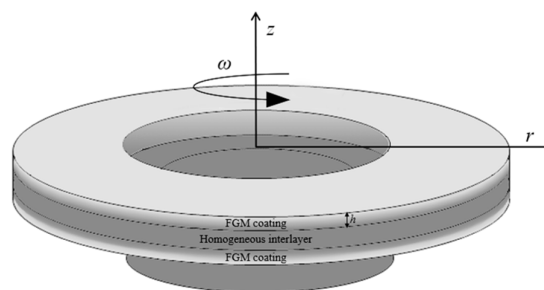


Figure 1. Schematic diagram of the brake disc during braking.

In addition to the frictional heat flux input on the contact area, the non-contact surfaces (such as the inner hole walls and outer cylindrical surface) undergo convective heat exchange with the ambient medium ($T_0 = 20^\circ\text{C}$). The convective heat transfer intensity between the fluid and the solid is generally described by the convective heat transfer coefficient. For disc brake problems, the empirical formula for convective heat transfer coefficient H_{conv} is given as

$$H_{conv} = \begin{cases} 0.70 \frac{k_a}{D} Re^{0.55}, & Re < 2.4 \times 10^5 \\ 0.04 \frac{k_a}{D} Re^{0.8}, & Re > 2.4 \times 10^5 \end{cases} \quad (2)$$

where k_a is the thermal conductivity of air, $Re = \rho_a v L / \mu_a$ is the Reynolds number [26–28]. Since the characteristic fluid velocity v varies with the radius, the velocity at the

average of the inner and outer radii is taken as the characteristic fluid velocity. L is the characteristic length of the fluid, i.e., the outer diameter of the brake disc D . Thus, the formula is transformed into

$$Re = \frac{\omega R \rho_a D}{\mu_a} \quad (3)$$

where ω is the angular velocity of the brake disc, ρ_a is the air density, R is the average of the inner and outer radii of the brake disc, and μ_a is the dynamic viscosity of air. In this model, the convective heat transfer coefficient is not a constant but a function of the Reynolds number. This approach is adopted to accurately capture the transient cooling effect as the angular velocity of the disc continuously decrease to zero during braking, which a linear constant convection law would fail to reflect

For the composite brake disc structure consisting of the aforementioned FGM coating and the substrate, the transient heat conduction equation can be expressed as

$$\frac{1}{r} \frac{\partial}{\partial r} \left(k_i r \frac{\partial T_i}{\partial r} \right) + \frac{\partial}{\partial z} \left(k_i \frac{\partial T_i}{\partial z} \right) = \rho_i c_i \frac{\partial T_i}{\partial t} \quad (4)$$

where the subscripts $i=1,2$ refer to the FGM coating and the substrate, respectively; T_i is the temperature field; and k_i , c_i , ρ_i represent the corresponding thermal conductivity, specific heat capacity, and density, respectively.

The thermal boundary condition on the coating surface ($z=0$) of the brake disc is governed by the frictional heat flux input:

$$-k_1 \frac{\partial T_1}{\partial z} = q(r, t) \quad (5)$$

At the coating–substrate interface ($z=-h$), the continuity conditions for the temperature field and heat flux density are satisfied:

$$T_1(r, -h, t) = T_2(r, -h, t) \quad (6)$$

$$k_1 \frac{\partial T_1}{\partial z} = k_2 \frac{\partial T_2}{\partial z} \quad (7)$$

In the cylindrical coordinate system (r, θ, z) , neglecting the influence of body forces, the three-dimensional axisymmetric thermoelastic equilibrium equation for the composite structure is

$$\begin{aligned} \frac{\partial \sigma_{rr}}{\partial r} + \frac{\partial \sigma_{rz}}{\partial z} + \frac{\sigma_{rr} - \sigma_{\theta\theta}}{r} &= 0 \\ \frac{\partial \sigma_{rz}}{\partial r} + \frac{\partial \sigma_{zz}}{\partial z} + \frac{\sigma_{rz}}{r} &= 0 \end{aligned} \quad (8)$$

