

Influence of sliding direction relative to layer orientation on tribological performance, noise, and stability in 3D-printed ABS components

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ABSTRACT

The tribological performance of 3D-printed ABS components is influenced by layer orientation, yet its effects on friction, wear, noise, and system stability remain underexplored. This study investigates how the angle between sliding direction and layer lines (0° , $\pm 45^\circ$, 90°) impacts these properties in FDM-printed ABS. Pin-on-disc tests (10–20 N loads, 0.314–0.628 m/s speeds) and modeling (FEM for wear and temperature, lumped-parameter for stability) were conducted. The 90° orientation showed the highest coefficient of friction (COF) due to mechanical interlocking but the lowest wear, while the 0° orientation had the lower COF and highest wear from interlayer shear. The -45° orientation produced the most noise due to debris-induced stick-slip, while the 45° orientation generated the least. FEM wear predictions aligned well with experiments ($<7\%$ error), but noise predictions had higher errors (up to 15 %). Increased wear depth raised vibration frequencies, and larger static-kinetic COF differences increased instability.

1. Introduction

3D printing, or additive manufacturing, has redefined modern manufacturing by delivering exceptional design flexibility [1–3], cost efficiency [4,5], and rapid prototyping [6,7]. This technology excels in producing intricate geometries and customized products with minimal waste, establishing its indispensability across sectors like automotive, aerospace, healthcare, and construction. By enabling precise, on-demand production, 3D printing fosters sustainable and efficient alternatives to traditional methods. The field has evolved to encompass diverse techniques, such as Stereolithography (SLA), Selective Laser Sintering (SLS), Digital Light Processing (DLP) [8], and Powder-Based 3D Printing [9]. Among these, Fused Deposition Modelling (FDM) stands out for its affordability and use of thermoplastics like Polylactic Acid (PLA) and Acrylonitrile Butadiene Styrene (ABS) [10]. FDM's accessible equipment, user-friendly operation, and minimal training needs [11] have made it a go-to solution for prototyping and small-scale manufacturing.

ABS, a durable thermoplastic, is prized for its high impact resistance, strength, and thermal stability [12,13], outperforming PLA in high-temperature environments. Its post-processing versatility—allowing drilling, sanding, and machining [14]—makes it ideal for functional

components like gears and seals. Understanding the wear performance of FDM-printed ABS is vital for ensuring its longevity in mechanical applications, prompting extensive research into optimizing printing parameters and material properties. For example, Sood et al. [15] demonstrated that parameters like layer thickness, build orientation, and raster angle significantly enhance wear resistance in sliding-contact parts. Equbal et al. [16] used response surface methodology to develop predictive wear models, while Wang et al. [17] linked higher nozzle temperatures to reduced wear rates. Hasdiansah et al. [18] identified layer thickness as critical for surface smoothness in TPU (Thermoplastic Polyurethane), and Zhang et al. [19] optimized PLA wear (300 μm thickness, 45° orientation, 225°C) for biodegradable applications using Taguchi and Artificial Neural Network (ANN) models.

Material-specific studies further illuminate ABS's potential. Vaught et al. [20] showed that the wear resistance of Polyethylene Terephthalate Glycol (PETG) improves with oxidation-induced transfer films and annealing, while Kumar et al. [21] achieved superior flexural strength (95.91 MPa) and wear resistance in PLA-PA6/TiO₂ composites for biomedical scaffolds. Dontsov et al. [22] enhanced Ultra-High Molecular Weight Polyethylene (UHMWPE) composites for medical implants, improving tensile strength and wear resistance. Tribological research offers additional insights: Hanon et al. [23] found inconsistent

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friction trends in ABS, and Ciobanu et al. [24] noted PLA gears outperform ABS in wear resistance under unlubricated conditions. Composite reinforcements also shine—Bodaghi et al. [25] reported that 5 % glass fibre in ABS boosts thermal stability and wear performance, while Grejtek et al. [26] identified optimal whisker orientations for epoxy nanocomposites, achieving peak wear resistance. Dhakal et al. [27] examined the influence of raster angle orientation and printing speed on processing defects, mechanical properties, surface quality, and tribological performance of 3D-printed ABS polymers. However, their study does not address frictional noise or the stability of tribological systems.

Despite these advances, a critical aspect of FDM remains underexplored: the layer lines inherent to the process create angles relative to sliding directions during frictional contact, yet their impacts on wear, frictional noise, and temperature remain unresolved. This study addresses this gap by validating Tian et al.'s model [28], which correlates frictional noise, wear, and the COF—previously validated in ball-disc sliding—with respect to FDM-printed ABS. This research investigates how layer line orientation affects frictional noise and wear, employing a finite element method (FEM) model to compare worn surface profiles, wear volume, and temperature with experimental data. By integrating contact area and asperity height into a lumped-parameter system, this study evaluates how wear depth and differences between static and kinetic COF influence system stability. These findings may guide the optimal design and installation of FDM-printed ABS components in sliding applications, thereby improving their practical utility.

2. Methodology

2.1. Preparation of 3D-printed specimens

The 3D-printed specimens were designed *NX 12.0* as cubes with dimensions of 6x6x3 mm. Each cube contains a hollow section measuring 3x3x1.5 mm, designed to form an interference fit with the tribometer's pin holder. The models were then imported into *UltiMaker Cura 5.8.1* to generate 3D printing codes for the *UltiMaker 2 +* printer. The material utilized for 3D printing is an ABS filament, manufactured by *UltiMaker*, with a diameter of 2.85 ± 0.10 mm, and it is a composite filament comprising modified ABS blended with polyethylene terephthalate and polycarbonate. Table 1 displays the mechanical and thermal characteristics of ABS. The printing parameters were set as shown in Table 2. The complete specimen preparation process is illustrated in Fig. 1.

2.2. Wear experiment setup

The wear experiments were performed in a dry environment utilizing the *Anton Paar TRB3 tribometer*. A visual representation of the experimental configuration is provided in Fig. 2. Pin-on-disc tests were conducted on 3D-printed ABS samples sliding against 6082 aluminium discs. The material properties of the disc are presented in Table 3. The layer lines of the ABS samples were oriented at specific angles (0° , -45° , 45° , and 90°) relative to the disc sliding velocity direction. Tests were performed for 2 minutes at sliding speeds of 0.314–0.628 m/s and loads of 10–20 N (see Fig. 3). The mass of the tested samples was measured

Table 2
3D print settings.

Setting	Value
Nozzle diameter	0.4 mm
Layer Height	0.1 mm
Initial Layer Height	0.27 mm
Line Width	0.4 mm
Wall Line Width	0.4 mm
Outer Wall Line Width	0.4 mm
Inner Wall(s) Line Width	0.4 mm
Infill Line Width	0.4 mm
Skirt/Brim Line Width	0.4 mm
Initial Layer Line Width	100.0 %
Print Speed	45.0 mm/s
Infill Speed	45.0 mm/s
Wall Speed	30.0 mm/s
Fan Speed	100.0 %

before and after the wear test to obtain the wear loss and calculate the wear coefficient, see Fig. 2(d). Temperature measurements were conducted in a laboratory environment with a temperature range of 21–23°C. The friction noise during the wear test was recorded using a microphone.

2.3. Simulation setup

The wear process encompasses multiple physical phenomena, including thermal effects, mechanical interactions, and material degradation. To ensure an accurate simulation, *COMSOL Multiphysics* was utilized, as it offers a robust platform for integrating these interrelated factors. The corresponding simulation framework is depicted in Fig. 4.

The simulated model featured a 3D-printed pin interacting with a disc surface, as shown in Fig. 5(a). During the simulation, the pin moved across the disc at a radial distance of 15 mm from the centre. The boundary conditions of the simulated system are illustrated in Fig. 5(b). The disk mesh used a tetrahedral mesh. To ensure precise results while maximizing computational efficiency, the mesh in the contact area between the disc and pin was locally refined. To enhance computational efficiency, the pin was meshed using a structured mesh (see Fig. 5(c)). Thus, the disc was segmented into 30,295 elements, and the pin into 18,207 elements.

