

Article

Interplay Between Wear and Thermal Expansion in 6082 Aluminum: A Simulation and Experimental Study

Yang Tian ¹, Mohamed Kalifa ¹, Muhammad Khan ^{1,*} and Yifan Yang ²¹ Centre of Life-Cycle Engineering and Management, Cranfield University, Milton Keynes MK43 0AL, UK; yang.tian.593@cranfield.ac.uk (Y.T.); mohamed.kalifa@cranfield.ac.uk (M.K.)² Centre for Autonomous and Cyberphysical Systems, Cranfield University, Milton Keynes MK43 0AL, UK; yifan.yang.380@cranfield.ac.uk

* Correspondence: muhammad.a.khan@cranfield.ac.uk

Abstract: This study investigates how wear and thermal expansion interact in 6082 aluminum, utilizing a pin-on-disc tribometer and finite element analysis under diverse mechanical conditions. The findings show that thermal expansion reduces contact area by forming a protrusion at the contact interface. This interaction between wear and thermal expansion causes dynamic shifts in the contact region and pressure distribution, affecting the disc center and altering wear progression and temperature patterns. High thermal expansion shifts maximum wear from the contact center to outer regions, especially at higher speeds and loads. Without thermal expansion, wear-only conditions overestimate friction dissipation, resulting in a higher peak temperature. These results highlight the critical role of thermal expansion in shaping wear patterns and contact behavior in sliding applications. This research offers insights for optimizing tribological performance in 6082 aluminum, with potential applications in other materials.

Keywords: wear; finite element analysis; pin-on-disc tribometer; thermal expansion



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1. Introduction

Materials engineering is pivotal in developing durable and efficient components across various industries, including aerospace and automotive engineering [1–3]. Among the numerous challenges in these fields, wear significantly reduces the lifespan of components and compromises their performance.

Wear, defined as the gradual removal of material due to mechanical action [4], and thermal expansion, the change in material dimensions in response to temperature variations [5], are critical phenomena affecting material performance. The interplay between wear and thermal expansion leads to complex changes in contact behavior at frictional interfaces, impacting functionality under operational conditions.

When frictional pairs slide against each other, friction generates heat at the contact interface, causing the material's temperature to rise [6,7]. This temperature increase leads to thermal expansion and material softening, altering the contact behavior [8]. As sliding continues, wear at the contact area changes the contact characteristics, such as contact area, contact pressure, and friction work, subsequently affecting the heat generation at the contact interface [9]. The interplay between wear and thermal effects influences both the wear evolution and temperature distribution of the material.

Previous research has explored these interactions. For instance, a simulation study demonstrated that thermal expansion and relative motion jointly affect the wear distribution of friction blocks, causing concave and eccentric wear patterns [10]. Another study used a coupled simulation method to model wear caused by thermal expansions in contact areas, considering factors such as friction, thermal expansion, wear, and temperature-dependent material properties [11]. Additionally, computer modeling of surfaces with microasperities revealed wear and friction regimes in model ceramic materials, dependent on normal

stress levels and sliding velocity [12–16]. Materials with a negative thermal expansion coefficient were found to achieve near-zero wear rates and maintain nano-rough surfaces under specific stress levels, presenting practical applications [17]. A finite element analysis of a disc-pad assembly highlighted that thermal effects influence friction levels, thereby impacting wear rates. This study showed that large deformation and high contact pressure due to thermal stress accelerate wear, emphasizing its importance in brake design and analysis [18].

The interplay between wear and thermal expansion is a complex phenomenon with significant implications for material performance and practical applications. However, to the best of our knowledge, no research has specifically investigated the influence of wearing and thermal expansion on each other. This paper aims to fill this gap by examining how the interplay between wear and thermal expansion influences wear evolution and temperature distribution using a finite element model and a pin-on-disc tribometer, focusing on 6082 aluminum under varying thermal and mechanical conditions. 6082 aluminum, known for its medium strength and good corrosion resistance, is widely used in various applications such as automotive components, aerospace components, and home appliances [19,20]. Therefore, 6082 aluminum was selected for this study.

The research objectives are as follows: Examining how thermal expansion affects the wear characteristics of 6082 aluminum under varying operating conditions; determining the impact of thermal expansion on contact behavior, such as contact area, contact pressure, and friction dissipation rate; investigating how the wear process alters the temperature distribution and thermal expansion of 6082 aluminum during sliding motion; understanding the reciprocal relationship between wear and thermal effects, including how wear-induced changes influence the heat source at the contact interface; exploring the combined impact of wear and thermal expansion on the performance of 6082 aluminum under operational conditions. By addressing these objectives, this study aims to enhance the understanding of the interplay between wear and thermal expansion during friction sliding.

The paper is structured as follows: Section 2 describes the methodology, detailing the simulation setup using COMSOL Multiphysics to analyze wear in a 6082 aluminum disc and a 440C steel pin; Section 3 presents the findings on contact area, wear rate, wear evolution, wear scar profiles, interface temperature, and thermal expansion. These results demonstrate the significant impact of thermal effects on wear behavior and temperature distribution. This section also analyzes the dynamic changes in frictional dissipation and temperature due to wear and thermal expansion and how thermal expansion influences the evolution of wear; Section 4 concludes with key insights on the interplay between wear and thermal expansion, discusses the limitations of the research, and suggests future research directions to explore other materials and lubrication effects.

2. Materials and Methods

2.1. Simulation Setup

The wear analysis was conducted using COMSOL Multiphysics software, version 6.2.0.339. The framework of the simulation is presented in Figure 1. The study focused on evaluating the wear characteristics of a disc made from 6082 aluminum and a steel pin made from 440C steel. The material properties used in this mode are shown in Table 1. The Young modulus of both the pin and disc is temperature-dependent, and their thermal expansion properties are also accounted for to accurately model the wear process.

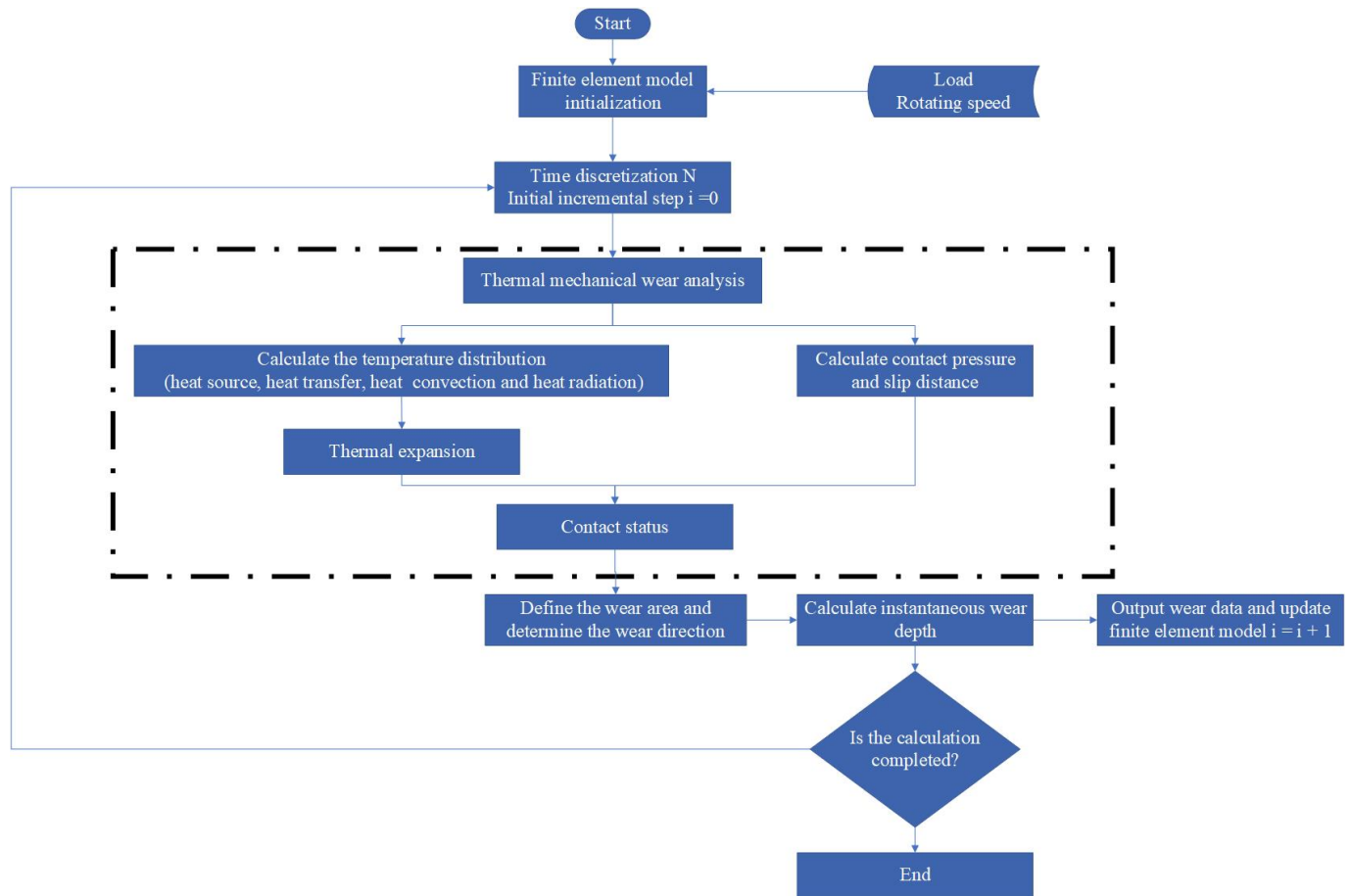


Figure 1. Framework of the simulation.