The thermoelastic constitutive relations are

$$\sigma_{ri}(r, z, T_i) = (\lambda_i + 2\mu_i) \frac{\partial u_i}{\partial r} + \lambda_i \left(\frac{u_i}{r} + \frac{\partial w_i}{\partial z} \right) - 3\alpha_i K_i T_i \quad (9)$$

$$\sigma_{zi}(r, z, T_i) = (\lambda_i + 2\mu_i) \frac{\partial w_i}{\partial z} + \lambda_i \left(\frac{u_i}{r} + \frac{\partial u_i}{\partial r} \right) - 3\alpha_i K_i T_i \quad (10)$$

$$\sigma_{\theta i}(r, z, T_i) = \mu_i \left(\frac{\partial u_i}{\partial r} + \frac{\partial w_i}{\partial z} \right) \quad (11)$$

where $\lambda_i = \nu_i E_i / (1 + \nu_i)(1 - 2\nu_i)$ and $\mu_i = E_i / 2(1 + \nu_i)$ are Lamé constants, $K_i = E_i / 3(1 - 2\nu_i)$ is the bulk modulus, and u_i and w_i are the displacements in the r - and z -directions, respectively.

At the coating-substrate interface $z = -h$, the displacement continuity and stress continuity conditions are satisfied:

$$u_2 = u_1, w_2 = w_1 \quad (12)$$

$$\sigma_{zz2} = \sigma_{zz1}, \sigma_{rz2} = \sigma_{rz1} \quad (13)$$

2.2. Material Parameters and Distribution Functions of the Graded Coating

In this model, a high-thermal-conductivity silicon carbide (SiC) ceramic graded coating is applied to the surface of a gray cast iron brake disc. The substrate is made of gray cast iron, and the outermost coating surface is pure SiC ceramic. The specific material parameters are listed in Table 1. To isolate the core influence of the spatial gradient function on the thermodynamic response, this model assumes that the material properties vary primarily with the thickness coordinate, temporarily neglecting the thermal softening effect of material properties with temperature.

Table 1. Thermo-mechanical properties of the brake disc materials.

Material	Thermal Conductivity (W/(m·K))	Specific Heat (J/(kg·K))	Coefficient of Thermal Expansion ($10^{-5}/K$)	Density (kg/m ³)	Elastic Modulus (GPa)	Poisson's Ratio
(Substrate)	57	460	1.085	7250	138	0.28
SiC ceramic coating	250	640	0.44	3200	466	0.2

To investigate the influence of the coating material's gradient variation, a field variable dependent on the spatial coordinate is introduced via the Abaqus user subroutine USDFLD. This enables the continuous definition of material properties along the thickness direction (z -direction). The thickness coordinate z is defined as $z = 0$ at the coating surface and $z = -h$ at the coating-substrate interface, where h is the coating thickness. Let $P(z)$ denote an arbitrary material property (elastic modulus, coefficient of thermal expansion, or thermal conductivity), with P_1 representing the SiC parameter and P_2 representing the substrate parameter. The gradient distribution functions are given as follows:

(1) Exponential function distribution:

$$P(z) = P_2 \exp\left(\frac{z+h}{h} \ln \frac{P_1}{P_2}\right) \quad (14)$$

(2) Power-law function distribution (with n as the gradient index):

$$P(z) = P_2 + (P_1 - P_2) \left(\frac{z+h}{h}\right)^n \quad (15)$$

(3) Sine function distribution:

$$P(z) = P_2 + (P_1 - P_2) \sin\left(\frac{\pi(z+h)}{2h}\right) \quad (16)$$

(4) Cosine function distribution:

$$P(z) = P_2 + (P_1 - P_2) \left[1 - \cos\left(\frac{\pi(z+h)}{2h}\right)\right] \quad (17)$$

2.3. Finite Element Model Establishment and Validation

This study utilizes a full-scale three-dimensional brake disc model for calculation to avoid the artificial, sudden changes in thermal stress that may occur at the symmetry boundaries of a quarter model. Braking energy input is simulated by applying a time-varying heat flux density based on the theoretical conversion of frictional power generation. Key simulation parameters include an initial braking angular velocity $\omega_0 = 300$ rad/s, a braking time $t = 4$ s, and axial normal contact pressure $p(r, t) = 1.3$ MPa, a friction coefficient $f = 0.35$, and ambient temperature $T_0 = 20$ °C.