The Penalty Method was employed to simulate the contact behaviour between the ABS pin and aluminium disc. Although the Augmented Lagrangian Method provides more accurate results, it has high computational overhead and can easily lead to convergence problems [29,31]. The Penalty Method is widely used in multiphysics field coupling due to its simplicity, efficiency, and better stability. The simulation of the wear process was conducted using the Archard model, as shown in Eq. (1).

$$w = k \cdot \frac{F_n \cdot V}{H} \quad \# \quad (1)$$

This equation establishes the relationship between the wear rate w , wear coefficient k , applied normal load F_n , relative sliding speed V , and the hardness of the softer material H . To enhance the stability and accelerate the convergence of the FEM model, the coefficient of friction

Table 1
Mechanical and thermal properties of ABS.

Notation	Characteristics	Value	Unit
ρ	Density	1050	kg/m ³
$E(T)$	Young's Modulus	$E(T) = \frac{1043.3}{(1 + \exp(0.2229 * (T - 376.0206)))}$	Pa
T	Temperature	293.15–413.15	K
ν	Poisson's Ratio	0.3	-
C_p	Heat Capacity	1350	J/(kg·K)
k	Thermal Conductivity	0.18	W/(m·K)
α	Thermal Expansion Coefficient	0.0000982	1/K

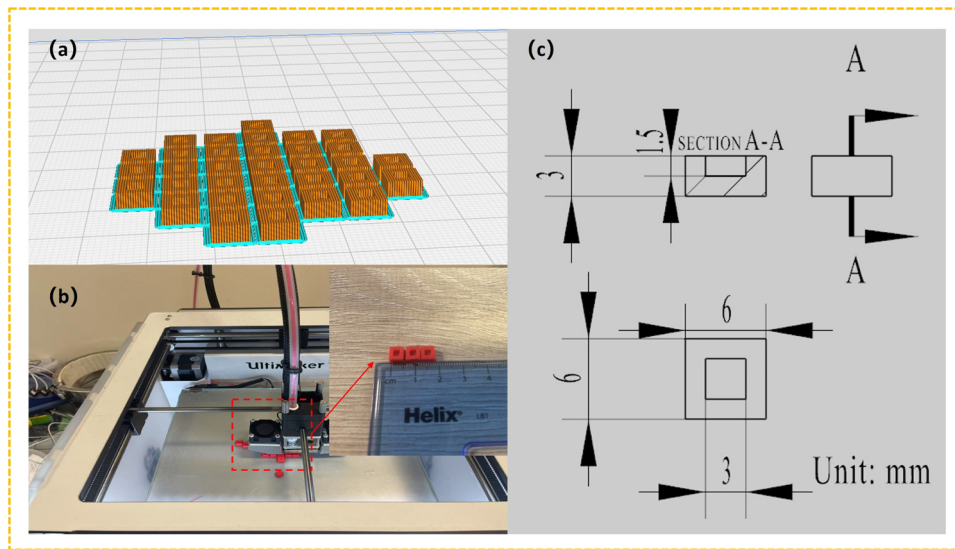


Fig. 1. illustrates the following steps: (a) slicing the sample and setting 3D printing parameters in Cura, (b) printing the samples, and (c) sample dimensions.

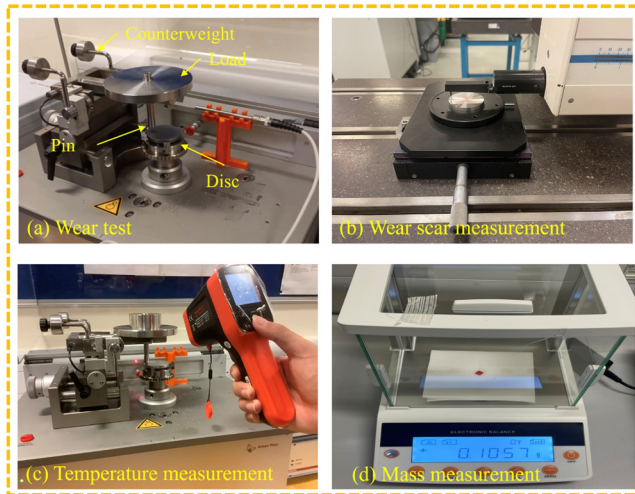


Fig. 2. (a) wear test, (b) profile measurement before and after wear test, (c) temperature measurement, (d) wear loss measurement before and after wear test.

μ is approximated as a constant average value.

The solid mechanics model's contact behaviour was transferred to the heat transfer model to identify the heat source from frictional dissipation. To improve the simulation's accuracy, heat transfer, convection, and radiation were taken into account. The Charron relationship was utilized to establish the thermal partition coefficient between the pin and disc.

The wear coefficient was determined based on the volume of material lost during the wear test. Predictions of surface wear and temperature from the FEM model were then evaluated against experimental results for validation.

2.4. Noise model validation

As wear advances, the surface morphology undergoes significant changes that in turn affect the interface contact (contact area and stiffness, friction, pressure distribution). These fluctuations act as sources of excitation, triggering vibrations that propagate across the system. Consequently, these vibrations generate noise, with their properties and volume directly related to the degree and severity of wear at the contact

Table 3

Thermal and mechanical properties of disc.

Symbol	Property	Value	Unit
ρ_1	Density	2700	kg/m ³
$E_1(T)$	Young's Modulus	$E_1(T) = E_{10} \left(1 - \frac{T}{T_m} \exp \left(\theta_g \left(1 - \frac{T_m}{T} \right) \right) \right)$ [30]	Pa
E_{10}	Young's modulus at 0 K	7.20E+ 09	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	1
C_{p1}	Heat Capacity	897	J/(kg·K)
k_1	Thermal Conductivity	195	W/(m·K)
α	Thermal Expansion Coefficient	2.30E−05	1/K
σ_{y50}	Initial Yield Stress	2.71E+ 08	Pa
H	Hardness	90	MPa

surface. Therefore, it can be used as a non-invasive inspection parameter to predict surface damage in mechanical contacts as listed in published reviews [32].

The complex interplay between wear and noise poses substantial challenges in developing a unified analytical model that effectively encapsulates their interdependent behaviours. Addressing this issue through finite element modelling (FEM) is particularly complex as it requires the integration of numerous interdependent factors. These include surface topography, variations in the COF during the transition from static to kinetic friction in sliding motion, and inclusion of inertial effects to replicate vibrational behaviour. Additionally, the necessity of coupling multiple physical domains, such as solid mechanics and acoustic pressure, further escalates computational demands while potentially compromising the precision of FEM simulations.

Basit et al. [33–35] developed an analytical model to bridge friction-induced noise and wear evolution based on a ball-on-disc system, but it accounts solely for the ball's role in generating acoustic emissions, which leads to an oversimplified representation of the phenomenon and limits its applicability to real-world scenarios. Lontin et al. [36–38] proposed an analytical model in which surface asperities are treated as beams whose vibrations generate noise upon impact. However, this model only considers tangential forces to determine wear and

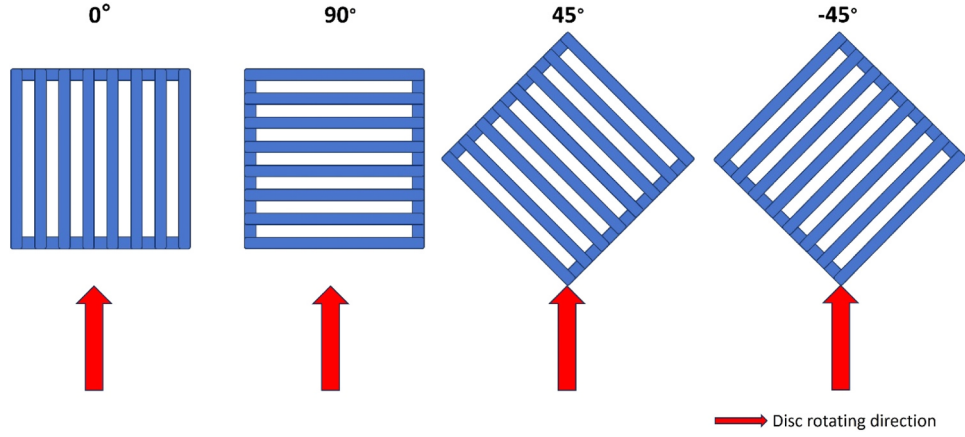


Fig. 3. Illustration of the angle between the layer lines and the disc's rotating direction.