Table 1. Material properties.

Symbol	Property	Value	Unit
Disc			
ρ_1	Density	2700	kg/m ³
$E_1(T)$	Young's modulus	$E_1(T) = E_{10} \left(1 - \frac{T}{T_m} \exp \left(\theta_g \left(1 - \frac{T}{T} \right) \right) \right)$ [21]	Pa
E_{10}	Young's modulus at 0 K	7.2×10^9	Pa
T	Temperature	Input from solid heat transfer model	K
T_m	Melting temperature	828.15	K
θ_g	Material constant	2.295	K
ν_1	Poisson's ratio	0.33	–
C_{p1}	Heat capacity	897	J/(kg·K)
k_1	Thermal conductivity	195	W/(m·K)
α	Thermal expansion coefficient	2.3×10^{-5}	1/K
σ_{ys0}	Initial yield stress	2.71×10^8	Pa
H	Hardness	90	MPa
Pin			
ρ_2	Density	7750	kg/m ³
$E_2(T)$	Young's modulus	$E(T)$	Pa
ν_2	Poisson's ratio	0.3	–
k_2	Thermal conductivity	24.2	W/(m·K)
C_{p2}	Heat capacity	430	J/(kg·K)
d_L	Thermal strain	$d_L = \frac{d_L(T) - d_L(\text{Tempref})}{1 + d_L(\text{Tempref})}$	–

2.1.1. Geometry and Mesh

The simulation geometry included a sphere pin and a disc surface, as per Figure 2. A pin with a diameter of 6 mm slid against a disc at a point 20 mm away from the disc's center; the disc had a radius of 50 mm and a thickness of 5 mm. The mesh was created using a tetrahedral configuration. In order to obtain accurate outcomes and optimize computational efficiency, the contact region between the disc and pin was locally refined.

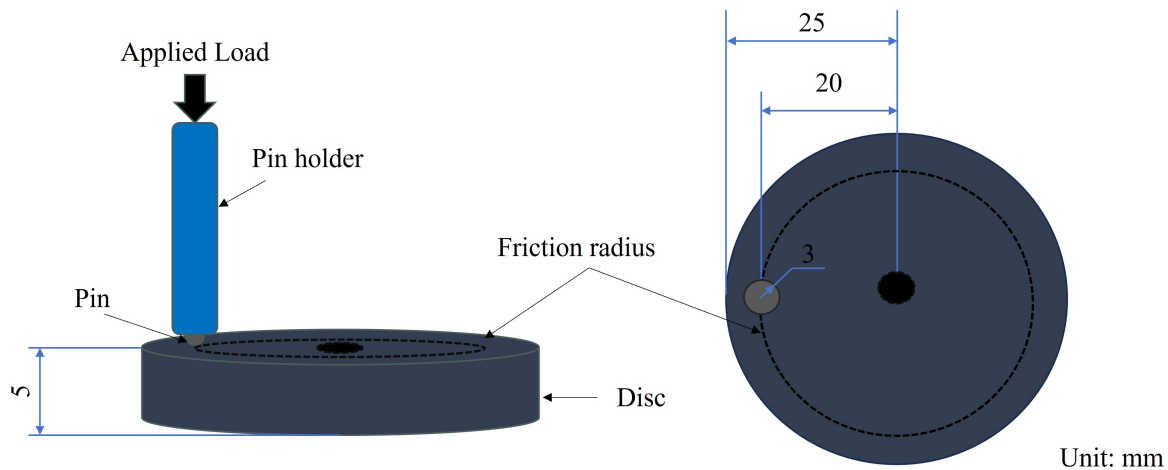


Figure 2. Geometry of the model.

2.1.2. Boundary Conditions

The boundary conditions were set as in Figure 3 to replicate real-world contact scenarios. The pin was subjected to a downward force simulating the applied load (5 N, 10 N, and 15 N) during the wear test. Additionally, the pin's movement is limited to the vertical direction, with no horizontal plane motion allowed. The bottom of the disc was fixed with different rotating speeds (100 RPM, 200 RPM, and 300 RPM). The ambient temperature was set at 293.15 K to maintain consistency in thermal conditions.

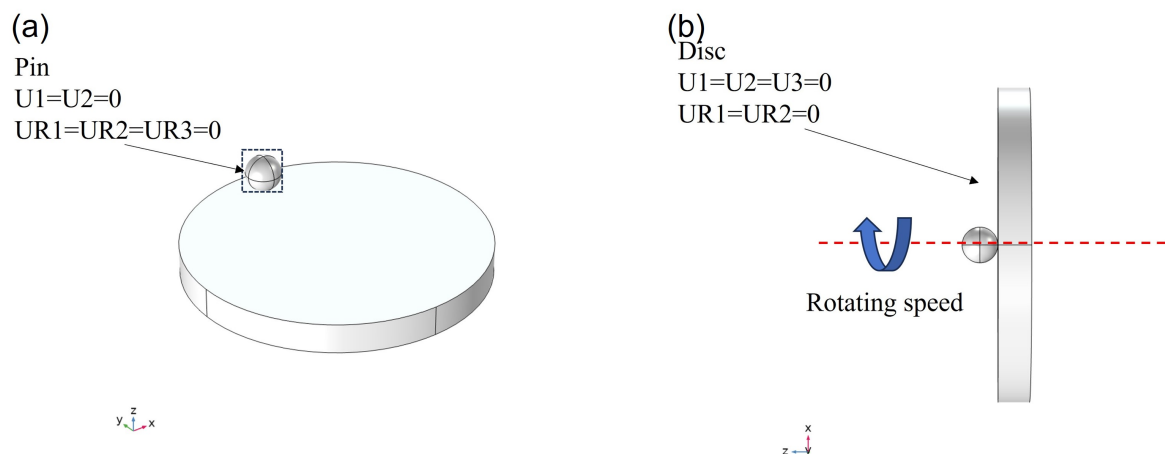


Figure 3. (a) Boundary conditions for the pin. (b) Boundary conditions for the disc.

2.2. Solid Mechanical Model

Contact Definition

The contact behavior between the pin and disc was described in terms of the augmented Lagrangian method because it provides more accurate results compared to the penalty method [22]. The traditional Archard model ignored the effect of contact between

two rough surfaces, resulting in an inaccurate wear rate [23]. Here, we used the wear model proposed by Sarkar, which introduces the coefficient of friction into the Archard model to consider the influence of contact on the wear process, given by Equation (1) [24].

$$w = k \left(\frac{F_n V \sqrt{1 + 3\mu^2}}{H} \right) \quad (1)$$

This equation describes that the wear rate w depends on wear coefficient k calculated from experimental measurement (0.0001), the normal force F_n , the sliding velocity V , the coefficient of friction μ —which varies over time and is averaged to 0.66 to enhance model convergence—and softer material hardness H .

2.3. Thermal Model

The contact behavior in the solid mechanism model was exported to the heat transfer model to determine the heat source using frictional dissipation, in which heat transfer, heat convection, and heat radiation were considered to ensure the accuracy of the simulation. The thermal partition coefficient between the ball and disc was determined using the Charron relationship built into COMSOL Multiphysics.