To ensure both accuracy and computational efficiency, a mesh independence test was conducted. A mesh independence curve (Figure 2) was obtained by monitoring the convergence relationship between the maximum surface stress and the number of elements. The results indicate that the calculated values stabilize when the characteristic mesh size is between 1 and 2 mm. The final model consists of 139,452 three-dimensional eight-node thermomechanically coupled hexahedral elements (C3D8T). The established finite element model is depicted in Figure 3.

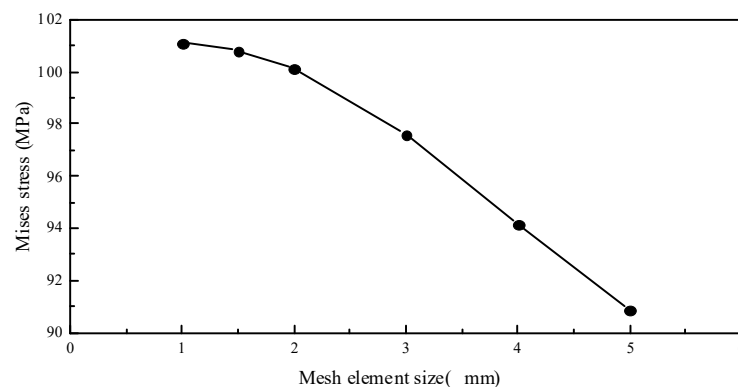


Figure 2. Mesh independence verification curve.

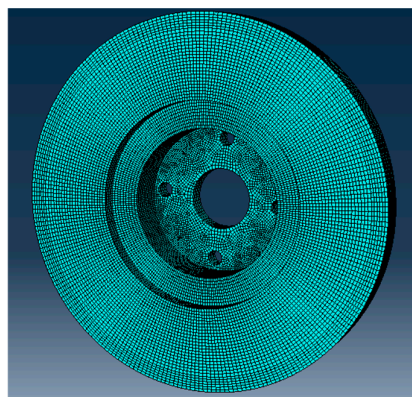


Figure 3. Finite element model of the brake disc.

To validate the reliability of the model, this paper reproduced the operating conditions described by Habtamu Dubale et al. [1] (Figure 4). It is worth noting that the literature results show an extremely high stress zone near the bolt holes connecting the brake disc and the hub. Analysis indicates that this phenomenon is caused by singular stresses induced by an overly rigid artificial mechanical fixation boundary, rather than the true thermoelastic response under actual service conditions. During the reproduction process, a kinematic coupling treatment was applied to the inner hole boundary to increase flexible, thereby releasing the over-constraint conditions. Following this improvement, the tem-

perature field (maximum value approximately 211 °C vs. 214 °C) and thermal stress distribution characteristics in the frictional working zone of the disc surface demonstrate excellent agreement with the literature, thus verifying the accuracy of the present model for high-gradient thermomechanical coupling calculations.

