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Comparative Thermo-Mechanical Analysis of Railway Wheel Treads Incorporating Variable Friction Coefficient Models under Long-duration Braking

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Abstract

Under long-duration braking conditions, particularly for freight trains on extended steep gradients, substantial heat accumulation can occur in wheel treads, resulting in complex thermo-mechanical responses and potential damage. A hybrid computational framework coupling MATLAB and ANSYS APDL is developed to iteratively update friction coefficients based on experimentally calibrated evolution laws. It enables the calculation of frictional power input, transient temperature fields, and thermal stresses while maintaining computational efficiency. Parametric analyses under varying gradient magnitudes and slope lengths are conducted to systematically compare constant and variable friction assumptions. The results show that incorporating time- and temperature-dependent friction introduces an intrinsic negative feedback mechanism into the thermo-mechanical system. It reduces peak tread temperature and thermal stress by up to 25–30% under severe braking conditions. Unlike the monotonic growth predicted with constant friction, the variable-friction model produces self-limiting "rise–fall" or "rise–stabilization" response patterns. It also promotes earlier attainment of dynamic thermal equilibrium. This study presents an advanced tribology-informed hybrid modeling framework for coupled thermo-mechanical analysis in sustained tread braking, enhancing physical fidelity while maintaining computational efficiency, with potential applicability to sliding contact systems under prolonged thermal exposure.

Keywords: freight wagon, tread braking, temperature-dependent friction, thermo-mechanical coupling, long downhill braking, hybrid numerical modelling

1 Introduction

In addition to supporting and guiding the vehicle, wheels also perform traction and braking functions, which introduce braking thermal loads. These braking thermal loads can induce various forms of damage to the wheels, seriously compromising their safety. This heat further transfers into the contacting components, raising the temperature of the contact pair. Due to the combined effects of severe thermal loads, residual thermal stresses, mechanical stresses, and phase transformations, the wheel tread may develop local hotspots, material delamination (shelling), spalling, thermally induced cracks from cyclic heating, and corrugated wear (corrugation). Therefore, accurately predicting friction-induced temperature rise during wheel tread braking is of critical research significance.

Numerical methods are widely employed in studying the thermo-mechanical coupling behavior of tread braking [1]. Finite element methods (FEM), as a commonly used numerical approach, are extensively applied to the analysis of rotational thermo-mechanical coupling between wheels and brake shoes. The friction coefficient, as a fundamental parameter characterizing interfacial behavior, serves as a key input in tribological analysis and thermo–mechanical coupling simulations of braking systems. Due to the lack of a thermo-mechanical coupled computational approach capable of capturing variable friction coefficients while maintaining both accuracy and efficiency, the friction coefficient has commonly been assumed to be constant. But under long-term operating conditions, continuous temperature rise at the contact interface may significantly alter the friction coefficient, thereby affecting the overall thermo-mechanical response of the system. Studies addressing this aspect remain limited.

This study aims to develop a numerical framework to investigate the influence of time- and temperature-dependent friction coefficients on the evolution of wheel tread temperature and thermal stress under ultra-long, steep-gradient conditions. The study further analyzes differences in tread thermo-mechanical responses under variable and constant friction coefficient conditions, as well as the sensitivity of temperature and thermal stress to gradient line parameters.

To achieve the aforementioned objectives, this study employs a hybrid numerical computation framework based on MATLAB and ANSYS APDL to compare the thermo-mechanical coupling responses under different friction coefficient assumptions across various gradient and slope-length conditions. Through bidirectional data transfer between MATLAB and ANSYS, the framework enables time-step iterative coupling calculations of temperature fields, frictional energy input, and thermal stress fields. Using experimental friction data for cast iron brake shoes and ER7 wheel steel, a dual-Sigma empirical model describing the friction coefficient as a function of time and temperature was fitted. Subsequently, a two-dimensional finite element numerical model based on a proportional reduction strategy was established, the influence of friction coefficient variation on tread temperature and thermal stress was systematically evaluated.

This study employs a wheel braking thermo-mechanical coupling modeling framework that balances computational accuracy and efficiency to analyze and compare the thermo-mechanical responses under time- and temperature-dependent friction coefficients versus constant friction coefficients, offering new numerical approaches and theoretical insights into the formation mechanisms of tread thermal damage.

2 Materials and methods

2.1 Material properties

In terms of material parameter selection, the wheel material from Ref. [2] is adopted to support the development of a friction coefficient fitting model that decays with time and temperature. To investigate the friction characteristics of the wheel tread and their temperature dependence, the experimentally measured data of friction coefficient, wheel rotation counts, and tread temperature from reference [2] were systematically fitted, as shown in Figure 1.