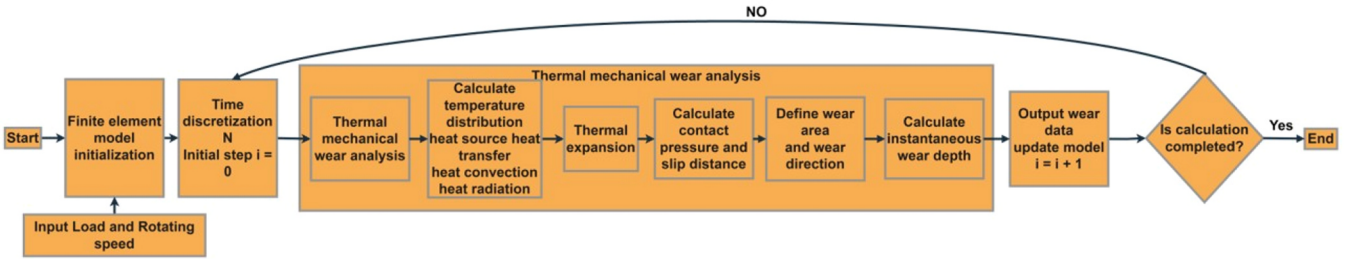


Fig. 4. FEM simulation framework [29].

vibration, neglecting the effects of normal forces as conventionally considered in simple beam damage dynamics [39]. Additionally, the model reduces the 3D frictional system to a 2D approximation, which creates a larger deviation from experimental results compared to Tian et al.'s approach [28]. Tian et al. [28] developed a symbolic regression-based model linking wear volume, cumulative noise pressure, and the COF, as shown in Eq. (2). However, this model was derived and validated exclusively for ball-on-disc sliding. In this study, we aim to verify its applicability to flat-on-disc sliding contact.

$$P_{\text{cumulative}} = A \cdot \sqrt{V_m} \cdot \mu \quad (2)$$

Here, $P_{\text{cumulative}}$ is the cumulative noise pressure (unit: Pa), V_m is wear volume (unit: m^3), μ is the COF, and A (unit: $\text{kg} \cdot \text{m}^{-4} \cdot \text{s}^{-2}$) is a constant derived by fitting data from various experimental conditions.

3. Result and discussion

3.1. Coefficient of friction

Fig. 6 illustrates the progression of the mean Coefficient of Friction (COF) derived from experiments conducted in triplicate over 120 seconds under varying conditions, presented in three subplots corresponding to applied force and sliding speed levels of 10N-0.314 m/s, 15N-0.471 m/s, and 20N-0.628 m/s. Each subplot contains several curves corresponding to the different angles between the sliding direction and the layer line. Error bands are shown around the curves, representing variability in the measurements. Across all subplots, COF increases non-linearly over time, reaching a plateau or near-stabilization after approximately 100 seconds. The rate of increase is higher in the initial phase (0–40 seconds) and slows down as time progresses. When the load increases from 10 N to 20 N and the sliding speed rises from 0.314 m/s to 0.628 m/s, the COF remains relatively stable. This phenomenon can be attributed to the opposing effects of increased load and

sliding speed. As the sliding speed increases, substantial frictional heat is generated. This heat softens the ABS material, thereby reducing the resistance of interacting asperities to deformation, and consequently lowering the coefficient of friction. On the other hand, higher loads increase the real contact area between surfaces, resulting in greater adhesive friction and a rise in the COF [40,41]. The simultaneous increase in both parameters leads to complex interactions. Frictional heat from high sliding speeds, combined with surface deformation under heavy loads, creates a balance: reduced COF due to surface smoothing and increased COF caused by wear and enhanced adhesion.

Under the same load and sliding speed, the angle between the disc's sliding direction and the 3D print layer lines significantly influences the COF. The 90° orientation exhibits the highest COF, followed by the 0° orientation, while the -45° and 45° orientations have the lowest COF. At 90°, the sliding motion is perpendicular to the layer line direction. This orientation creates significant resistance because the asperities (ridges and valleys formed during 3D printing) directly obstruct the sliding motion. Each ridge acts as a barrier, causing mechanical interlocking and intensifying ploughing effects, thereby increasing the COF. At 0°, the sliding motion aligns with the layer lines. This alignment minimizes asperity engagement and reduces ploughing effects compared to the 90° orientation. At $\pm 45^\circ$, the sliding motion cuts diagonally across the layer lines. This orientation reduces direct asperity engagement and facilitates a smoother, more continuous sliding motion compared to the 90° orientation [40,41].

Additionally, Fig. 9 and Fig. 10 (a) show that the $\pm 45^\circ$ orientation has a lower wear rate than the 0° orientation. This indicates reduced energy dissipation as mechanical deformation during sliding, under the same sliding speed and distance, which corresponds to lower friction. Therefore, the $\pm 45^\circ$ orientation reduces frictional energy dissipation, namely COF, due to its lower wear rate. However, Hanon et al. [41] showed that the horizontal orientation exhibited the lowest friction coefficient, the vertical orientation showed the highest, and the 45° orientation displayed intermediate COF values. This difference could be

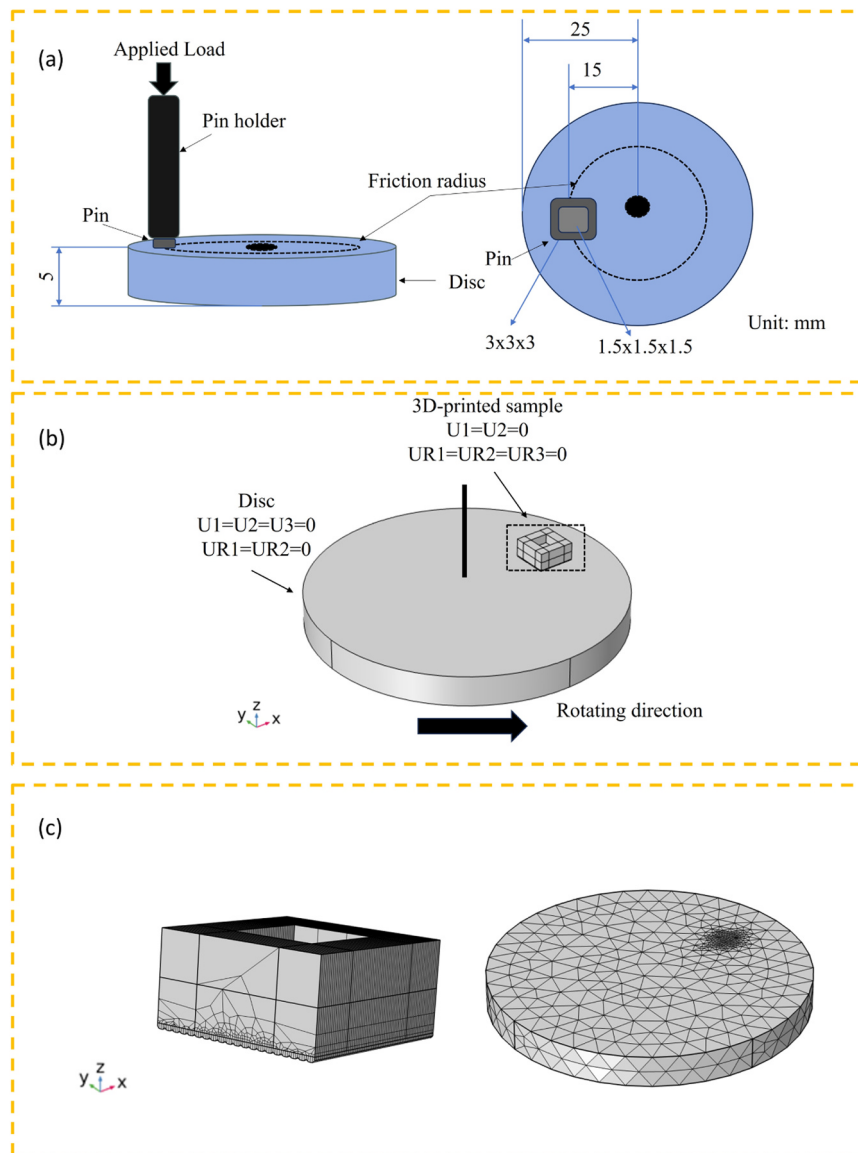


Fig. 5. (a) Dimensions of the sample, (b) Boundary conditions, (c) FEM meshing.

attributed to the use of a cylinder-flat contact pair in their study, whereas this research employs a flat-flat contact pair.