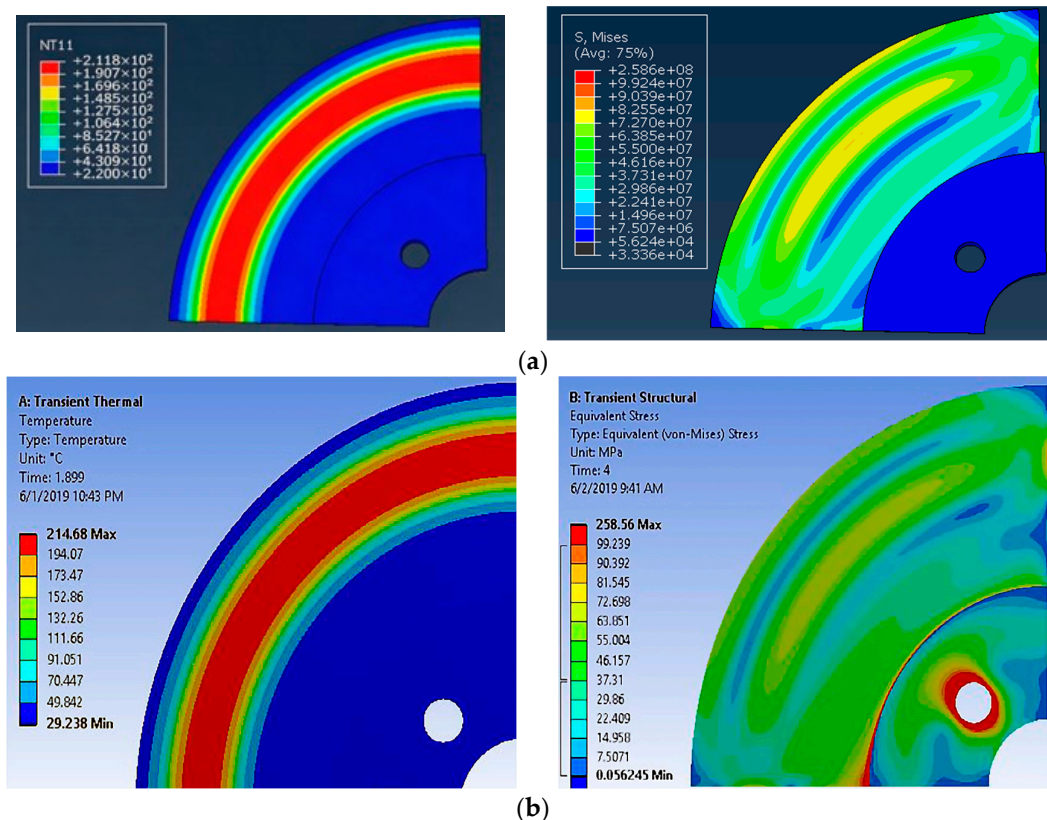


Figure 4. Comparison of temperature and stress fields between the present results (a) and those in the literature (b).

3. Results and Discussion

3.1. Performance Analysis of the Brake Disc with an Exponentially Graded Coating

Figure 5 illustrates the time-history responses of the maximum temperature and stress for brake discs featuring either a homogeneous SiC coating or an exponentially graded coating. The homogeneous ceramic coating achieves a notable cooling effect—reducing the maximum temperature by approximately 10—due to its high thermal conductivity. However, the abrupt interface of this hard coating/soft substrate configuration leads to sharp increase in disc surface stress, reaching a maximum stress of 346.6 MPa during braking. The FGM exponentially graded coating effectively mitigates this issue; while providing a considerable cooling effect, it efficiently suppresses the peak stress to 235.8 MPa through a smooth transition of the elastic modulus.

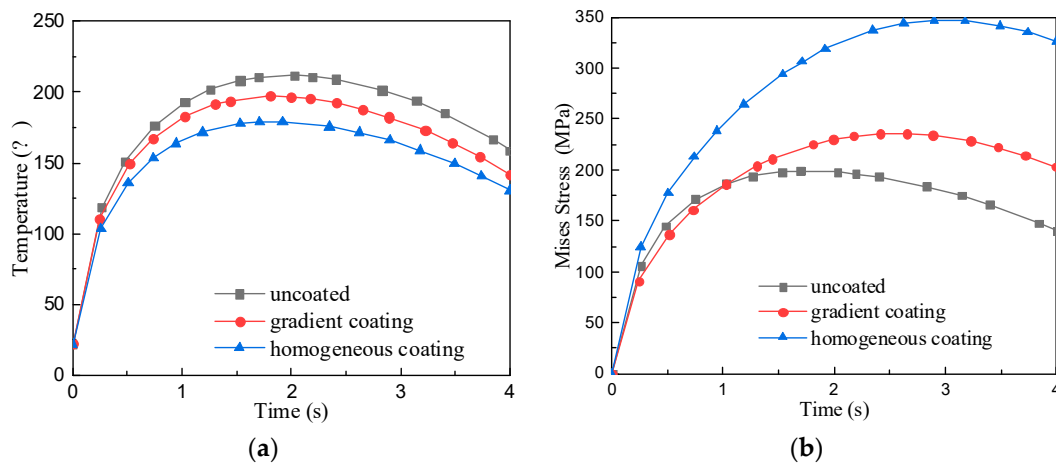


Figure 5. Effect of the coating on the surface temperature (a) and stress (b) of the brake disc.