2.3.1. Thermal Contact Definition

The contact behavior in the solid mechanism model was exported to the heat transfer model to determine the heat source using frictional dissipation. The thermal contact was defined by Equations (2)–(7).

$$-\mathbf{n}_{\text{dst}} \cdot \mathbf{q}_{\text{dst}} = -h(T_{\text{sct}} - T_{\text{dst}}) + rQ_b \quad (2)$$

Equation (2) represents the heat flux balance at the destination surface (dst); for the disc, $\mathbf{n}_{\text{dst}}$ is the unit normal vector at the destination surface, $\mathbf{q}_{\text{dst}}$ is the heat flux vector at the destination surface, h is the heat transfer coefficient, T_{sct} and T_{dst} are the temperatures at the scatter (source) and destination surfaces, respectively, r is a reflection coefficient, and Q_b is the heat source term. The left side of the equation represents the heat flux leaving the destination surface. The right side combines the convective heat transfer due to the temperature difference between the source and destination surfaces and a portion of the heat source term, which is adjusted by the reflection coefficient r .

$$-\mathbf{n}_{\text{src}} \cdot \mathbf{q}_{\text{src}} = -h(T_{\text{sct}} - T_{\text{src}}) + (1 - r)Q_b \quad (3)$$

Equation (3) represents the heat flux balance at the source surface (src); for the pin, $\mathbf{n}_{\text{src}}$ is the unit normal vector at the source surface, $\mathbf{q}_{\text{src}}$ is the heat flux vector at the source surface, and T_{src} is the temperature at the source surface.

Similar to the first equation, the left side represents the heat flux leaving the source surface. The right side includes the convective heat transfer due to the temperature difference between the source and destination surfaces and the remaining portion of the heat source term (adjusted by $1 - r$).

$$r = \frac{1}{1 + \xi}, \quad \xi = \sqrt{\frac{\rho_{\text{src}} c_{p,\text{cr}} (k_{\text{src}} \mathbf{n}_{\text{src}}) \cdot \mathbf{n}_{\text{src}}}{\rho_{\text{dst}} c_{p,\text{dst}} (k_{\text{dst}} \mathbf{n}_{\text{dst}}) \cdot \mathbf{n}_{\text{dst}}}} \quad (4)$$

Equation (4) defines the reflection coefficient r based on the relative thermal properties of the source and destination materials. r is the reflection coefficient. The parameter ξ involves the densities, specific heat capacities, and thermal conductivities of the two surfaces; ξ is a dimensionless parameter that depends on the properties of the source and destination materials, ρ is the density, c_p is the specific heat capacity, and k is the thermal conductivity.

$$h_c = 1.25k_{\text{contact}} \frac{m_{\text{assp}}}{\sigma_{\text{asst}}} \left(\frac{p}{H_c} \right)^{0.95} \quad (5)$$

Equation (5) estimates the contact heat transfer coefficient based on the contact mechanics and material properties at the interface. h_c is the contact heat transfer coefficient, k_{contact} is the effective thermal conductivity at the contact interface, m_{assp} is the mean asperity slope (0.6), σ_{asst} is the standard deviation of the asperity heights (3.181 μm), p is the contact pressure, and H_c is the hardness of the softer material.

$$\frac{2}{k_{\text{contact}}} = \frac{1}{(k_{\text{src}} \mathbf{n}_{\text{src}}) \cdot \mathbf{n}_{\text{src}}} + \frac{1}{(k_{\text{dst}} \mathbf{n}_{\text{dst}}) \cdot \mathbf{n}_{\text{dst}}} \quad (6)$$

Equation (6) calculates the effective thermal conductivity k_{contact} at the contact interface based on the thermal conductivities of the source and destination materials.

$$h = h_c + h_g + h_r \quad (7)$$

Equation (7) sums the contributions of different mechanisms (contact conductance, gap conductance, and radiation) to the overall heat transfer coefficient. h is the total heat transfer coefficient, h_c is the contact heat transfer coefficient, h_g is the heat transfer coefficient due to the gap conductance, and h_r is the radiative heat transfer coefficient. We assume that there is no gap and no radiative heat transfer at the contact between the disc and pin, so h_g and h_r are ignored, both equaling 0.

2.3.2. Surface-to-Environment Radiation

This equation represents the radiative heat flux between a surface and its ambient environment.

$$\mathbf{n} \cdot \mathbf{q} = \varepsilon \sigma (T_{\text{amb}}^4 - T^4) \quad (8)$$

$\mathbf{n} \cdot \mathbf{q}$ is the radiative heat flux vector in the direction of the normal $\mathbf{n}$, and the ε of the surface is a dimensionless coefficient that measures the efficiency of the surface in emitting thermal radiation, ranging from 0 to 1. For the disc, the emissivity is 0.165, and for the pin, it is 0.13 [25,26]. σ is the Stefan-Boltzmann constant, which is approximately $5.67 \times 10^{-8} \text{ W m}^{-2} \text{ K}^{-4}$, T_{amb} is the ambient temperature in Kelvin, and T is the surface temperature in Kelvin. The term $\varepsilon \sigma (T_{\text{amb}}^4 - T^4)$ represents the net radiative heat exchange between the surface and the surrounding environment. This equation accounts for the fact that the heat radiated by a surface depends on the fourth power of its absolute temperature, as well as the fourth power of the ambient temperature.

2.3.3. Heat Convection

Since air flows over the disc surface, the forced thermal convection from the disc surface to an external airflow can be described by Equation (9):

$$\begin{aligned} -\mathbf{n} \cdot \mathbf{q} &= q_0 \\ q_0 &= h(T_{\text{ext}} - T) \\ h &= \begin{cases} 2 \frac{k}{L} \frac{0.3387 \text{Pr}^{1/3} \text{Re}_L^{1/2}}{\left(1 + \left(\frac{0.0468}{\text{Pr}}\right)^{2/3}\right)^{1/4}} & \text{if } \text{Re}_L \leq 5 \cdot 10^5 \\ 2 \frac{k}{L} \text{Pr}^{1/3} \left(0.037 \text{Re}_L^{4/5} - 871\right) & \text{if } \text{Re}_L > 5 \cdot 10^5 \end{cases} \end{aligned} \quad (9)$$

$-\mathbf{n} \cdot \mathbf{q} = q_0$ represents the heat flux normal to the surface, q_0 is the convective heat flux, $q_0 = h(T_{\text{ext}} - T)$ indicates that the convective heat flux is proportional to the temperature difference between the external fluid (T_{ext}) and the surface (T). The convective heat transfer coefficient h is determined based on the Reynolds number (Re_L) and the Prandtl number (Pr): k is the thermal conductivity of the fluid, L is the characteristic length, Pr is the

Prandtl number—a dimensionless number that characterizes the relative thickness of the velocity and thermal boundary layers— Re_L is the Reynolds number, a dimensionless number that characterizes the flow regime (laminar or turbulent) based on the fluid velocity, characteristic length, and kinematic viscosity. These equations provide the framework for calculating the convective heat transfer coefficient h based on the flow conditions and fluid properties.

Since the pin is stationary, the h under natural convection is given by

$$h = \frac{k}{D} \left(2 + \frac{0.589 Ra_D^{1/4}}{\left(1 + \left(\frac{0.469}{Pr} \right)^{9/16} \right)^{4/9}} \right) \quad (10)$$

This equation describes the convective heat transfer coefficient h for natural convection around a spherical object, where h is the convective heat transfer coefficient, k is the thermal conductivity of the fluid, and D is the diameter of the pin, Ra_D is the Rayleigh number, a dimensionless number that characterizes the natural convection flow. Pr is the Prandtl number, a dimensionless number that represents the ratio of momentum diffusivity (kinematic viscosity) to thermal diffusivity.

2.4. Experimental Setup

To validate the accuracy of the finite element method (FEM) model, pin-on-disc experiments were conducted at varying sliding speeds (100–300 RPM) and load conditions (5–15 N). These experiments were performed using an Anton Paar TRB3 tribometer (Anton Paar, Graz, Austria) without using any lubricants. Figure 4a illustrates the experimental setup. A profilometer was used to assess the wear level, as depicted in Figure 4b. The maximum temperature at the contact area was measured using an infrared thermometer with an accuracy of ± 2 °C. Temperature measurements were taken under a laboratory temperature range of 20–23 °C.

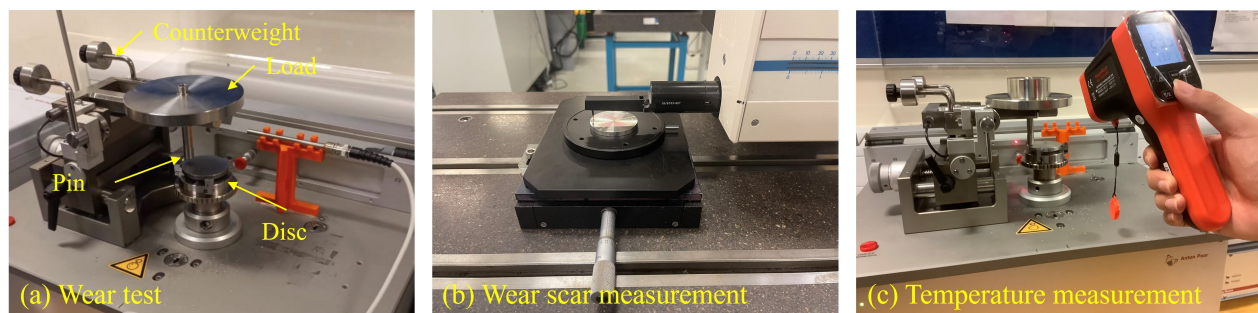


Figure 4. (a) Wear experiment setup. (b) Wear scar profile measurement. (c) Temperature measurement using a thermal gun.