3.2. Wear surface morphology

Fig. 7 presents the wear surface morphology observed using optical microscopy, under a load of 20 N and a sliding speed of 0.628 m/s. The forefront of the surface, which faces the sliding disc, exhibits the highest wear depth. The wear depth gradually decreases along the sliding direction, reaching the lowest at the farthest extremity. Similarly, regions farther from the disc centre exhibit higher wear depths, attributed to the increased relative sliding speed.

Fig. 8 shows the wear depth distribution abstracted from FEM simulations. The depth distributions align with the trend observed from the wear surface morphology in Fig. 7. Considering the overall trend, 0° exhibits the highest maximum wear depth of 180 μm . Both -45° and 45° show relatively lower wear depths of 160 μm , while 90° presents the lowest maximum wear depth of only 35 μm .

The wear loss at the leading edge of a flat pin sliding against a disc is greater than that at the trailing edge (the direction from the leading edge to the trailing edge aligns with the direction of disc motion). This occurs

because the forefront of the pin typically experiences higher contact pressures during sliding, resulting from uneven load distribution. This is due to the leading edge being the initial contact point with the disc's surface asperities, which causes concentrated stress and generates higher frictional heat, as shown in Fig. 18. The generated heat softens the material, reducing its resistance to abrasion and adhesion, thereby exacerbating wear [41]. Material displaced by wear at the forefront is transferred to the back end. This dynamic causes more severe wear at the forefront because debris reduces friction at the trailing edge [42].

To compare the worn surface morphology between experiments and simulations, the profiles marked in Fig. 7 (red, black, and blue dashed lines) were measured. Fig. 9 compares the experiment wear profile and wear depth distribution abstracted from FEM simulations.

In the upper 2 mm region, 0° exhibits a maximum wear depth of approximately 135 μm , while -45° and 45° show comparable values. Specifically, -45° has a slightly higher maximum wear depth of 110 μm compared to 45° with 100 μm . In the middle region, 0° exhibits a maximum wear depth of about 160 μm . -45° and 45° have similar values, with -45° showing a slightly higher maximum wear depth of 150 μm compared to 45° at 145 μm . In the lower 2 mm region, 0° exhibits a maximum wear depth of approximately 120 μm . -45° and 45°

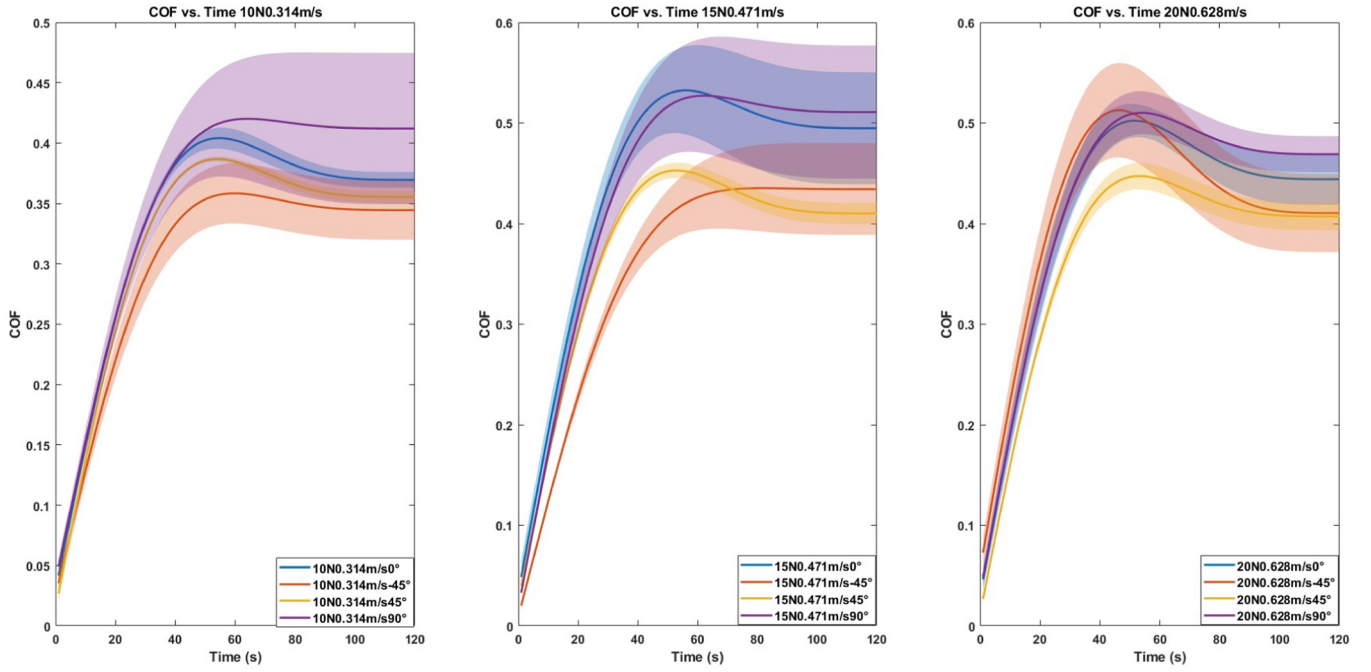


Fig. 6. Coefficient of friction under different conditions.

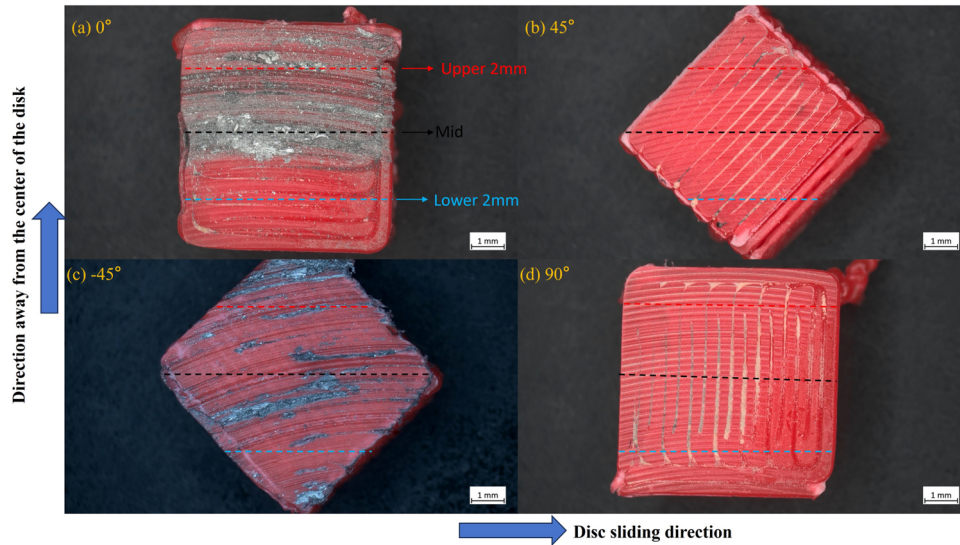


Fig. 7. Wear surface morphology observed using optical microscopy under 20N-0.628 m/s.

are again comparable, with -45° showing a slightly higher wear depth of $100\ \mu\text{m}$ compared to 45° at $95\ \mu\text{m}$.

The results above demonstrate that layer orientation has a significant impact on the wear resistance of 3D-printed ABS. Sliding parallel to the layer lines increases interlayer shear stress and promotes material delamination, resulting in higher wear. In contrast, sliding perpendicular to the layer lines often demonstrates better wear resistance because of reduced interlayer shear stress [41,43].

Overall, at the forefront, where the highest wear depth occurs, the FEM results align well with the experimental wear morphology. However, at the back end, the FEM predicts almost no wear loss, which significantly deviates from the experimental results. Three factors contribute to this discrepancy. First, while the FEM model incorporates the 3D-printed surface morphology, the roughness of the disc is excluded to avoid high computational costs and convergence

difficulties. Second, when measuring the wear profile, only regions near the desired area can be measured rather than the exact target region, which may introduce discrepancies. Third, wear debris significantly influences the wear process, as shown in Fig. 7. The debris from the aluminium disc becomes trapped and adheres to the 3D-printed pins. However, incorporating wear debris into simulations that couple solid mechanics and heat transfer is nearly impossible with current computational capabilities. These factors likely explain the discrepancies in wear distributions between the FEM results and experimental observations at the back end.