Figure 6 shows the effect of coating thickness on the maximum surface temperature and stress of the brake disc with an exponentially graded coating. Increasing the coating thickness further reduces the disc surface temperature and stress during the initial stage. However, when the thickness exceeds 1 mm, the marginal benefit of temperature improvement decreases significantly. According to the interface cracking criterion in contact mechanics, an excessively thick coating is more prone to delamination failure under thermal cycling due to its high overall stiffness [29]. Therefore, the optimal coating thickness is set to 1 mm for subsequent investigations in this study.

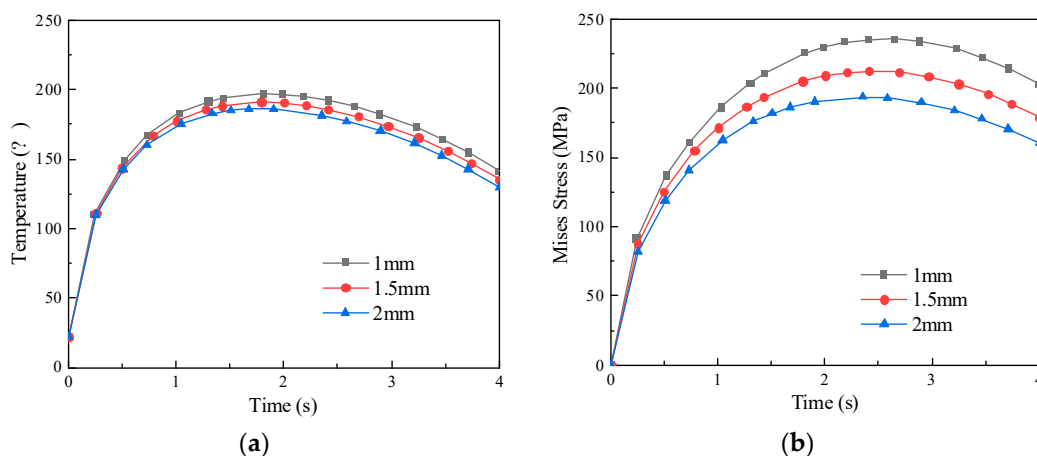


Figure 6. Effect of coating thickness on the surface temperature (a) and stress (b) of the brake disc.

3.2. Sensitivity Analysis of Material Properties on Contact Mechanical Behavior

To isolate the independent effects of coating material parameters on the thermo-mechanical coupling behavior of the brake disc, a sensitivity analysis was conducted by individually varying the elastic modulus, thermal conductivity, and coefficient of thermal expansion of the coating, using the gray cast iron substrate as a reference. As shown in Figure 7, variations in the elastic modulus and coefficient of thermal expansion have a negligible influence on the transient temperature field distribution of the disc surface but significantly impact the thermal stress field. In contrast, thermal conductivity not only directly determines the rate of heat dissipation into the substrate—thus affecting the temperature field—but also effectively improves the macroscopic stress field by smoothing the thermal gradient.