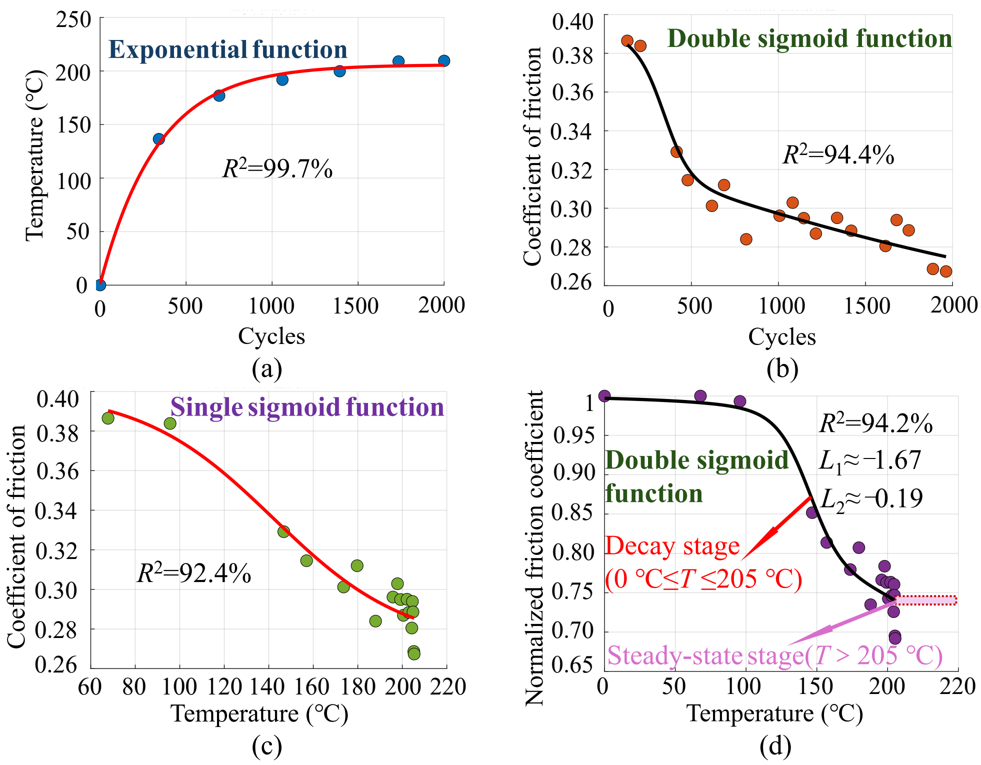

Figure 1: Fitting curves. (a) relationship between temperature and tread braking cycles. (b) relationship between friction coefficient and tread braking cycles. (c) relationship between friction coefficient and temperature. (d) relationship between normalized friction coefficient and temperature.

First, the relationship between temperature and rotation counts was modeled using an exponential fitting function. During the processing of friction coefficient data, due to experimental fluctuations, a differential method was employed to detect outliers: points where the change in friction coefficient between consecutive rotation counts exceeded a predefined threshold were considered outliers and removed. After outlier removal, the relationship between friction coefficient and rotation counts was fitted using a double Sigmoid function.

Reference [3] provides a fitting formula for the relationship between the friction coefficient and temperature of a disc brake and its friction pad as an example. As expressed in Equation (1), the variable friction coefficient $\mu(t, v, T)$ depends on two independent variables, v and T, and is governed by five parameters: $\mu_0, n_v, m_v, n_T, m_T$.

$$\mu(t,v,T) = \mu_0 \left(n_v e^{-m_v v(t)} + 1\right)\left(n_T e^{-m_T T(t)} + 1\right) \tag{1}$$

In the formula, μ_0 represents the steady-state friction coefficient, n_v denotes the speed-dependent scaling factor, and m_v is the exponential coefficient associated with friction velocity. Similarly, n_T accounts for the effect of temperature rise, while m_T serves as the exponential coefficient related to the temperature effect. The friction velocity refers to the relative sliding velocity between the rotating contact pairs (in m/s), and $T(t)$ denotes the temperature increment of the disc surface (in degrees Celsius).

Ref. [4] further established a temperature-dependent friction coefficient model using a normalized formulation. A normalization approach is also employed in this study, where the friction coefficient is divided by its maximum value, and the relationship between the normalized friction coefficient and temperature is fitted using a double Sigmoid function, ensuring that the normalized friction coefficient equals 1 at the initial temperature (T = 0). The normalized friction coefficient as a function of temperature and time is expressed by the fitting formula shown in Eq. (2). From the results reported in both Refs. [3] and [4], it can be observed that when the temperature exceeds 200 °C, the friction coefficient decreases with increasing temperature but gradually approaches a stable value. Therefore, for temperatures above 205 °C, the friction coefficient is assumed to remain stable. Accordingly, in this study, the decay factor of the friction coefficient for temperatures above 205 °C is close to 0.74. The wheel specimens were fabricated from modified ER7-type steel, while the brake shoe was made of cast iron. The corresponding material properties can be obtained from Refs. [2] and [5]. Therefore, for a constant friction coefficient [5] under constant-speed braking, Equation (2) applies. To include the effect of temperature variation, Equation (2) is further multiplied by Equation (3).