3.3. Wear volume

Fig. 10 (a) illustrates the wear volume progression over time for four angles: 0° , 45° , -45° , and 90° . The curves exhibit a clear linear increase

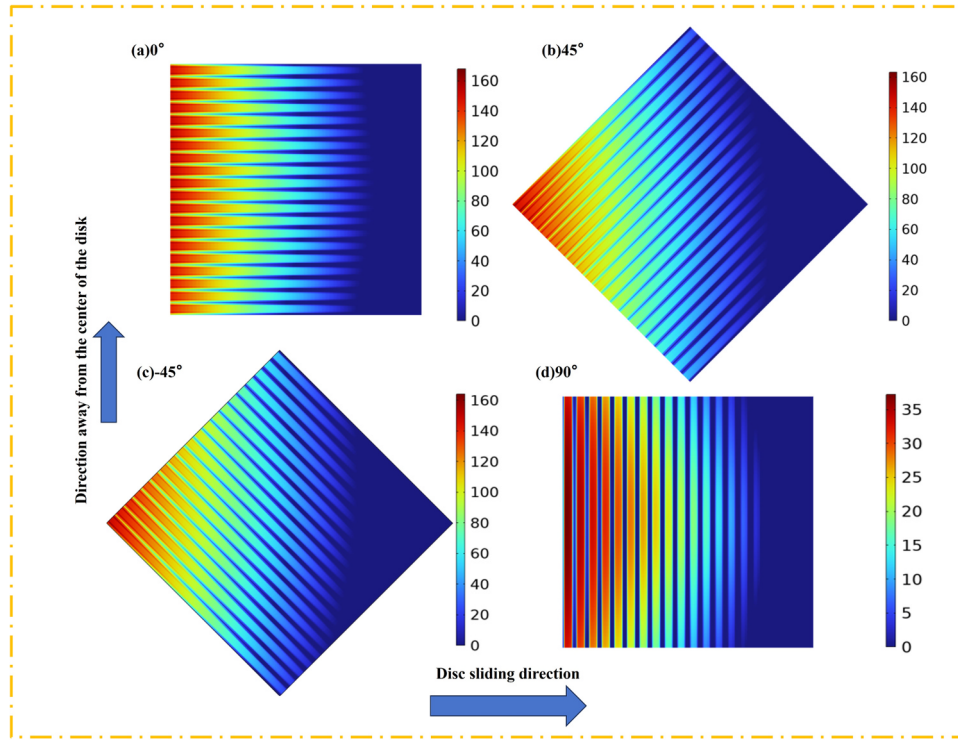


Fig. 8. Wear depth distribution from FEM simulation under 20N-0.628 m/s at 2 minutes.

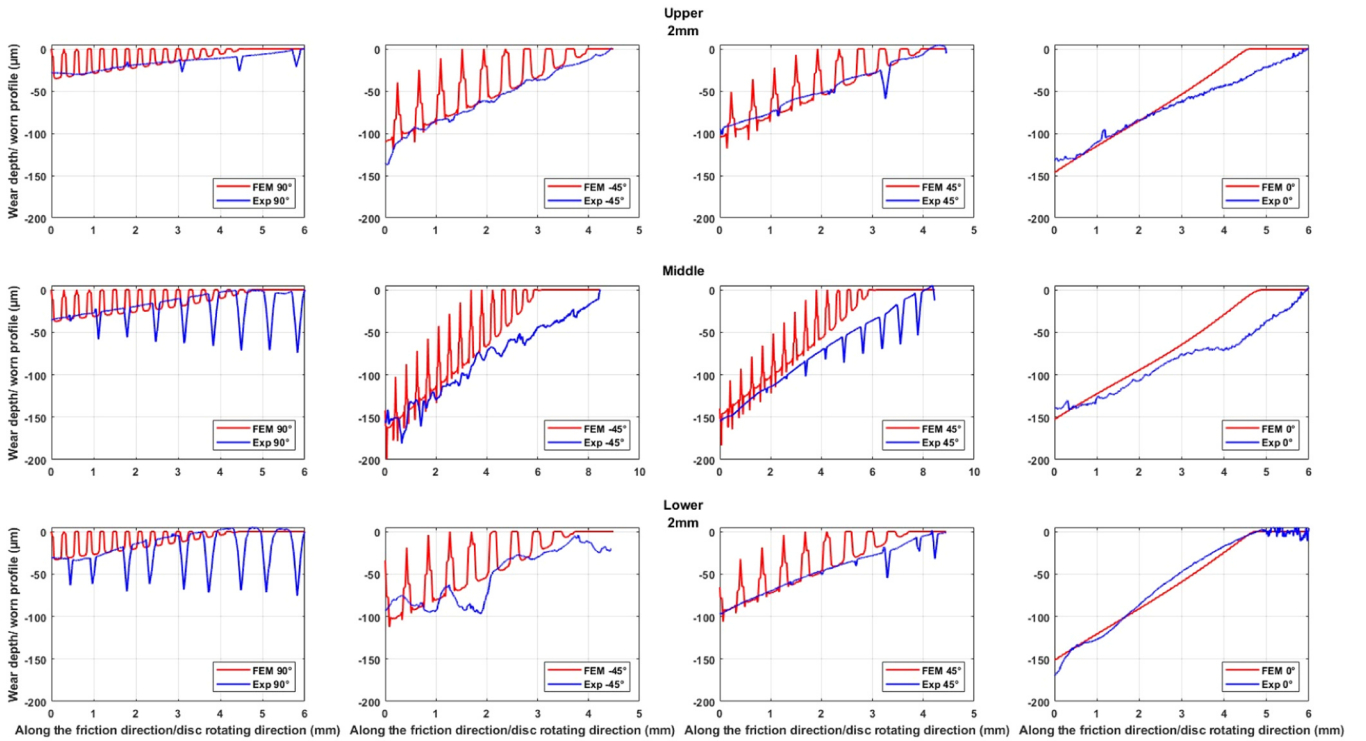


Fig. 9. Comparison between the experimental wear profile and the FEM wear depth distribution under 20N-0.628 m/s at 2 minutes.

in wear volume over time for all angles. The 0° configuration exhibits the highest wear volume, followed sequentially by -45° , 45° , and 90° .

Fig. 10 (b) compares the simulation results (blue bars) with the experimental mean values (orange bars) of wear volume at the 2-minute mark for the same angles. The experimental results are consistently lower than the simulation results, with minor deviations highlighted by

the percentage errors. The error percentages (0° : 6.91 %, 45° : 4.17 %, -45° : 5.08 %, 90° : 6.77 %) are relatively small, indicating that the simulation results closely match the experimental data. The close agreement between simulation and experimental results validates the FEM model. Minor discrepancies may arise due to factors such as material properties, environmental conditions, or simplifications in the