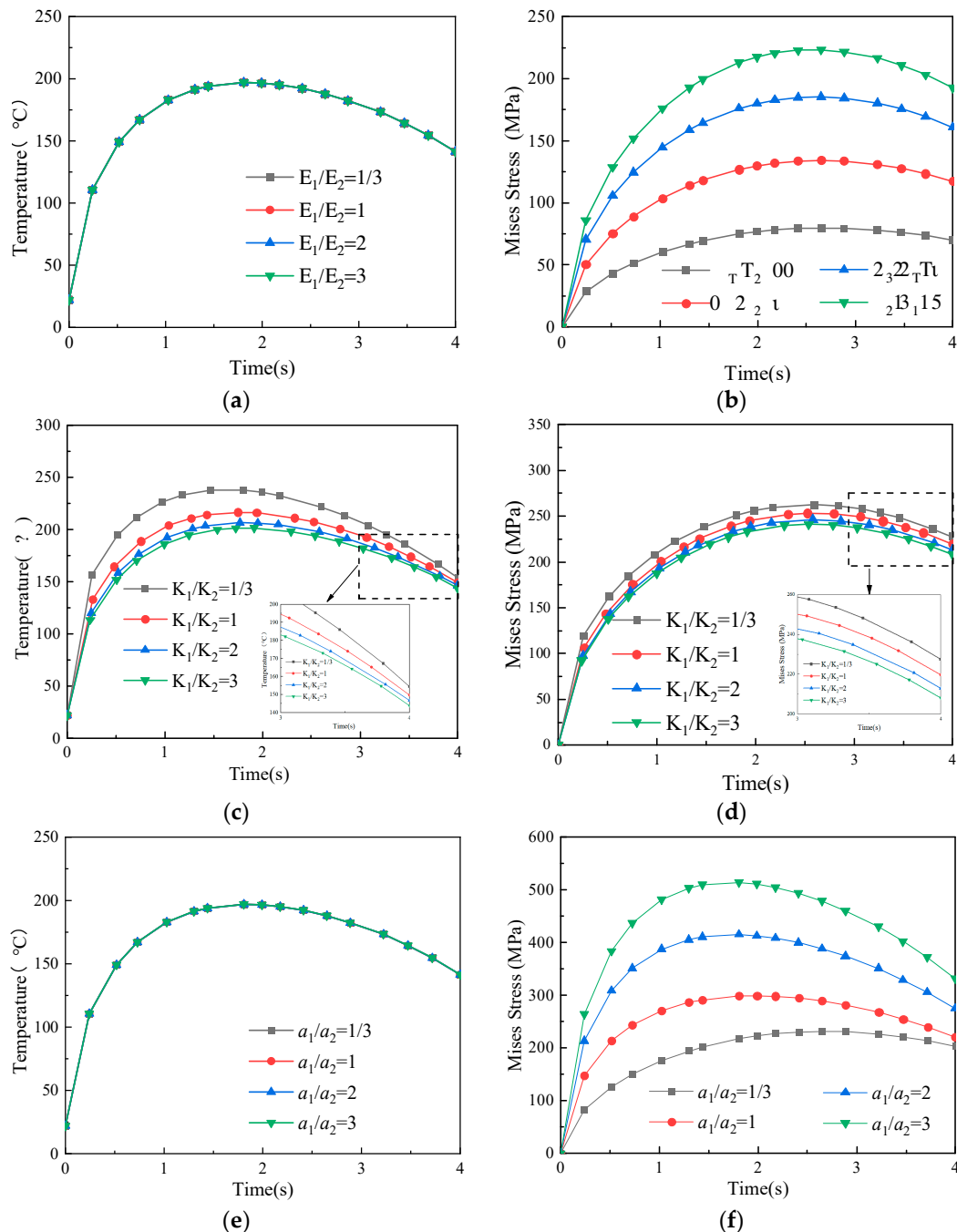


Figure 7. Effects of coating material properties on maximum surface temperature and stress of the brake disc: (a) temperature and (b) stress varying with elastic modulus; (c) temperature and (d) stress varying with thermal conductivity; (e) temperature and (f) stress varying with the coefficient of thermal expansion.

These analyses indicate that the sharp increase in disc surface stress induced by the ceramic coating is mechanically driven by the material's extremely high elastic modulus. This high modulus causes a sudden increase in local contact stiffness at the surface layer, leading to mechanical stress concentration in the contact region. Therefore, in practical gradient design, priority should be given to optimizing the elastic modulus gradient to minimize stress, followed by optimizing the thermal conductivity gradient to control temperature.

It should be emphasized that the decoupled sensitivity analysis presented above serves strictly as a numerical parametric study to isolate the independent contribution of each material property. In a physically fabricated functionally graded coating, however, the effective elastic modulus, thermal conductivity, coefficient of thermal expansion, and strength are inherently coupled through the underlying microstructure (e.g., SiC volume fraction and porosity). Independent variation in a single property cannot be achieved in practice without simultaneously altering other properties.