$$\mu^*(T) = 1 + \frac{L_1}{1+\exp\left[-k_4\left(T-c_4\right)\right]} + \frac{L_2}{1+\exp\left[-k_5\left(T-c_5\right)\right]} \tag{2}$$

$$\mu_{\mathrm{wb}}(V_{\mathrm{s}}) = 0.6\frac{V_{\mathrm{s}}+100}{5V_{\mathrm{s}}+100}\frac{16F_{\mathrm{nwb}}/\mathrm{g}+100}{80F_{\mathrm{nwb}}/\mathrm{g}+100} \tag{3}$$

where g denotes the gravitational acceleration (m/s^2), F_{nwb} is the normal force between the wheel and brake shoe (kN), V_{s} represents the relative velocity between the wheel and the brake shoe (km/h).

2.2 Numerical Thermo-Mechanical Coupling Method

The numerical computation framework of the model is schematically presented in Figure 2. This framework leverages the data exchange capability between MATLAB and the parametric ANSYS APDL program, combining MATLAB's rapid solution advantages for nonlinear problems with ANSYS APDL's efficiency in solving tread braking thermo-mechanical coupling equations. The framework is applicable for investigating the frictional thermo-mechanical coupling and thermal damage of wheel treads under ultra-long steep gradient conditions. While the present study focuses on constant-speed braking scenarios, the framework can also be extended to emergency braking and other operating conditions.

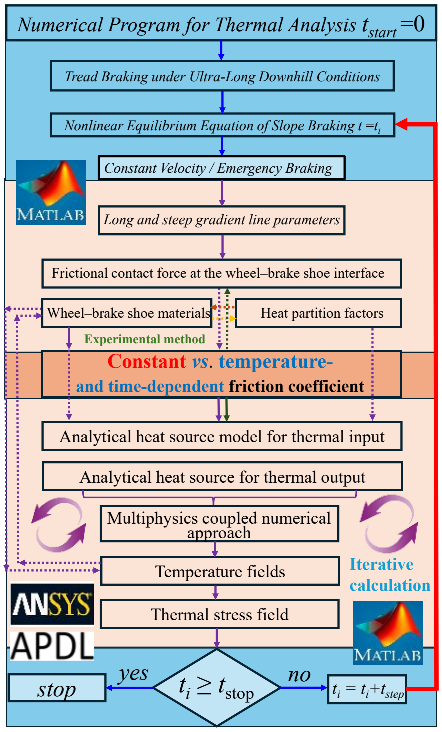

Figure 2: Fully parametric numerical analysis workflow for thermo-mechanical coupled closed-loop chain.

The numerical framework is flexible and convenient, allowing various thermal boundary conditions to be applied efficiently, and the detailed definition of the heat source boundary in this study is adopted from Ref. [6]. To improve computational efficiency, a thermo-mechanical indirect coupling method is employed: the temperature field is first computed and then used to determine the evolution of thermal stresses in the wheel tread. Additionally, as shown in Figure 3, a 3D-to-2D model reduction scheme combined with a 1/6 simplification was implemented to minimize computational cost while enabling rapid prediction of tread temperature rise.

In the 1/6 reduced thermo-mechanical model, the wheel domain was divided into refined and non-refined regions. To capture fine-scale temperature evolution on the wheel tread, mesh refinement was applied circumferentially and radially. In the refined region, mesh refinement was applied within a 10 mm radial extent, where the radial node spacing was set to 1 mm and the circumferential element spacing to 0.75°.

In the finite element numerical model, the PLANE77 element was employed to accurately resolve transient heat conduction and temperature distribution. During the subsequent thermoelastic stress analysis, this element was converted to the structural element type PLANE183. The developed thermo-mechanical indirect coupling model of the wheel contains 8800 elements and 26781 nodes, substantially decreasing the degrees of freedom relative to the 3D model and thereby improving computational efficiency.

Because a reduced 2D modeling approach was adopted, adiabatic boundary conditions were imposed on both ends of the 1/6 wheel model during thermal analysis to prevent nonphysical heat flux leakage, while displacement constraints were applied in the stress analysis to ensure a mechanically consistent response. A transient thermal analysis was conducted, employing the Newton–Raphson iterative method to solve the nonlinear system at each time step until convergence was reached. During the transient simulation of tread braking, a transient heat source was applied to the wheel tread to perform the thermo-mechanical analysis.