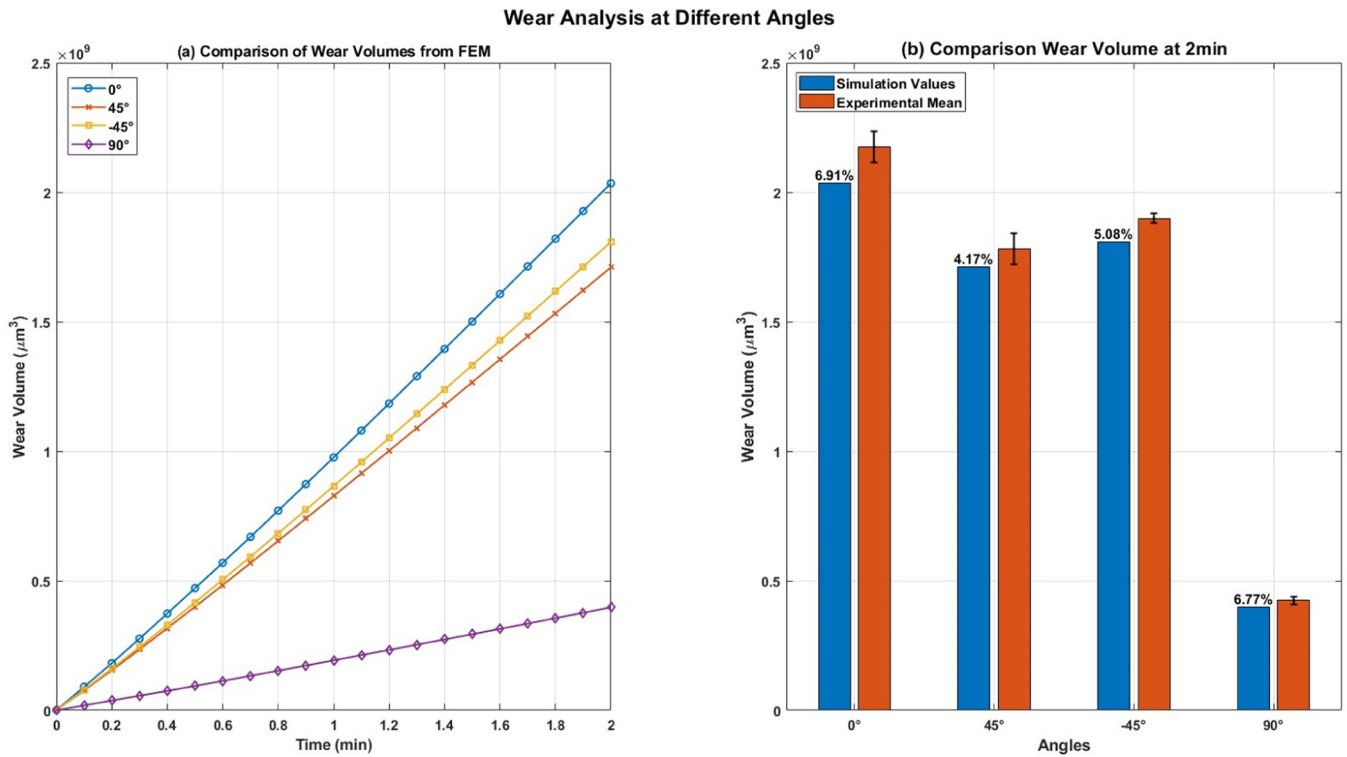


Fig. 10. (a)Wear volume development from FEM, (b) Comparison of wear volume between FEM and experiment with errors (%).

simulation.

The 0° configuration exhibits the highest wear volume in both simulation and experiment, indicating that this orientation is the most susceptible to wear. Wear volume decreases progressively for the other angles, with the 90° configuration exhibiting the least wear.

3.4. Wear mechanism

Fig. 11 presents the worn surface morphology of the samples under high resolution. As shown in Fig. 11 (b) and (d), abrasion was identified as the predominant wear mechanism for 3D-printed acrylonitrile butadiene styrene (ABS) tested at 45° and 90° orientations. Rough asperities

on the disc, in contact with the 3D-printed pins, caused deformation during sliding. As material removal commenced, grooves formed on the pin surface. In contrast, as shown in Fig. 11 (a) and (c), aluminium debris was found to adhere to the pin's surface. Two possible mechanisms may explain this adhesion. Initially, adhesive wear predominates at -45° and 0° orientations, where contacting surfaces form localized bonds at their asperities (microscopic surface irregularities) due to high pressure or friction, causing material to be transferred from one surface to another or detached as wear debris. However, if this were the case, aluminium debris would also be expected at 45° and 90° cases orientations, as the load and sliding conditions are identical. The second mechanism is more complex: at -45° and 0° orientations, aluminium

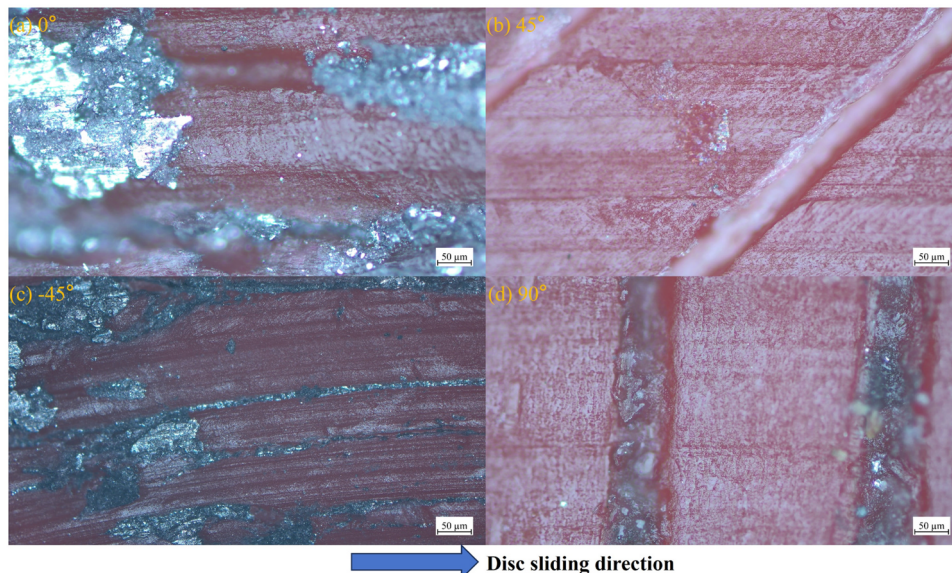


Fig. 11. Worn surface morphology under 20N-0.628 m/s observed by optical microscopy.

debris formed cannot be immediately ejected from the contact area and becomes trapped, being compressed into the pins. The aluminium debris adheres to the pin sliding against the aluminium disc, while in other regions, contact and wear between ABS and aluminium persist. This suggests that the orientation of layer lines relative to the sliding direction may influence debris ejection, thereby affecting the subsequent wear process; however, this requires further investigation.

To elucidate why different orientations exhibit varying wear resistance, the samples were sectioned as shown in Fig. 12 (a). The 90° orientation exhibits the fewest voids, whereas the 0° orientation displays the highest void content, with $\pm 45^\circ$ orientations intermediate. Voids between adjacent rasters represent gaps between extruded beads or layers, resulting from inadequate bonding, particularly at elevated printing speeds. Voids within infill represent micropores within a single bead, attributed to air entrapment or incomplete filling. Upon initiation of frictional sliding, these pre-existing voids serve as initiation sites for crack growth and propagation, thereby accelerating wear in individual layers [27]. Thus, internal defects in 3D-printed ABS significantly influence wear behaviour. All ABS samples in this study were printed using identical printing parameters, indicating that the total void content across samples is comparable. However, the spatial distribution of voids in samples is anisotropic, resulting in varying wear resistance under shear stresses induced by frictional forces, which can be leveraged in practical applications to minimize wear loss. Furthermore, void formation can be mitigated by reducing printing speeds, employing a $\pm 45^\circ$ raster angle, controlling build chamber temperature, and decreasing layer thickness. Readers interested in this topic are recommended to consult [27].

3.5. Noise signal and noise model validation

Fig. 13 illustrates the cumulative noise under different operating conditions. As the load and sliding speed increase from 10 N at 0.314 m/s to 15 N at 0.471 m/s, and further to 20 N at 0.628 m/s, the cumulative noise rises significantly. Under constant load and sliding speed, the cumulative sound pressure rises rapidly with increasing sliding distance but eventually stabilizes at a constant value. Furthermore, under the same load and sliding speed, the angle between the disc's sliding direction and the 3D print layer lines significantly influences the cumulative noise. Among the tested angles, -45° produces the highest cumulative noise, followed by 90° and 0°, with 90° slightly exceeding

0°. In contrast, the 45° angle results in the lowest cumulative noise. Fig. 14

This result is unexpected, as the cumulative noise trend differs from the COF trend shown in Fig. 6 and wear trend shown in Fig. 9. The orientation with a higher COF or wear rate does not necessarily exhibit a higher sound pressure level. This phenomenon may be associated with trapped wear debris during frictional sliding. As shown in Fig. 7, Fig. 15, and Fig. 16, the -45° orientation exhibits significantly more wear debris adhered to the surface compared to other orientations. Trapped debris acts as a third body, altering contact dynamics and inducing oscillations that generate noise [44,45]. Moreover, trapped debris induces stick-slip phenomena, characterized by abrupt transitions between sticking and sliding that generate noise [46]. As shown in Fig. 14, the -45° orientation exhibits significantly higher instability and a larger dynamic COF gap $\Delta\mu$ compared to other orientations, caused by trapped wear debris. Additionally, compacted debris forms micro-asperities that generate high-frequency noise through cyclical breaking and regeneration [47]. Therefore, the -45° orientation exhibits the highest sound pressure level.