3.3. Performance Analysis of Brake Disc with Power-Law Function Gradient Coating

By adjusting the gradient index n of the power-law gradient coating, the evolution of the transient temperature field and stress response on the brake disc surface can be observed, as shown in Figure 8. As the gradient index n increases, the maximum temperature of the disc surface shows a significant decreasing trend, while the maximum thermal stress increases monotonically, highlighting an obvious mutual restriction relationship between the two.

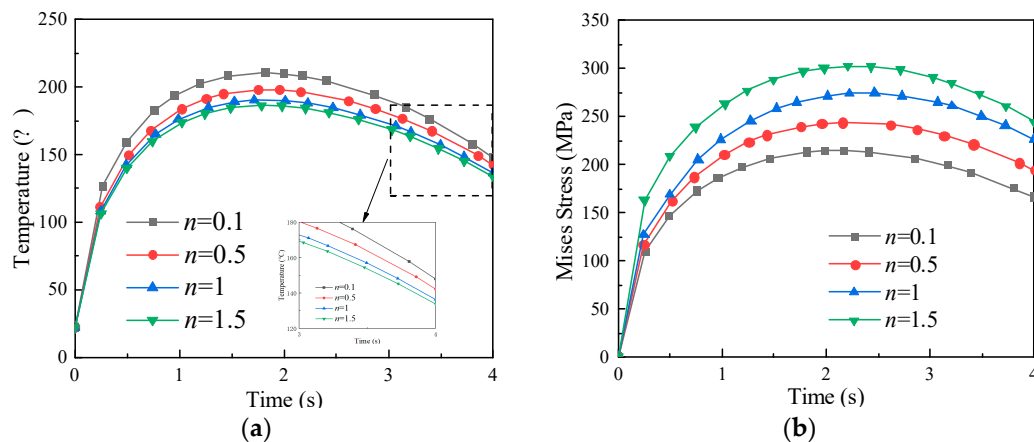


Figure 8. Influence of the power-law gradient index n on the surface temperature (a) and stress (b) of brake disc.

The physical essence of this inverse evolution between temperature and stress lies in the fact that the gradient index n dictates the gradient distribution form of the material properties within the coating along the thickness direction. When n is small (e.g., $n = 0.1$), the material physical properties present a concave distribution along the thickness. In the shallow region close to the coating surface, the high elastic modulus and high thermal conductivity of ceramic materials decay rapidly. From a contact mechanics perspective, this rapid attenuation of elastic modulus forms a flexible buffer layer within the coating, which effectively reduces the local contact stiffness of the surface layer, thereby greatly relieving thermal stress. From a heat transfer perspective, however, the sharp decrease in thermal conductivity increases the local thermal resistance within the coating. This hinders the rapid conduction of frictional heat to the substrate, resulting in a relatively high peak temperature.

Conversely, when the gradient index n increases (e.g., $n = 1.5$), the distribution of material properties transforms into a convex pattern, causing the ceramic-dominated region with high thermal conductivity and high rigidity to extend deeper into the coating. This effectively broadens the heat conduction path, achieving a significant temperature reduction. However, the thicker surface layer with a high elastic modulus inevitably enhances the overall rigid constraint of the structure, ultimately leading to a remarkable increase in macroscopic thermal stress at the interface.

3.4. Regulation Mechanism of Gradient Function Form on Thermo-Mechanical Response of Coated Brake Disc

To evaluate the comprehensive influence of different coating gradient forms on brake discs performance, Tables 2 and 3 summarize the surface temperature and stress data during the braking process under different distribution functions, when a large-exponent power-law function ($n = 1.5$) is adopted, the brake disc exhibits an optimal cooling effect due to the large proportion of high-thermal-conductivity ceramic materials in the surface layer. Its maximum temperature during braking and the temperature at the end of braking are effectively suppressed to 186.4 °C and 133.9 °C, respectively. Nevertheless, this distribution inevitably enhances the mechanical constraint of the surface layer, causing the maximum thermal stress on the disc surface to rise to 302.7 MPa. Therefore, $n = 1.5$ can be considered an upper-bound design where cooling performance is prioritized, accepting a moderate increase in thermal stress compared to smaller exponents.