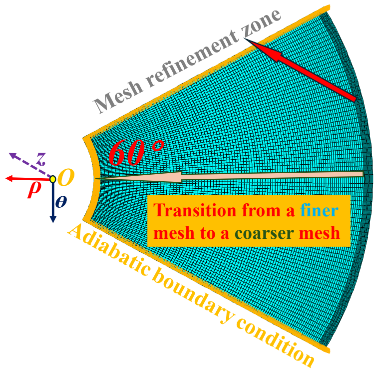

Figure 3: Two-dimensional parametric 1/6 partial wheel finite element numerical model.

To verify the convergence and stability of the finite element model, three wheel FE models with different element counts were analyzed: a fully refined global model, a 1/6 refined partial model, and the proposed reduced model. The fully refined and partial models employed tetrahedral elements with a radial and circumferential spacing of 0.1 mm. Under identical boundary and thermal load conditions, the maximum tread temperature difference between the global and partial models was only 0.003%, and that between the partial and reduced models was less than 0.0001%. These results demonstrate that the proposed FE model achieves excellent numerical convergence and stability.

3 Results

This study focuses on long-duration braking on ultra-long, steep-gradient slopes to examine, under relatively extreme conditions, the differences in predicted wheel tread temperature and thermal stress between dynamically varying and constant friction coefficients. Reference [7] reports that downhill ramps may reach gradients of up to 30‰ and lengths of 74 km. Based on this ultra-long ramp scenario, the constant-speed braking condition is set as follows: ambient temperature of 30 °C, ramp gradient of 12‰, axle load of 17 t, the train's constant braking speed of 45 km/h, braking distance of 30 km, and braking duration of 2000 s.

3.1 The effect of ramp gradient

This section investigates the temporal evolution of wheel tread temperature and thermal stress under different track gradient levels. When the friction coefficient is independent of time and temperature, and depends only on the train's braking speed and braking force, it can be treated as constant during constant-speed braking on long downhill ramps. Figure 4 presents the time evolution of wheel tread temperature and thermal stress under varying gradient conditions. The temperature–time profile of the wheel tread shows a monotonic increase, as the heat flux input to the tread significantly exceeds the heat dissipated to the environment via convection, indicating a typical energy accumulation-dominated thermal process.

Figure 5 summarises the maximum temperatures and thermal stresses attained at the conclusion under varying gradient conditions. When the friction coefficient is assumed constant, the maximum wheel tread temperature shows an approximately linear increase as the track gradient rises from 6‰ to 30‰. As the gradient increases, the gravitational potential energy to be dissipated per unit time rises significantly. As the gradient increases, the normal force on the brake shoe also increases. The frictional heating power increases linearly with normal and braking forces, thereby continuously raising the surface heat flux on the wheel. Since the braking speed is constant, the convective heat transfer coefficient is also set as constant. Meanwhile, the temperature-dependent increase in the specific heat capacity of the wheel material slows down the temperature rise, resulting in a diminishing slope of the temperature–time curve and suggesting that the system is gradually reaching thermal steady state.

After introducing the temperature-dependent friction coefficient, the overall temperature rise is significantly reduced. Compared with the constant friction coefficient model, the temperature rise rate exhibits a nonlinear trend: it first increases, then decreases, and rises again as the track gradient increases uniformly. Comparing cases with and without consideration of friction coefficient variations with time and temperature, significant differences in tread temperature characteristics are observed: at low gradients, the maximum temperatures are similar, whereas at high gradients, the differences are pronounced. This is primarily because, when the tread temperature exceeds the inflection point around 205 °C, the friction coefficient significantly decays according to empirical formulas (approaching 0.74 of its base value), reducing the frictional heating power and decreasing the heat input rate, thereby establishing a thermal negative feedback mechanism. This feedback mechanism mitigates thermal accumulation at elevated temperatures, particularly under steep gradients, thereby elucidating the nonlinear dependence of wheel tread temperature on track gradient.

As the track gradient increases, the unit braking work and surface heat flux rise, causing the wheel surface temperature to enter the friction coefficient decay range earlier. For the 18‰ gradient scenario, the peak temperature predicted by the temperature-dependent friction coefficient model is 52.5 °C lower than that of the constant friction coefficient model; for the 30‰ gradient, the difference increases to 113.8 °C. This indicates that the temperature-dependent friction model significantly reduces the thermal peak at high gradients. Therefore, the gradient not only determines the heat input intensity but also amplifies the nonlinear feedback effect of friction coefficient–temperature coupling. During simulations of braking under steep gradients on ultra-long downhill ramps, ignoring the temperature-induced decay of the friction coefficient may result in an overestimation of the maximum wheel surface temperature by about 25.4%.