3. Results and Discussion

3.1. Contact Area

The contact area between friction pairs changes due to wear and temperature fluctuations. To assess the impact of thermal expansion and material softening, Figure 5 shows contact area variations with and without thermal effects. Significant differences in contact area are evident between scenarios with and without thermal effects. Thermal expansion is more pronounced at higher speeds and loads.

Without thermal effects, the contact area between friction pairs gradually increases due to wear between the disc and pin [27]. As wear progresses, the pin penetrates further into the disc.

In contrast, with thermal effects considered, the contact area is smaller and exhibits significant fluctuations. This results from dynamic contact behavior due to the combined

effects of wear and thermal expansion. Under 15 N and 300 RPM, as the material absorbs frictional heat, thermal expansion becomes apparent (after 0.5 min), causing the interface to become more convex and reducing the actual contact area, as per Figure 6. In [10], reduced contact area is also observed and attributed to protrusions created by expansion, where their simulation used a flat pin surface sliding against a disc.

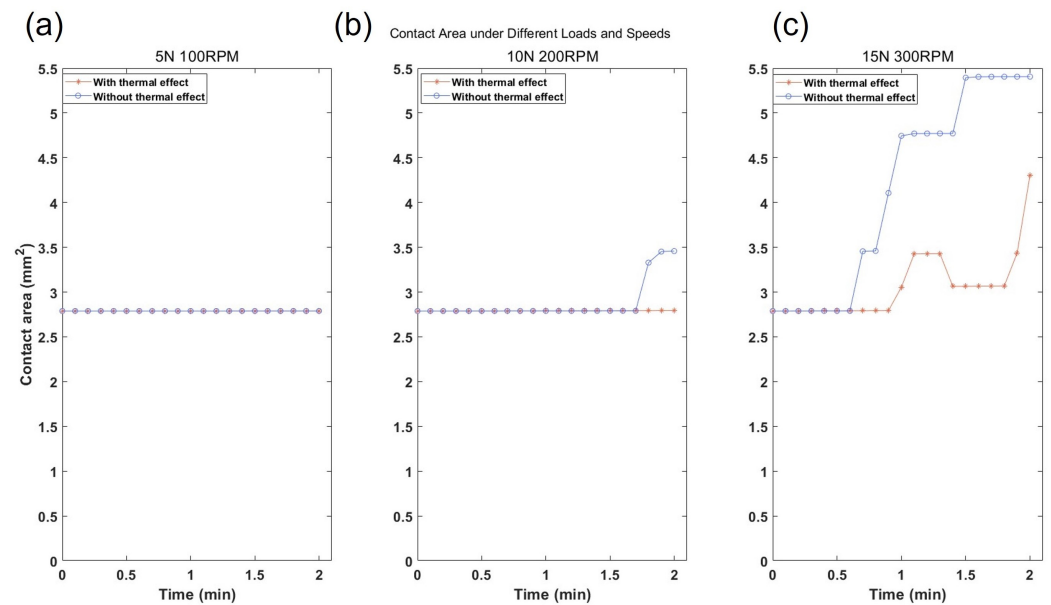


Figure 5. Comparison of contact area with and without thermal effects at (a) 5 N and 100 RPM, (b) 10 N and 200 RPM, and (c) 15 N and 300 RPM (the results are from the FEM simulation).

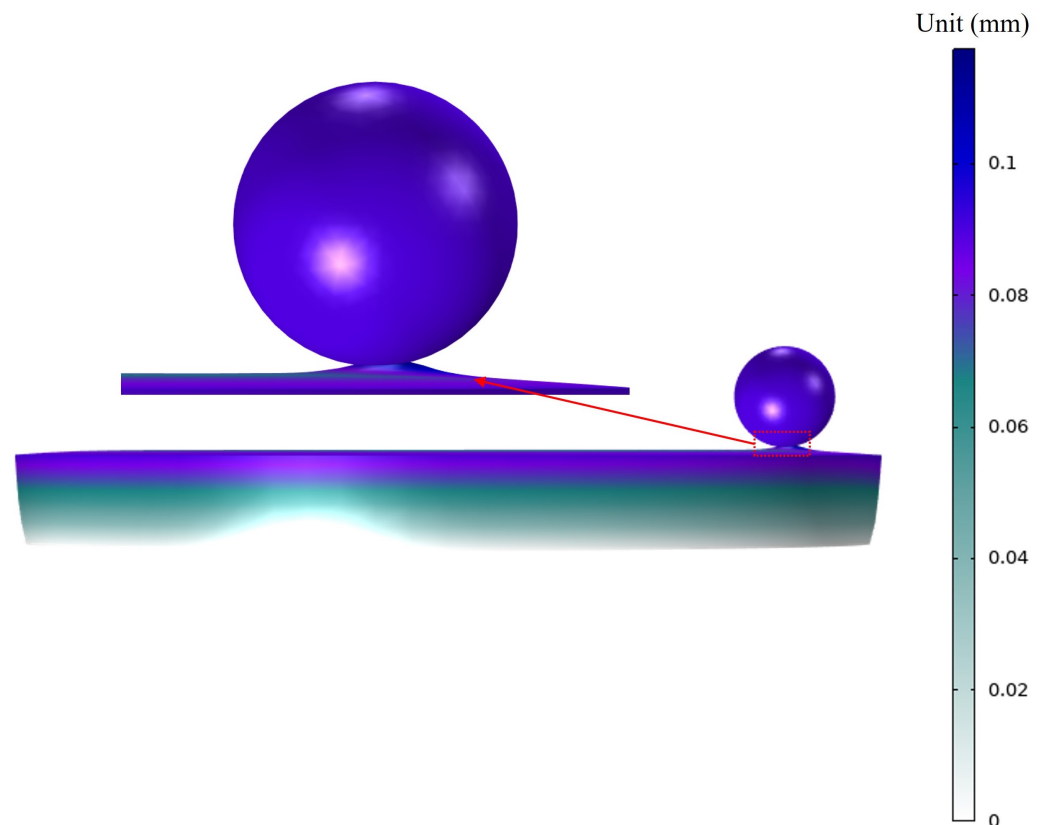


Figure 6. Protrusion at contact interface under 15 N and 300 RPM (the result is from the FEM simulation).

This analysis shows that thermal expansion directly affects the contact area between the pin and disc, influencing other wear characteristics. Therefore, studies on the tribological behavior of the pin should consider the effects of thermal expansion.

3.2. Effective Wear Rate

Figure 7 illustrates the effective wear rate under varying operating parameters. Generally, the effective wear rate increases with higher load and rotational speed. Simulations with thermal effects show greater fluctuations in the wear rate due to dynamic contact changes. In contrast, simulations without thermal effects show a more stable wear rate due to consistent contact behavior.