Table 2. Surface temperatures of the brake disc with different coating gradient forms (°C).

Gradient Form	Exponential Function	Uniform Layer	Power Law ($n = 0.1$)	Power Law ($n = 0.5$)	Power Law ($n = 1$)	Power Law ($n = 1.5$)	cos	sin
Maximum temperature during braking	197	178.9	210.7	198	190.3	186.4	190.3	197.7
Temperature at braking termination	141.4	130.5	148	142.3	136.4	133.9	136.4	140.6

Table 3. Surface stresses of the brake disc with different coating gradient forms (MPa).

Gradient Form	Exponential Function	Uniform Layer	Power Law ($n = 0.1$)	Power Law ($n = 0.5$)	Power Law ($n = 1$)	Power Law ($n = 1.5$)	cos	sin
Maximum stress during braking	235.8	346.6	214.6	244.3	274.5	302.7	275.3	243.6
Stress at the end of braking	203.1	325.8	166.5	194.5	226.8	245	227.2	193.6

In contrast, when the small-exponent power-law function ($n = 0.1$) is applied, the flexible buffer layer formed by the rapid decrease in elastic modulus effectively releases the interfacial contact stress. Consequently, its maximum stress during braking (214.6 MPa) and residual stress at the end of braking (166.5 MPa) drop to the lowest levels among all gradient models. However, this comes at the cost of diminished heat transfer performance. The reduction in high-thermal-conductivity pathways in the surface layer increases thermal resistance, pushing the maximum surface temperature to 210.7 °C, the highest among all gradient designs.

This quantitative comparison concludes that, in gradient coating design, extreme optimization of either the temperature field or the stress field alone leads to a significant deterioration in the other. In practical engineering applications, it is necessary to consider the specific failure tolerance of the braking system (i.e., prioritizing thermal fade resistance versus fatigue cracking resistance) to achieve a coordinated balance of thermo-mechanical performance across various gradient distribution forms.

4. Conclusions

This study establishes a thermo-mechanical coupled finite element model of an FGM gradient coating for high-heat friction components and systematically analyzes the influ-

ence of coating gradient distribution forms on contact behavior. The main conclusions are as follows:

- (1) Applying high-thermal-conductivity ceramic materials to the brake disc surface effectively reduces the disc surface temperature. However, material stiffness mismatch and abrupt changes in interfacial properties significantly aggravate the local thermal stress concentration.
- (2) The implementation of a gradient design greatly reduces the interfacial contact stress while maintaining the cooling effect. A coating thickness of 1 mm provides optimal comprehensive performance.
- (3) The physical properties of coating materials exhibit distinct sensitivity mechanisms to multi-field coupling. Thermal conductivity acts as the dominant factor governing temperature field evolution, while the elastic modulus gradient configuration is critical for regulating peak contact stress. Consequently, a specific design priority is proposed: optimizing the modulus gradient first to reduce stress, followed by the thermal conductivity gradient to control temperature.
- (4) Constrained by the physical coupling mechanism of material properties, simultaneously optimizing the disc surface temperature and stress field is highly challenging. Comprehensive quantitative comparisons indicate that gradient coatings featuring an exponential function, a small-exponent power-law function, or a sine function distribution achieve an excellent thermo-mechanical balance, successfully alleviating interfacial thermal stress while controlling the surface temperature rise. For instance, a power-law exponent in the range of $n = 0.5$ to 1.0 yields an optimal balance, while $n = 1.5$ can be considered an upper bound where cooling performance is prioritized, accepting a moderate increase in thermal stress compared to smaller exponents.

Author Contributions: Conceptualization, J.L.; Methodology, J.L.; Software, X.L., C.Z. and L.G.; Formal analysis, X.L., C.Z. and L.G.; Investigation, X.L. and J.L.; Resources, J.L.; Data curation, C.Z. All authors have read and agreed to the published version of the manuscript.

Funding: This research was funded by National Natural Science Foundation of China under Grant No. 12172147.

Data Availability Statement: The original contributions presented in this study are included in the article. Further inquiries can be directed to the corresponding author.

Conflicts of Interest: The authors declare no conflicts of interest.

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