The 90° orientation exhibits the second-highest sound pressure level due to the ridges and valleys of the layer lines interlocking with the disc surface, which amplifies vibration and noise during sliding. Moreover, the 90° orientation displays a greater $\Delta\mu$ than 0° and 45°, as shown in Fig. 14, resulting in stronger stick-slip phenomena that further intensify vibration and noise.

While 0° orientation exhibits lower $\Delta\mu$ than 90°, but the significant wear loss, indicating material transfer and the formation of adhesive junctions. The cyclic breaking of these adhesive junctions generates high-frequency noise [48]. In comparison, the 45° orientation exhibits medium wear, no trapped debris, lowest $\Delta\mu$, and smoother sliding, which explains its lowest noise level.

Fig. 17 presents a comparison of experimental and predicted noise pressure results, highlighting error bands. Each condition was tested three times. The constant A in Eq. (2) was determined by fitting it to the data from the first experiment. Then, this A value, along with the corresponding wear volume and COF from other two tests, was then applied to predict noise. The mean predicted noise was compared with the mean experimental noise.

Sound pressure exhibits a non-linear increase over time across all conditions. The growth rate of sound pressure depends on the applied force and velocity, with more pronounced increases observed at higher

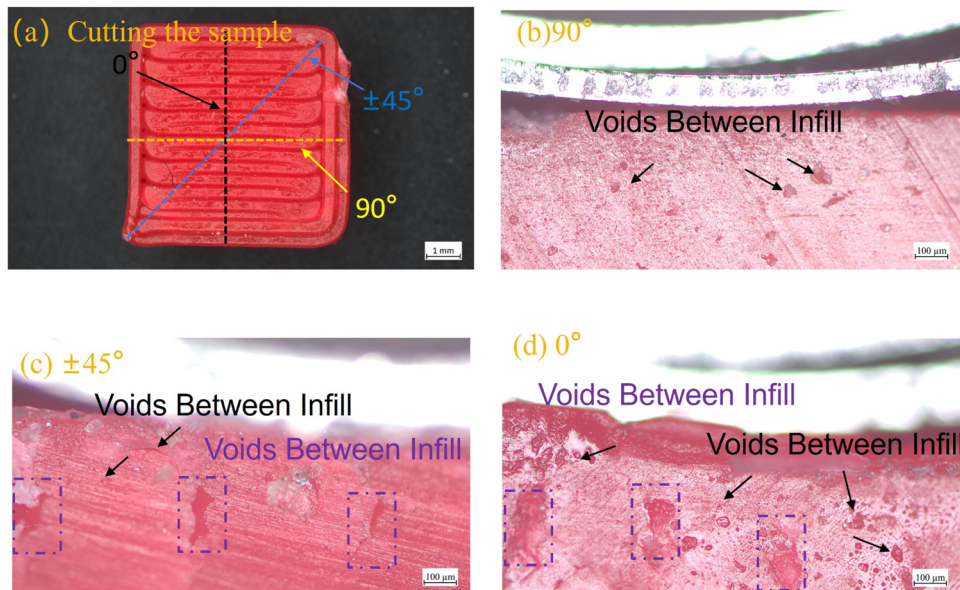


Fig. 12. Cross-section morphology observed by optical microscopy.

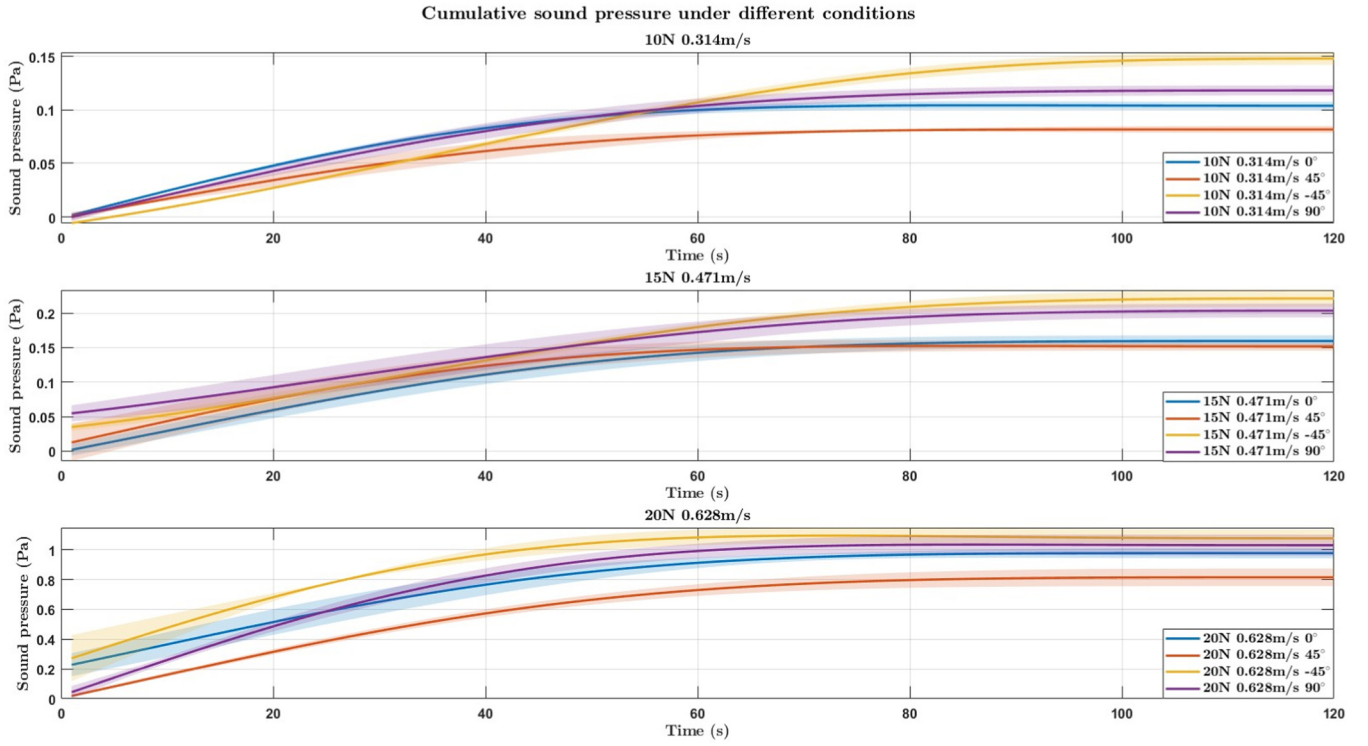


Fig. 13. Cumulative noise under different conditions.