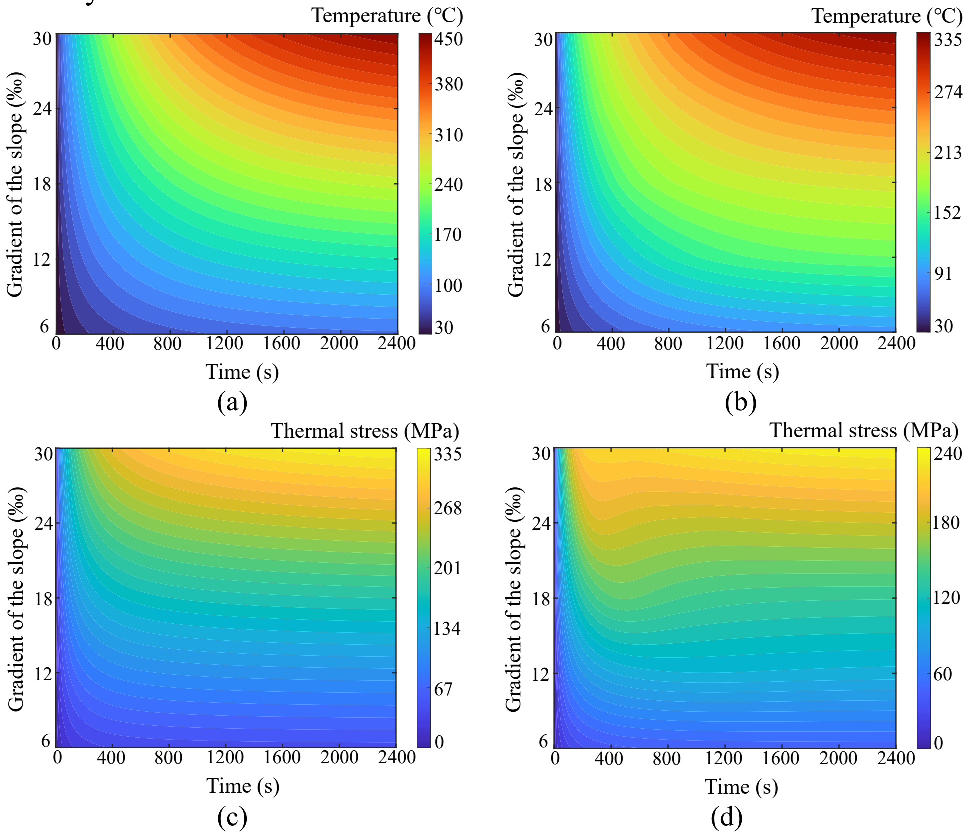

Figure 4: Effects of time- and temperature-dependent friction coefficient on the temporal evolution of temperature and thermal stress under different gradients. Subplots show: (a) Temperature response with constant friction coefficient (T–constant μ), (b) Temperature response with time- and temperature-dependent friction coefficient (T–$\mu(t,v,T)$), (c) Thermal stress response with constant friction coefficient (σ–constant μ), (d) Thermal stress response with time- and temperature-dependent friction coefficient (σ–$\mu(t,v,T)$).

Under constant-speed braking conditions, the temporal evolution of wheel tread thermal stress is closely related to the spatial distribution of the temperature field. When the friction coefficient is constant, thermal stress rises rapidly at the initial stage of braking and then gradually stabilizes, indicating that thermal stress is primarily driven by temperature gradients rather than solely by temperature magnitude. The peak von Mises stress increases from 49.2 MPa to 333.0 MPa, approximately 5.8

times, exhibiting a pronounced nonlinear amplification trend. This nonlinearity mainly arises from the superposition of temperature gradients: at high gradients, increased frictional heating accelerates surface temperature rise, while internal heat conduction lags, producing substantial surface-to-core thermal gradients and consequently inducing high thermal stress.

The thermal stress curves tend to stabilize during the later stages of braking, indicating that as heat conduction gradually homogenizes the internal temperature distribution of the wheel, the temperature gradients diminish, and the rate of thermal stress increase decreases significantly. Even with a continuous increase in tread temperature, the thermal stress approaches a quasi-equilibrium state as the temperature gradient between the surface and the core stabilizes, reflecting the strong dependence of stress on the internal temperature distribution.

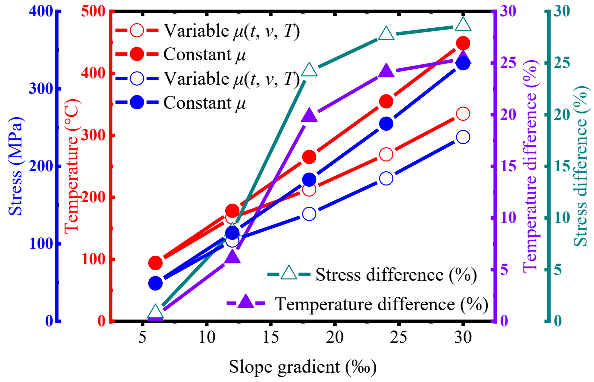

Figure 5: Sensitivity of temperature and thermal stress to gradient: constant vs. variable friction.