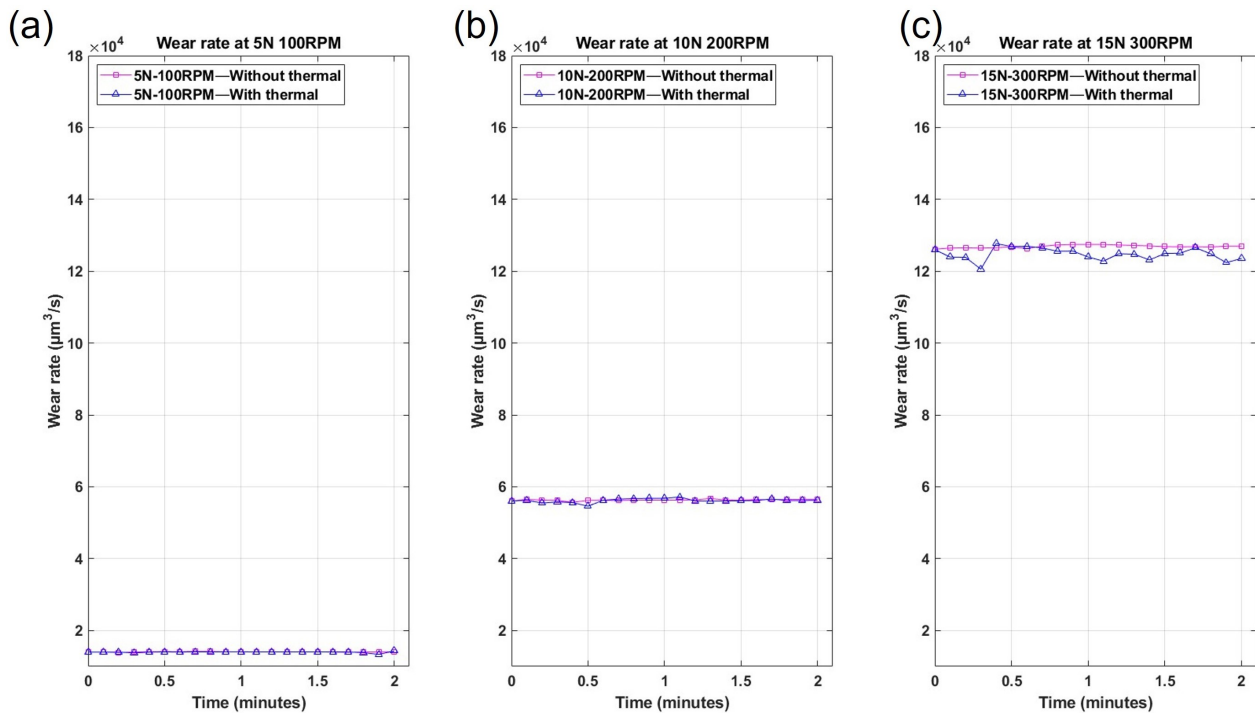


Figure 7. Comparison of the effective wear rate with and without thermal effects at (a) 5 N and 100 RPM, (b) 10 N and 200 RPM, and (c) 15 N and 300 RPM (the results are from the FEM simulation).

At lower loads and speeds, the wear rate difference between scenarios with and without thermal effects is minimal. However, as load and speed increase, the difference becomes more pronounced, especially at 15 N and 300 RPM. With thermal effects, the wear rate decreases and becomes lower than in scenarios without thermal considerations. This change is explained by differences in contact pressure and contact area. Under identical conditions, the differences between scenarios with and without thermal expansion lie in contact pressure and contact area. As previously analyzed, thermal expansion reduces contact area (decelerating wear) but increases average contact pressure (accelerating wear). These opposing trends create a competition mechanism influencing wear rate. Figure 7 shows that reduced contact area dominates this competition mechanism.

The authors of [28,29] also observed a decrease in wear rate with thermal expansion, but it cannot be concluded that thermal expansion universally decelerates wear. However, the authors of [30] reported an increase in wear rate with thermal expansion. Additionally, the experiments show that wear rate initially increases rapidly, then decreases as temperature rises further [31]. Thus, the relationship between thermal expansion and wear rate may depend on material properties and experimental parameters.

3.3. Wear Evolution

Figure 8 shows the evolution of wear distribution on the disc, highlighting significant differences between cases with and without thermal effects. Without thermal effects, wear primarily concentrates at the nominal contact center (20 mm from the disc center) between the disc and pin. From 0 to 0.4 min, wear depth develops rapidly. This occurs due to the small actual contact area from poor geometrical conformity during the running-in phase, concentrating the load on a few points and causing higher stress and wear. From 0.4 to 2 min, wear depth develops more slowly as the contact area enlarges, with the wear region gradually expanding outward from the nominal contact areas. However, the maximum wear depth remains at the nominal contact center. As the rotational speed increases from 100 to 200 and 300 RPM, wear depth slightly increases.

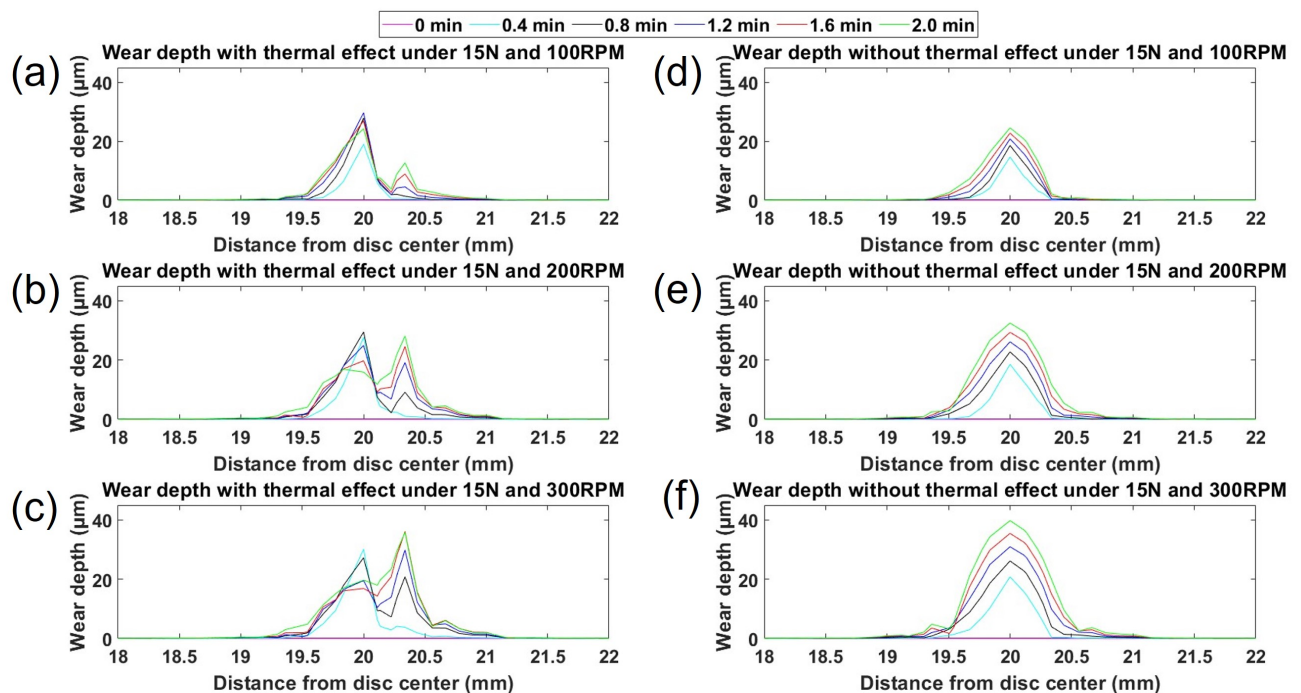


Figure 8. Wear evolution under (a) 15 N and 100 RPM with the thermal effect, (b) 15 N and 200 RPM with the thermal effect, (c) 15 N and 300 RPM with the thermal effect, (d) 15 N and 100 RPM without the thermal effect, (e) 15N and 200 RPM without the thermal effect, and (f) 15 N and 300 RPM without the thermal effect (the results are taken from the FEM simulation).

In contrast, thermal effects result in deeper wear. At 100 RPM, wear mainly develops at the nominal contact center, but at 200 RPM, an additional wear center appears outside this area. At 300 RPM, the depth of the newly formed wear center surpasses the original wear center. This shift in wear center location is attributed to changes in contact pressure, as shown in Figure 9. The outer region of nominal contact, with higher sliding speed, experiences greater thermal expansion. This increases pin-to-disc contact pressure, causing the original contact center to wear down quickly, shifting the wear center outward. This phenomenon appears only at higher loads (above 10 N) and speeds (200 RPM or more), indicating that sufficient energy is needed for the material to expand thermally enough to alter contact behavior.

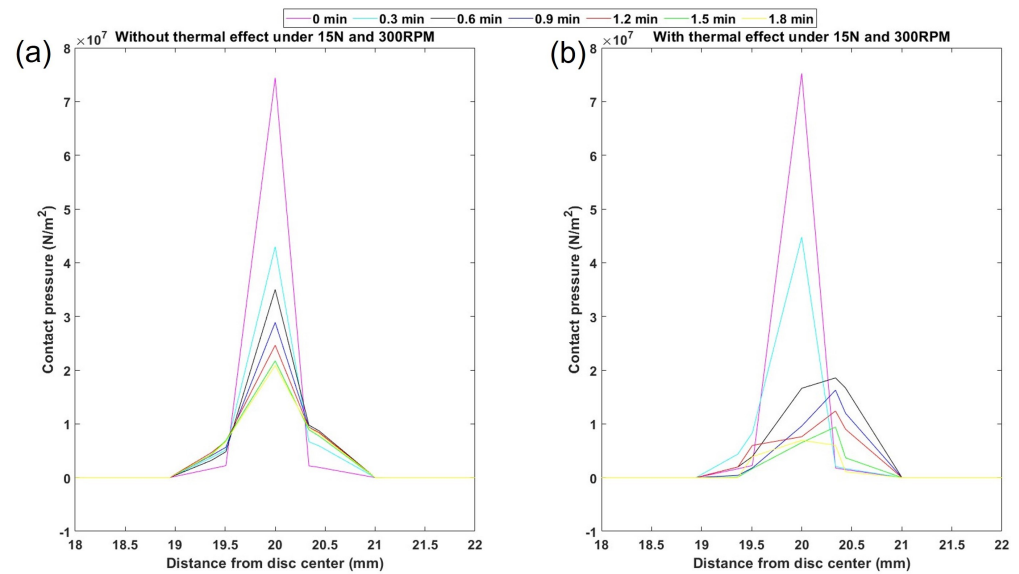


Figure 9. Comparison of contact pressure distribution along the radial direction for (a) 15 N and 300 RPM without the thermal effect, and (b) 15 N and 300 RPM with the thermal effect (the results are taken from the FEM simulation).