When a time- and temperature-dependent friction coefficient model is introduced, the evolution trend of thermal stress changes markedly. Relative to the constant friction coefficient model, the peak thermal stress is reduced, with the magnitude of reduction becoming increasingly significant at higher gradients. This indicates that under high thermal load conditions, the negative feedback effect of temperature on the friction coefficient effectively suppresses thermal stress levels.

It is noteworthy that when the gradient exceeds 12‰, the thermal stress–time curve is no longer monotonically increasing but exhibits a “rise-then-fall” pattern. During the initial stage of braking, the tread temperature rises rapidly while the friction coefficient remains relatively high, resulting in substantial frictional heating power and increasing thermal stress. When the surface temperature exceeds approximately 205 °C—the friction coefficient’s inflection point—the friction coefficient begins to decrease significantly (to 0.74 times its baseline value), causing the frictional heat generation rate to decline. Consequently, heat input slows, surface temperature gradients weaken, and thermal stress decreases and approaches equilibrium.

From a parametric perspective, increasing the gradient not only enhances frictional heat generation but also causes the tread temperature to enter the friction coefficient decay region earlier, thereby amplifying the relative reduction effect on thermal stress. For instance, under an 18‰ gradient, introducing a temperature-dependent friction coefficient reduces the peak von Mises stress by approximately 44.2 MPa (about 24.2%), whereas under a 30‰ gradient, the reduction increases to 95.2 MPa (about 28.6%). These results suggest that the temperature dependence of the friction coefficient not only affects the temperature rise but also indirectly influences the temporal and spatial distribution of thermal stress by modulating the dynamic evolution of temperature gradients. As the gradient increases, this thermo-mechanical feedback becomes more pronounced, manifested as delayed thermal stress peaks, flattened curves, and stress saturation during high-temperature stages.

The temperature decay characteristic of the friction coefficient results in a nonlinear, self-limiting evolution of wheel tread thermal stress: under low-gradient conditions, the limited temperature increase fails to trigger substantial frictional decay, leading to a nearly linear response of thermal stress; whereas under high-gradient conditions, once the temperature exceeds the critical range, the reduction in the friction coefficient suppresses heat input, causing the thermal stress peak to decrease significantly and stabilize earlier.

3.2 The effect of ramp length

This section This section investigates the effects of varying gradient lengths on wheel tread temperature and thermal stress. Figure 6 presents the time histories of wheel tread temperature and thermal stress under different ramp lengths. Since the braking speed is constant at 45 km/h, the corresponding braking durations differ for each gradient length: 2400 s for 30 km, 2800 s for 35 km, 3200 s for 40 km, 3600 s for 45 km, and 4000 s for 50 km. Figure 7 summarizes the maximum wheel tread temperature and thermal stress at the end of braking under different ramp lengths. For a constant friction coefficient, the wheel tread temperature gradually increases with braking distance, with the peak temperatures exhibiting a nonlinear rise that slowly approaches a plateau. When the friction coefficient varies with temperature and time, the temperature increase is more gradual, and the peak temperatures also rise nonlinearly with braking distance, but at a significantly reduced rate compared to the constant coefficient case. This behavior reflects the combined effects of heat input and dissipation gradually reaching a dynamic equilibrium as braking continues, while the temperature- and time-dependent decay of the friction coefficient suppresses frictional power input, leading to a smoother temperature evolution. Regarding thermal stress, under a constant friction coefficient, the tread stress progressively approaches a steady-state equilibrium as braking distance increases. When the friction coefficient is temperature- and time-dependent, the thermal stress exhibits a similar approach to equilibrium, following a temporal pattern of initial rise followed by a slight decline.

Overall, increasing the gradient length has a limited effect on tread temperature and thermal stress, while considering a temperature- and time-dependent friction coefficient significantly attenuates the increase in both temperature and thermal stress, allowing the system to reach a thermal–mechanical equilibrium more rapidly. The results further indicate that under long-duration constant-speed braking on steep gradients, the wheel tread temperature and thermal stress fields readily approach thermal equilibrium. When the braking speed is maintained at 45 km/h, the tread temperature gradually increases with gradient length from 30 km to 50 km and eventually stabilizes.

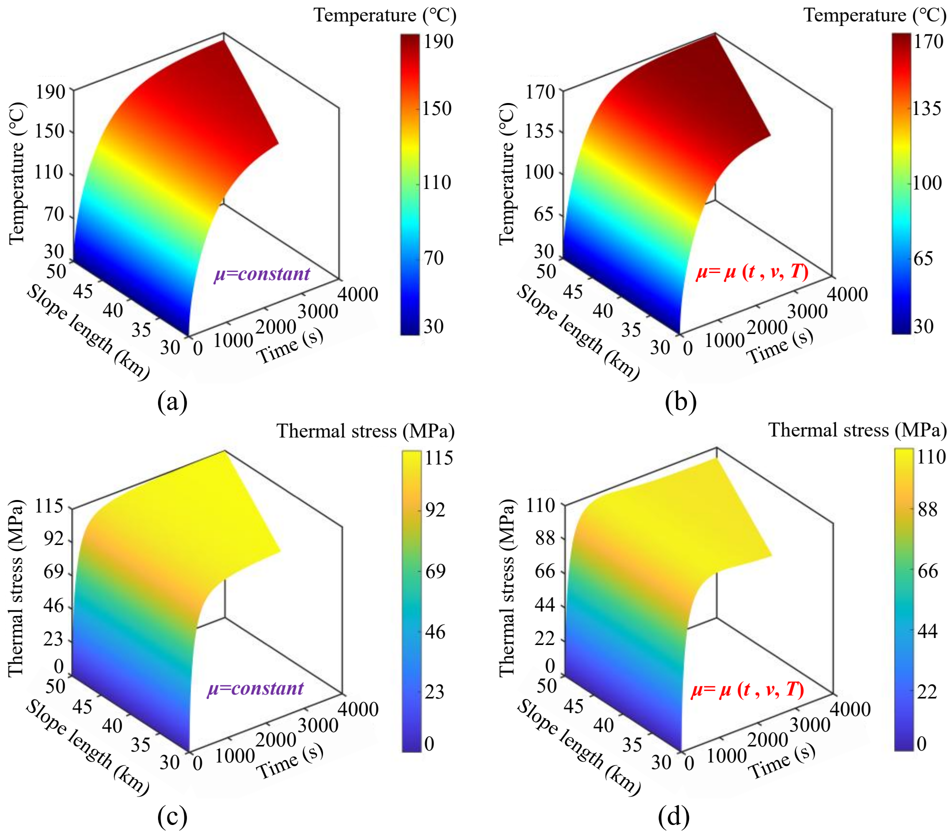

Figure 6: Effects of time- and temperature-dependent friction coefficient on the temporal evolution of temperature and thermal stress at various gradient lengths. Subplots show: (a) T–constant μ, (b) T–$\mu(t,v,T)$, (c) σ–constant μ, (d) σ–$\mu(t,v,T)$.

Comparing the constant friction coefficient model with the temperature- and time-dependent friction coefficient model at a braking speed of 45 km/h demonstrates that the latter substantially moderates the temperature rise. With increasing gradient length, the temperature difference between the two models gradually enlarges. The temperature- and time-dependent friction coefficient effectively reduces the thermal response of the wheel tread, yielding a more gradual temperature increase while accentuating the nonlinearity of the temperature–braking distance relationship. From the perspective of thermal stress evolution, when the friction coefficient is constant, the wheel tread thermal stress gradually increases with gradient length and rapidly escalates during the initial braking phase. For braking distances ranging from 30 km to 50 km, the maximum tread thermal stress rises marginally from 114.2 MPa to 115.0 MPa, an increase of only 0.69%, indicating that the system rapidly reaches thermal equilibrium in the early stage. When the friction coefficient is temperature- and time-dependent, the thermal stress initially rises, then decreases, and finally stabilizes, with the steady-state value slightly decreasing from 104.3 MPa to 103.6 MPa, a reduction of 0.67%. Comparison of the two sets of results suggests that, under long-duration constant-speed braking, the growth rate of thermal stress is significantly lower than in short-term braking, indicating that thermal conduction and convective dissipation progressively counteract thermal stress accumulation.

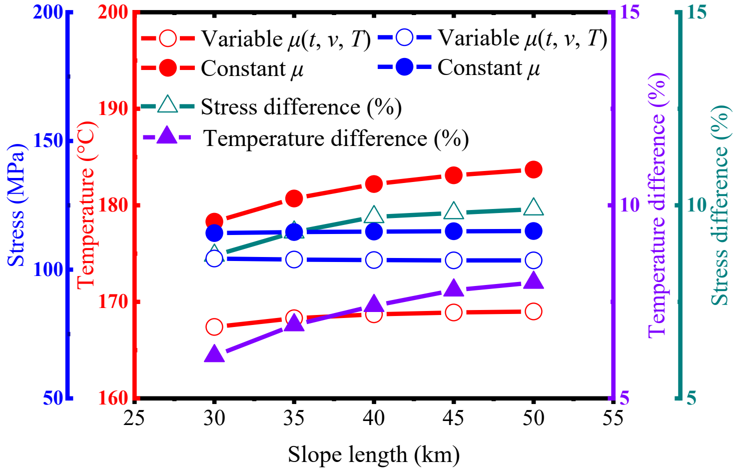

Figure 7: Sensitivity of temperature and thermal stress to gradient length: constant vs. variable friction.