3.4. Wear Scar Profile

Figure 10 compares the wear scar profiles under different operating parameters in both simulations and experiments. Under a 5 N load, 300 RPM, and 1 min duration, the experimental wear scar profile aligns closely with the simulation results that include thermal effects. Under a 10 N load, the wear center shifts slightly, revealing two wear scar valleys. Under a 15 N load, these two wear scar valleys become more pronounced. Compared to the simulation without thermal effects, which shows wear only at the nominal contact center, the simulation with thermal effects better matches the experimental wear scar profile, showing two wear valleys. However, some overall differences remain between the experimental and thermal simulation wear scar profiles. This deviation may result from unaccounted randomness in the wear process or slight environmental temperature differences between the experimental and simulation conditions.

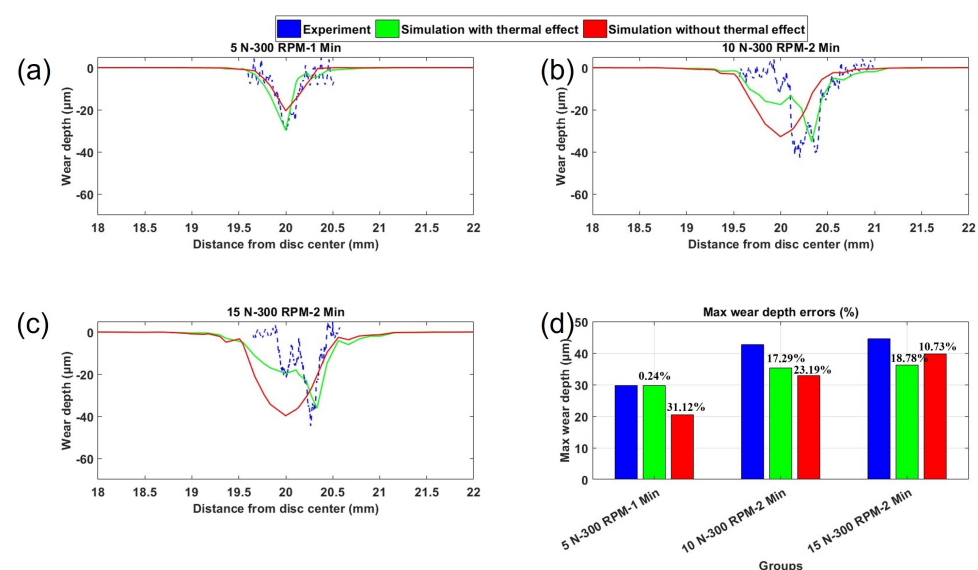


Figure 10. Comparison of the wear scar profiles between the experiments and simulations for (a) 5 N, 300 RPM, and 1 min, (b) 10 N, 300 RPM, and 2 min, and (c) 15 N, 300 RPM, and 2 min. (d) The maximum wear depth difference between the experiments and simulations with and without thermal effects.

Overall, when load and speed are sufficient to activate thermal expansion, the simulations with thermal effects provide better predictions of the wear scar profiles. However, they still tend to overestimate wear scar width compared to the experimental results.

3.5. Interface Temperature

Figure 11 shows the temperature on the pin and disc, comparing scenarios with and without wear. Note that in the scenario without wear, thermal expansion and material softening are still considered. The highest temperature occurs at the contact center, forming a comet-shaped pattern on the disc that decreases along the direction of friction. This observation aligns with the findings in [32,33]. On the pin, the temperature spreads in a circular pattern, decreasing uniformly in all directions.

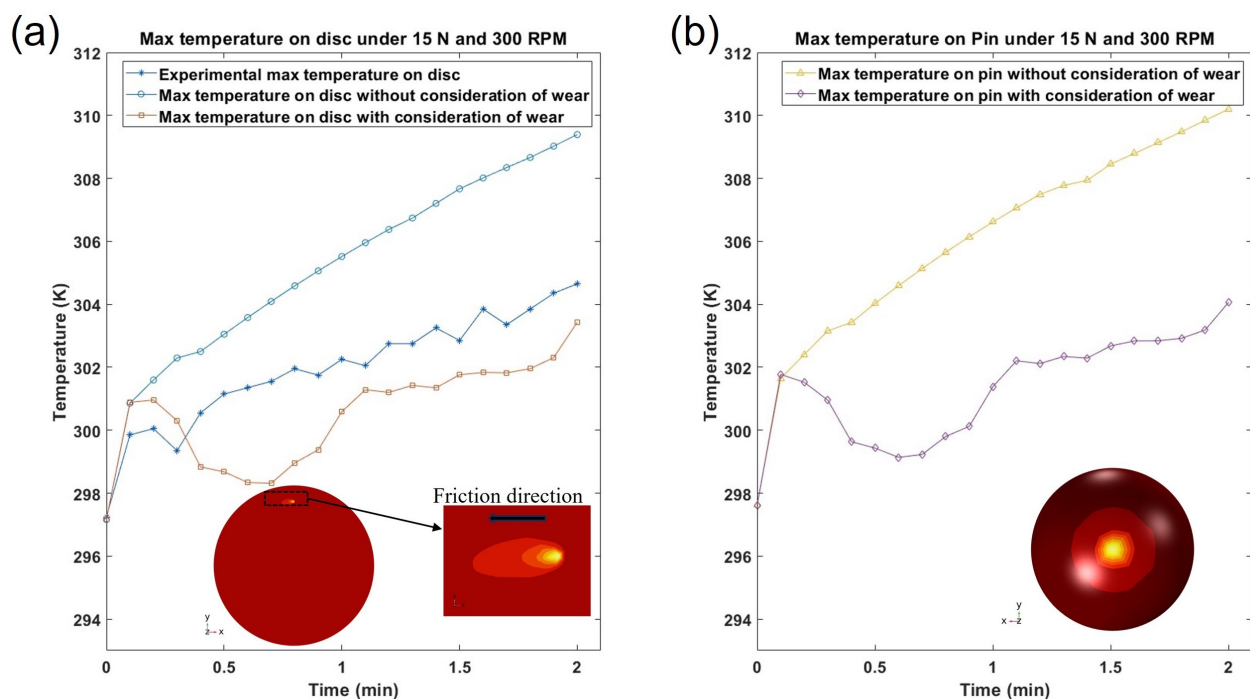


Figure 11. Comparison of the maximum temperature between the experiments and simulations on the (a) pin and (b) disc under 15 N and 300 RPM.

Under identical conditions, the maximum temperature on the pin and disc shows a similar fluctuating pattern, with the pin slightly hotter. The rise in maximum temperature varies between scenarios with and without wear. For the first 0.1 min, the maximum temperatures in both scenarios are identical, as wear has not yet significantly affected contact behavior. As wear progresses and further impacts contact behavior, temperature differences become apparent. The simulation without wear shows a higher, steadily increasing temperature during sliding. In contrast, the simulation with wear exhibits fluctuations due to more complex contact behavior. The measured temperatures align more closely with the simulation, which includes wear. However, discrepancies remain, possibly due to the initial temperature differences between the simulations and laboratory conditions.

Figure 12 compares the temperature distribution evolution on the pin. An interesting observation emerges: with wear, the heat source expands along the friction direction due to sliding and shifts toward the center of the disc. The heat source then shifts in the opposite direction, eventually positioning itself within the nominal contact center. In contrast, without wear, the heat source remains nearly stationary at the pin center but still expands along the friction direction.

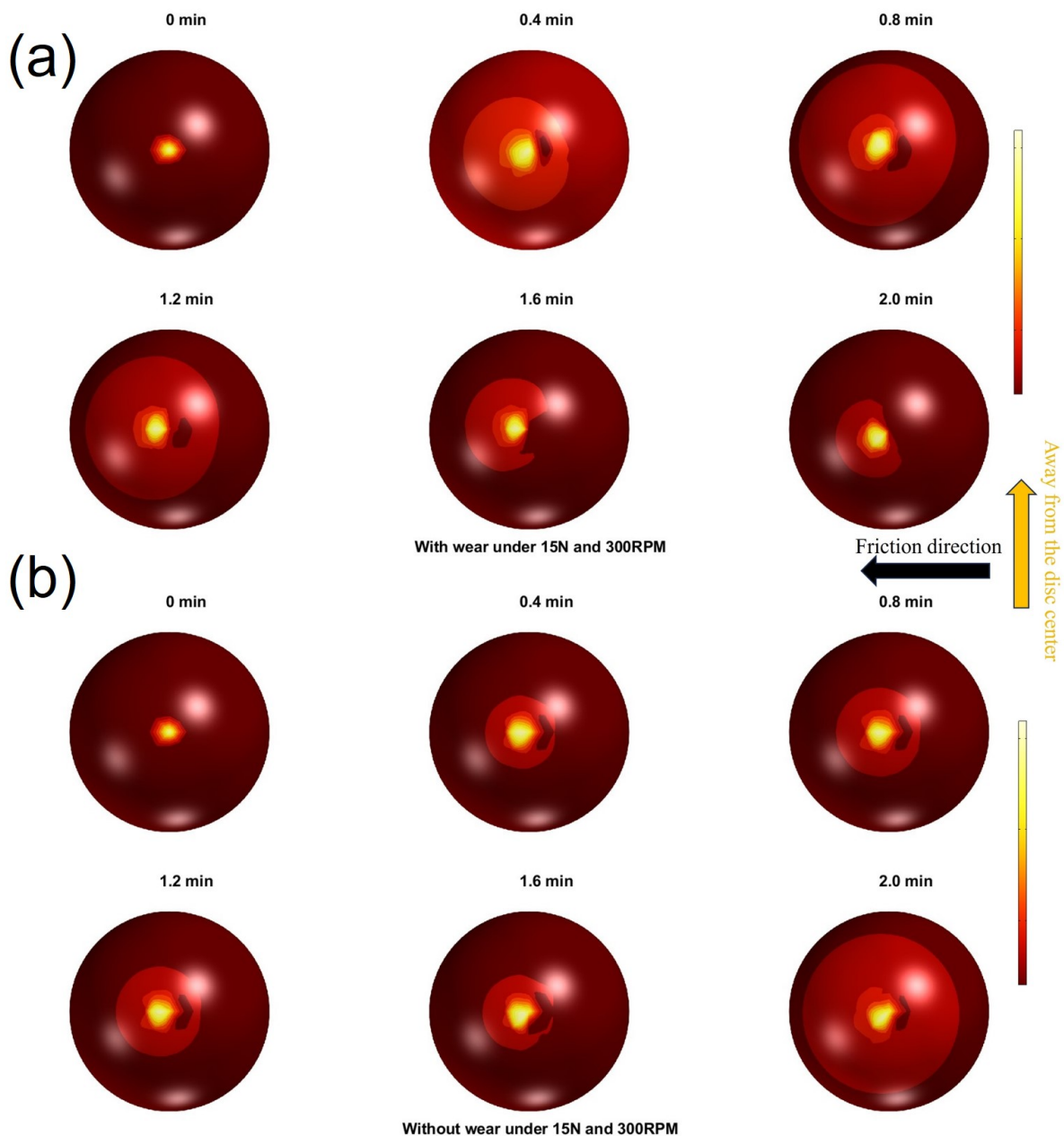


Figure 12. Heat source movement under 15 N and 300 RPM; (a) with wear considered; (b) without wear considered (the results are taken from the FEM simulation).

3.6. Thermal Expansion of Disc

Figure 13 illustrates deformation from thermal expansion in the tribological system, scaled eight times for clarity. With increased contact time between the pin and disc, the system absorbs more thermal energy, resulting in greater thermal expansion. Although the pin experiences a higher temperature increase than the disc, its thermal expansion remains minimal due to its thermal properties. In contrast, the disc shows significant, uneven deformation.

On the disc's top surface, the outer region, closer to the heat source during sliding, accumulates more thermal energy, causing more deformation than the inner region. This results in circular deformation bands on the upper surface. Vertically, the top surface exhibits the most deformation, causing the radial expansion of the disc. As frictional sliding continues, this deformed region gradually moves downward, giving the disc a causeway-like shape with narrow ends and a wider center.

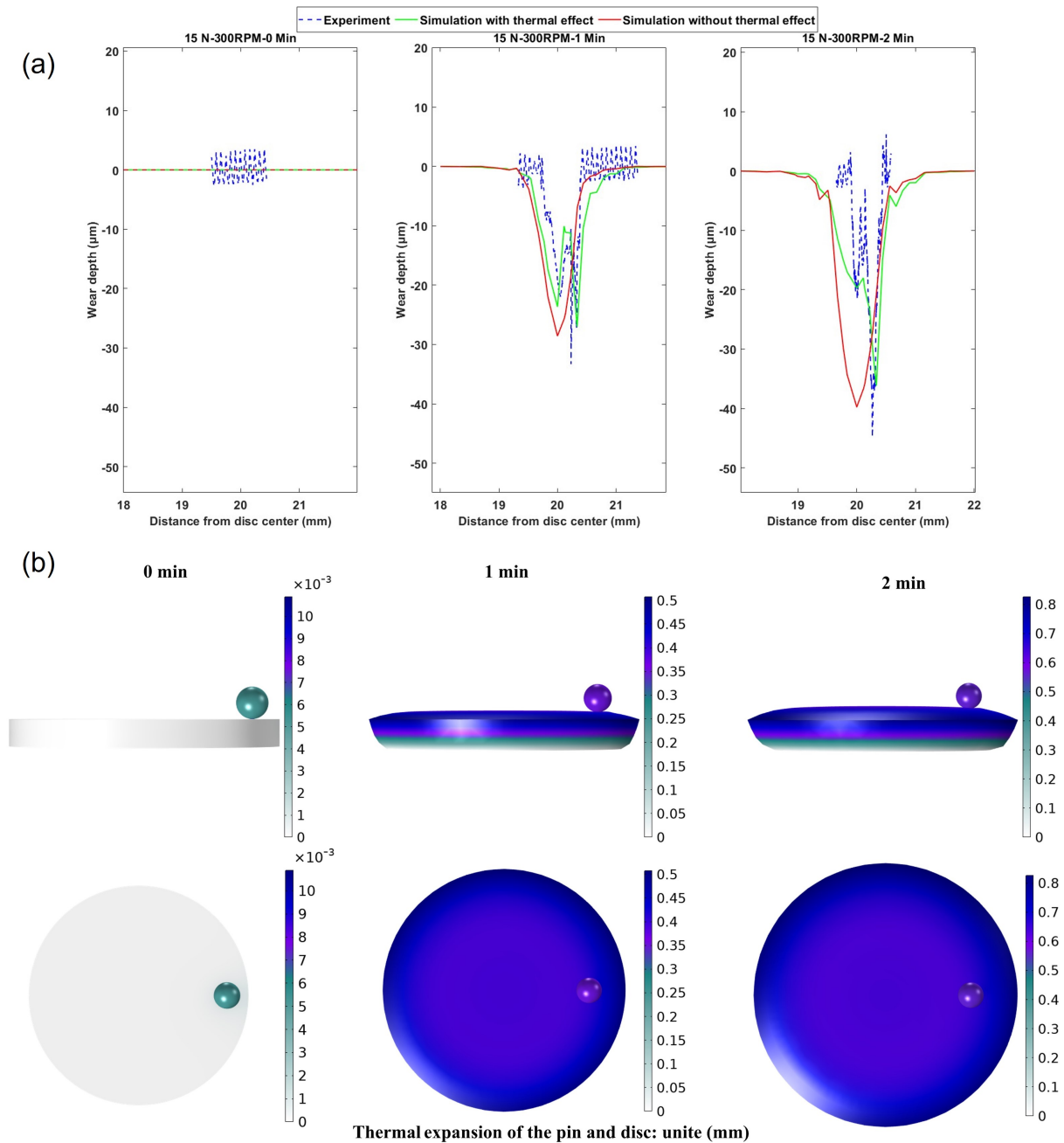


Figure 13. (a) Comparison of the wear scar profiles between the experiments and simulations under 15 N and 300 RPM, with and without wear considered. (b) Deformation of the pin and disc due to thermal expansion under 15 N and 300 RPM from the FEM result, with wear considered.

3.7. How Do Wear and Thermal Expansion Influence Temperature

Wear and thermal expansion influence contact behavior between frictional pairs, impacting interface friction and thereby affecting the frictional dissipation rate. Consequently, the effects of wear and thermal expansion on temperature rise can be analyzed through their impacts on friction and frictional dissipation during sliding.