During long-gradient constant-speed braking, the wheel tread temperature and thermal stress eventually stabilize, demonstrating a dynamic energy balance between conductive, frictional, and convective heat processes. In the initial braking stage, the frictional heat power is much greater than the convective heat dissipation, leading to a rapid increase in temperature. As time progresses, the rising tread temperature induces thermal degradation of the friction coefficient, while the increasing temperature difference enhances convective cooling. At this point, the system enters a quasi-steady thermal equilibrium, and the rates of change of temperature and thermal stress approach zero. This indicates that the influence of gradient length on temperature and thermal stress is primarily concentrated in the early braking stage. When the braking duration exceeds a critical value, the wheel's thermal response is primarily governed by conduction and convection, exhibiting characteristic thermal saturation behavior.

Under long-gradient constant-speed braking, both wheel tread temperature and thermal stress exhibit clear equilibrium characteristics. As braking distance increases, the system gradually stabilizes, and further extension of braking distance exerts minimal effect on thermal loading. For gradients exceeding 30 km, the tread temperature and thermal stress stabilize around 169 °C and 104 MPa, with fluctuations below 1%. This trend has significant engineering implications for the thermal safety assessment of railway vehicles under long downhill constant-speed braking: neglecting the temperature dependence of the friction coefficient may lead to slight overestimation of thermal stress while exerting limited influence on the overall steady-state trend. Therefore, under long-duration constant-speed braking conditions, priority should be given to controlling the early-stage temperature rise and protecting against material thermal fatigue rather than on steady-state accumulation resulting from extended braking duration.

4 Conclusions and Contributions

In this study, a coupled thermo-mechanical simulation framework based on ANSYS APDL and MATLAB is developed. A two-dimensional thermo-mechanically coupled finite element model is established to capture the temperature- and time-dependent decay characteristics of the friction coefficient. The effects of dynamically evolving friction coefficients on wheel tread temperature and thermal stress fields under key gradient parameters are systematically analyzed. The main findings are summarized as follows:

Incorporating a time- and temperature-dependent friction coefficient mitigates thermal accumulation and reduces peak wheel tread temperature and thermal stress, whereas neglecting this effect can overestimate thermal loads by up to 25–30% under long-downhill braking conditions.

Under long-duration constant-speed braking on long downhill ramps, wheel tread temperature and thermal stress gradually stabilize, with the system reaching a dynamic thermal–mechanical equilibrium. Considering a time- and temperature-dependent friction coefficient significantly moderates the early-stage temperature rise and thermal stress, while further extension of braking distance or gradient length has minimal effect on the steady-state thermal response.

Incorporating temperature- and time-dependent friction fundamentally alters the evolution of wheel tread temperature and thermal stress, giving rise to self-limiting "rise–fall" or "rise–fall–stabilization" response modes rather than monotonic growth assumed under constant friction conditions.

Although this study analyzes the effects of temperature- and time-dependent friction coefficient decay under varying track gradient parameters on wheel tread temperature and thermal stress, and provides a series of qualitative and quantitative findings, certain limitations remain and should be addressed in future research.

To isolate the effect of temperature-dependent friction on wheel tread temperature and thermal stress, the train speed was assumed constant. In reality, friction decay would reduce braking force and may slightly increase train speed, which could affect absolute temperature and stress values. Nevertheless, this controlled approach allows a clear comparison between constant and temperature-dependent friction models.

In this study, the friction coefficient is assumed to be fixed at a constant value for temperatures above 205 °C. In practice, due to its time-dependent decay characteristics, the friction coefficient may fall below this fixed value. However, according to Ref. [3,4], the friction coefficient tends to approach a stable value with increasing temperature, and the deviation below this fixed value is relatively small. This implies that the calculated difference between the two cases in this study is conservative, and the actual difference may be slightly larger.

As noted in Ref. [4], numerous factors influence the actual braking response process. In thermal analysis, parameters such as the heat partition coefficient and convective heat transfer coefficient also introduce additional uncertainties into the model. Therefore, conducting experimental validation of the friction coefficient considered in this study is challenging, and further experimental work will be carried out in future research.

Future research will focus on extending the present framework to more realistic operating conditions and improving model fidelity. In particular, variable train speed and braking dynamics will be incorporated to better reflect actual braking processes. The friction coefficient model will be further refined, especially at high temperatures, and validated through dedicated experiments. The findings provide valuable engineering guidance for enhancing numerical modeling tools used in thermo-mechanical simulations for the design and assessment of railway braking systems. Ultimately, the study supports the broader societal goals of transportation safety, infrastructure resilience, and environmental sustainability through improved engineering design and risk-informed decision-making.

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