Figure 14 compares friction dissipation rates under various operating parameters. Increased load and rotational speed both elevate the friction dissipation rate. Under a low load or speed, the differences in friction dissipation rates are minimal. Under higher loads or speeds, the differences between the three conditions become more pronounced. Initially, the friction dissipation rate, with both wear and thermal effects considered, is similar to scenarios without wear or thermal effects. Over time, the rate decreases compared

to the other two conditions, and this difference becomes more pronounced as sliding distance increases.

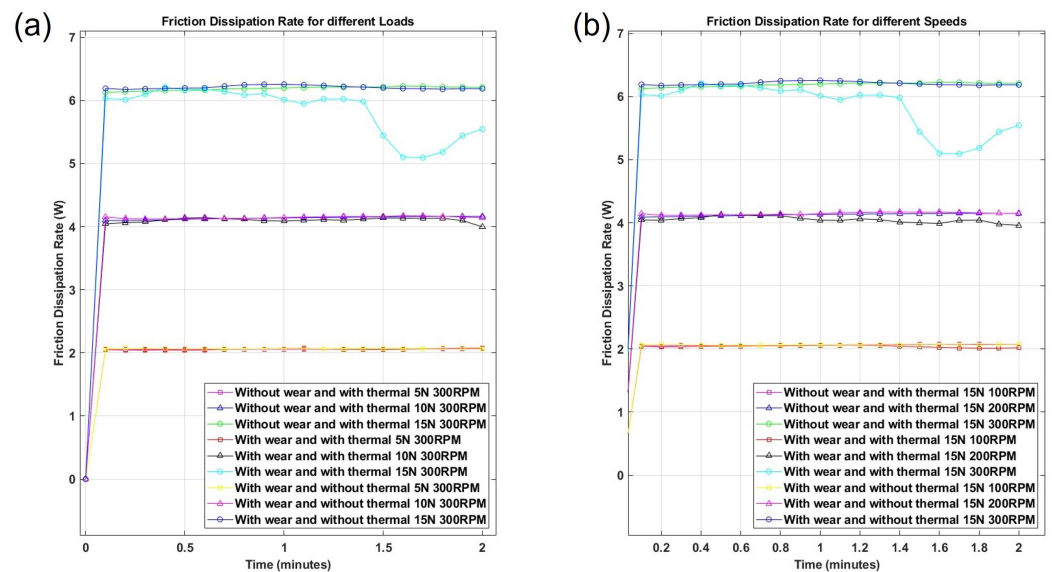


Figure 14. Frictional dissipation rate under (a) different sliding speeds and (b) different loads (the results are taken from the simulation).

To further analyze this phenomenon, Figure 15 shows the friction distribution along the disc radius under 15 N and 300 RPM. With only thermal effects considered, friction distribution remains nearly constant during sliding, maintaining a steady friction dissipation rate. When only wear is considered, friction decreases over time while contact area increases, resulting in dynamic competition, which explains the friction dissipation rate fluctuation at 15 N and 300 RPM in Figure 14. With both thermal effects and wear considered, maximum friction is lower compared to scenarios with only wear or thermal effects (Figure 15).

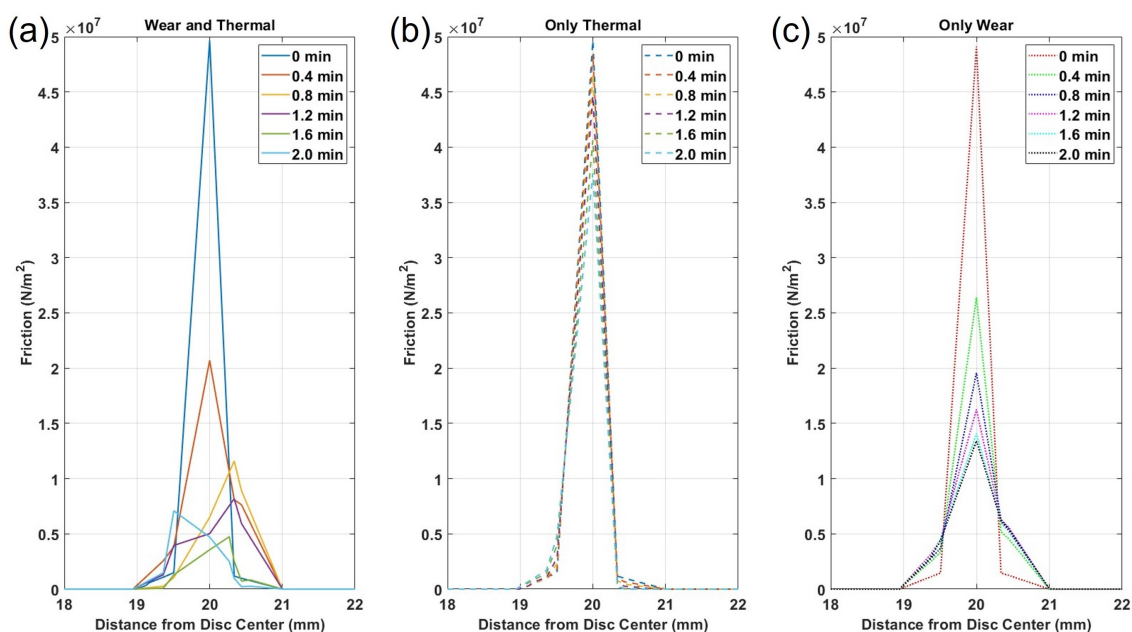


Figure 15. Friction distribution along the disc radius under 15 N and 300 RPM, (a) considering both wear and thermal effects, (b) considering only thermal effects, and (c) considering only wear (the results are taken from the simulation).

Figure 14 compares the friction dissipation rate under various operating parameters. Both increased load and rotating speed elevate the friction dissipation rate. Under a low load or speed, the differences in friction dissipation rates are not significant. However, with a higher load or speed, the differences between the three conditions become more pronounced. Initially, the friction dissipation rate, when considering both wear and thermal effects, is close to the scenarios without wear or without thermal effects. Over time, the rate becomes lower than the other two conditions, and this difference becomes more significant as the sliding distance increases. Additionally, as shown in Figure 5, the contact area decreases when both factors are considered. This reduction in friction and contact area leads to a lower friction dissipation rate compared to the other two scenarios (Figure 14).

In summary, the combined effects of wear and thermal expansion result in a lower temperature rise than considering each factor alone, as they alter the friction, contact area, and frictional dissipation rates.

4. Conclusions

This study investigates the interplay between wear and thermal expansion in 6082 aluminum, using a pin-on-disc tribometer and FEA under various mechanical conditions. The focus is on understanding how these phenomena influence each other, impacting wear evolution and temperature rise during sliding contact.

The key findings from our research include the following:

- Thermal expansion creates a protrusion at the contact area, reducing the contact area compared to wear-only conditions.
- The combined effects of wear and thermal expansion create a dynamic interplay, shifting the contact region and pressure distribution outward from the disc center. This dynamic contact behavior leads to distinct wear evolution and temperature distribution compared to scenarios with only wear or thermal expansion.
- Thermal expansion alters wear patterns, with high expansion shifting the maximum wear depth from the nominal contact center to outer regions, especially under higher rotational speeds and loads.
- Without thermal expansion, wear-only conditions overestimate the friction dissipation rate, resulting in a higher maximum temperature than when both wear and thermal expansion are considered.
- The disc undergoes significant radial expansion due to uneven thermal energy accumulation, with the outer regions deforming more than the inner ones. Over time, this deformation progresses downward, giving the disc a causeway-like shape with narrow ends and a wider center.

These findings enhance our understanding of the complex relationship between wear and thermal expansion in 6082 aluminum under sliding contact. However, this study has limitations, including not accounting for the randomness of wear. This omission may cause the model's predicted wear depth to differ significantly from experimental results under light loads and slow sliding speeds, where surface roughness and trapped debris dominate wear. Additionally, focusing solely on 6082 aluminum limits the results' applicability to other materials without further investigation.

Future research could explore this wear-thermal expansion interplay in other materials and examine the effects of lubrication and surface treatments. Such studies could offer deeper insights for optimizing tribological performance. Thermal expansion shifts wear patterns from the center to the periphery under specific conditions, helping engineers optimize operating conditions or redesign components to prolong the lifespan of frictional parts. Considering wear-induced changes in contact behavior is essential for the accurate design and evaluation of frictional heat, as ignoring thermal effects may result in temperature overestimation. Thermal expansion affects frictional dissipation rates during sliding, and managing this interaction helps reduce frictional losses. This optimization lowers energy consumption in mechanical systems such as engines and motors, supporting

energy-efficient designs. Additionally, air cooling can be used to reduce the effects of thermal expansion to validate future simulations that ignore thermal effects.

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Interplay between wear and thermal expansion in 6082 aluminum: a simulation and experimental study

Tian, Yang; Kalifa, Mohamed; Khan, Muhammad; Yang, Yifan

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