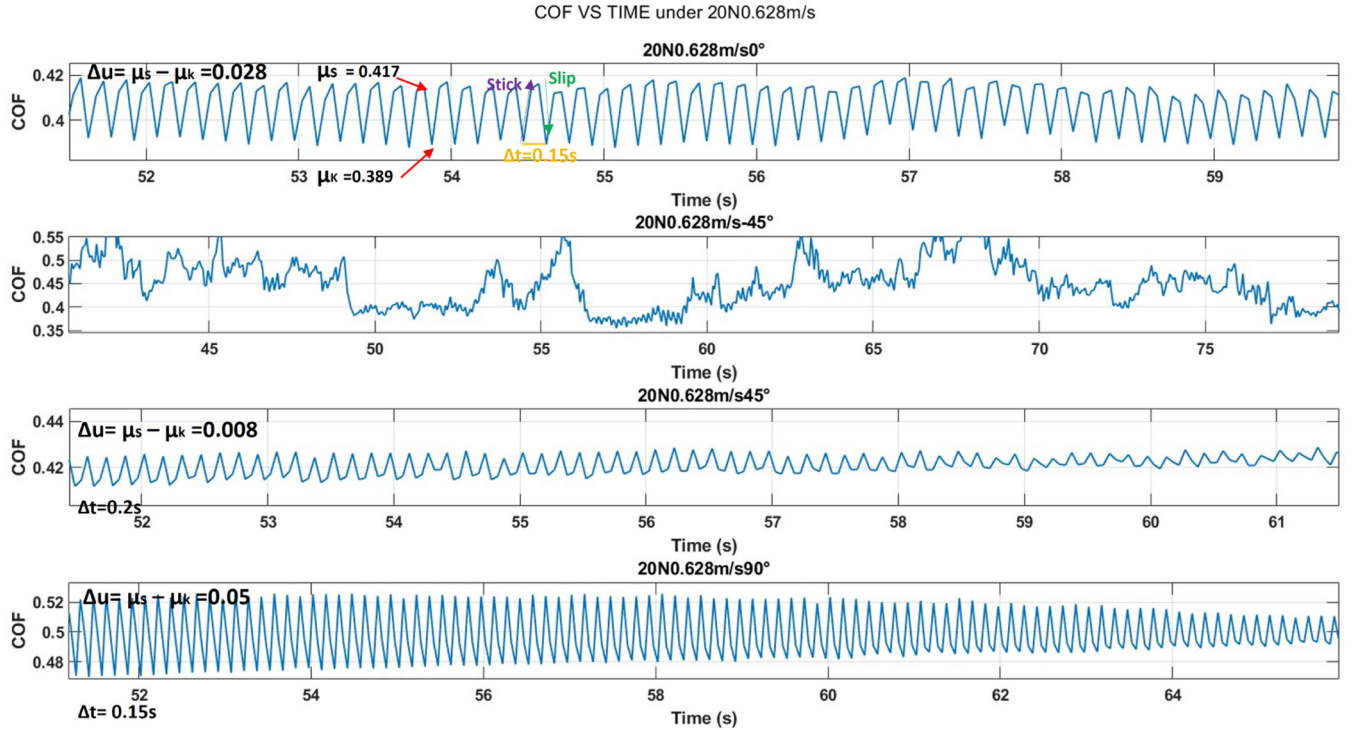


Fig. 14. Real time COF of different orientations under 20N0.628 m/s.

values of both. An increase in velocity (from 0.314 m/s to 0.628 m/s) and load (from 10 N to 20 N) results in a significant rise in sound pressure, highlighting the strong dependency of noise pressure on these factors.

During the running-in phase, a notable discrepancy exists between the experimental and predicted noise levels. This deviation arises

because certain noise-contributing factors are not incorporated into the current model. Firstly, the transition from static to kinetic friction is not accounted for. As the surfaces first come into contact, microscopic asperities interlock, necessitating a higher force to overcome initial resistance compared to sustaining movement [49]. The release of strain energy during this process contributes to acoustic emissions. As this

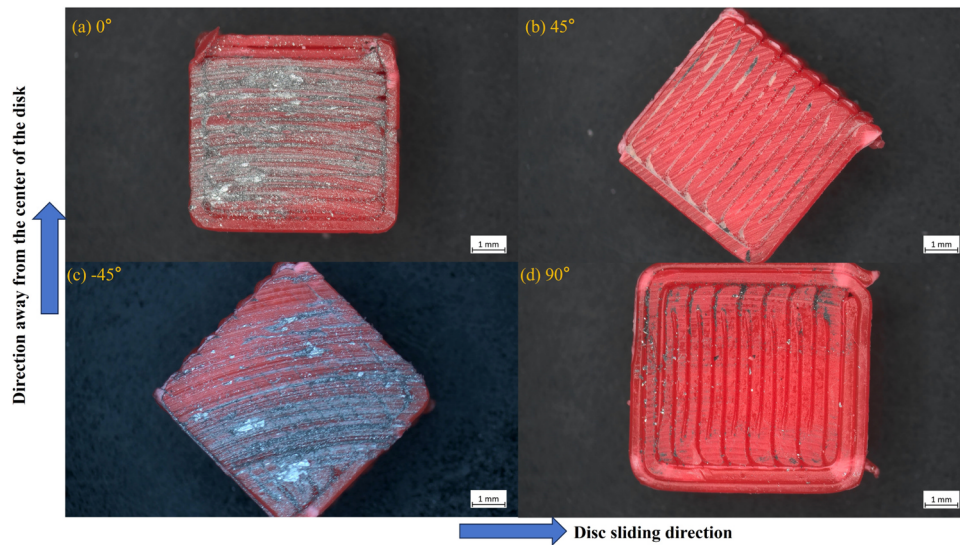


Fig. 15. Wear surface morphology observed using optical microscopy under 10N-0.314 m/s.

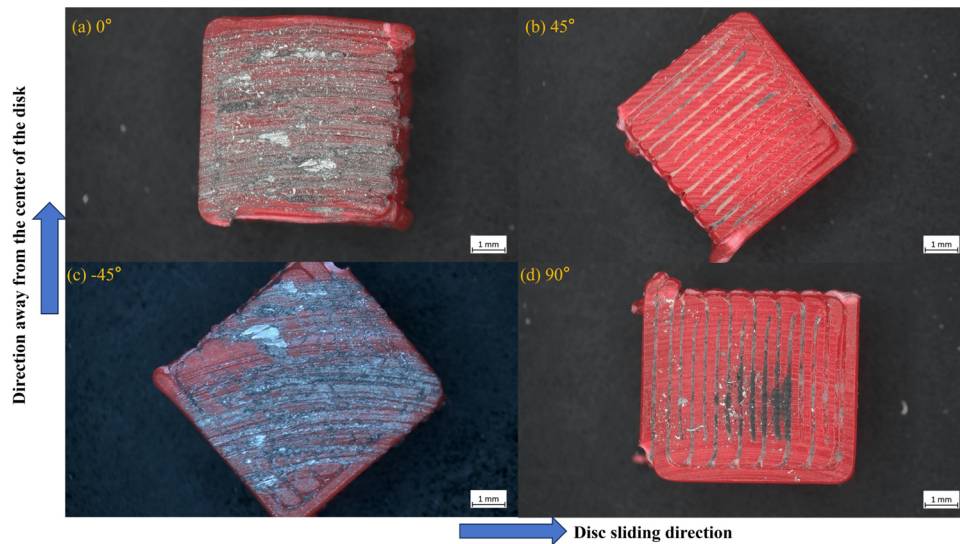


Fig. 16. Wear surface morphology observed using optical microscopy under 15N-0.471 m/s.

phenomenon is not incorporated into the current model (Eq. (2)), the symbolic regression method demonstrates significant discrepancies in relative error when evaluated against experimental noise data across various configurations at initial sliding stage. Moreover, the symbolic regression model considers only COF and wear loss during the frictional sliding process, excluding other contributors to noise, such as motor-generated noise from the tribometer and vibrations induced by contact dynamics, which also influence the overall noise characteristics and may play a significant role during the initial phase of sliding. As wear progresses, frictional noise from the pin-on-disc interface rapidly becomes the dominant component, thus reducing the relative error [28].

The coefficient A , labelled in each subplot, varies under different conditions:

Force Dependence: Higher forces lead to larger A values (e.g., $A=57,468.3547$ for 20 N at 0°), indicating increased noise generation at higher loads.

Velocity Dependence: As velocity increases, A values rise correspondingly, suggesting a proportional relationship between velocity and noise intensity.

Angle Effects: A values peak at 90° , reflecting directional sensitivity

in noise production.

3.6. Temperature

Fig. 18 illustrates the FEM-predicted temperature distribution of the disc and pin under a load of 20 N and a sliding speed of 0.628 m/s at 2 minutes. A comet-like temperature pattern along the sliding path has been observed, resulting from heat generated at the contact center as the pin moves across the disc, spreading radially [50]. This observation is consistent with previous studies [51,52]. For the pin, the front end exhibits significantly higher temperatures compared to the back end.

Fig. 19 compares the maximum temperature evolution of the disc between FEM simulations and experiments at four angles: 0° , 45° , -45° , and 90° . At all angles, both simulation and experimental temperatures increase over time, ranging from 293 K to 320 K, indicating a consistent warming trend. The simulation temperatures are generally higher than the experimental temperatures because the thermal gun measures only the exposed surface temperature, while the covered interface area remains unmeasured. Nevertheless, the relative error is minimal, staying below 4 % in all cases, demonstrating the accuracy of the simulations.

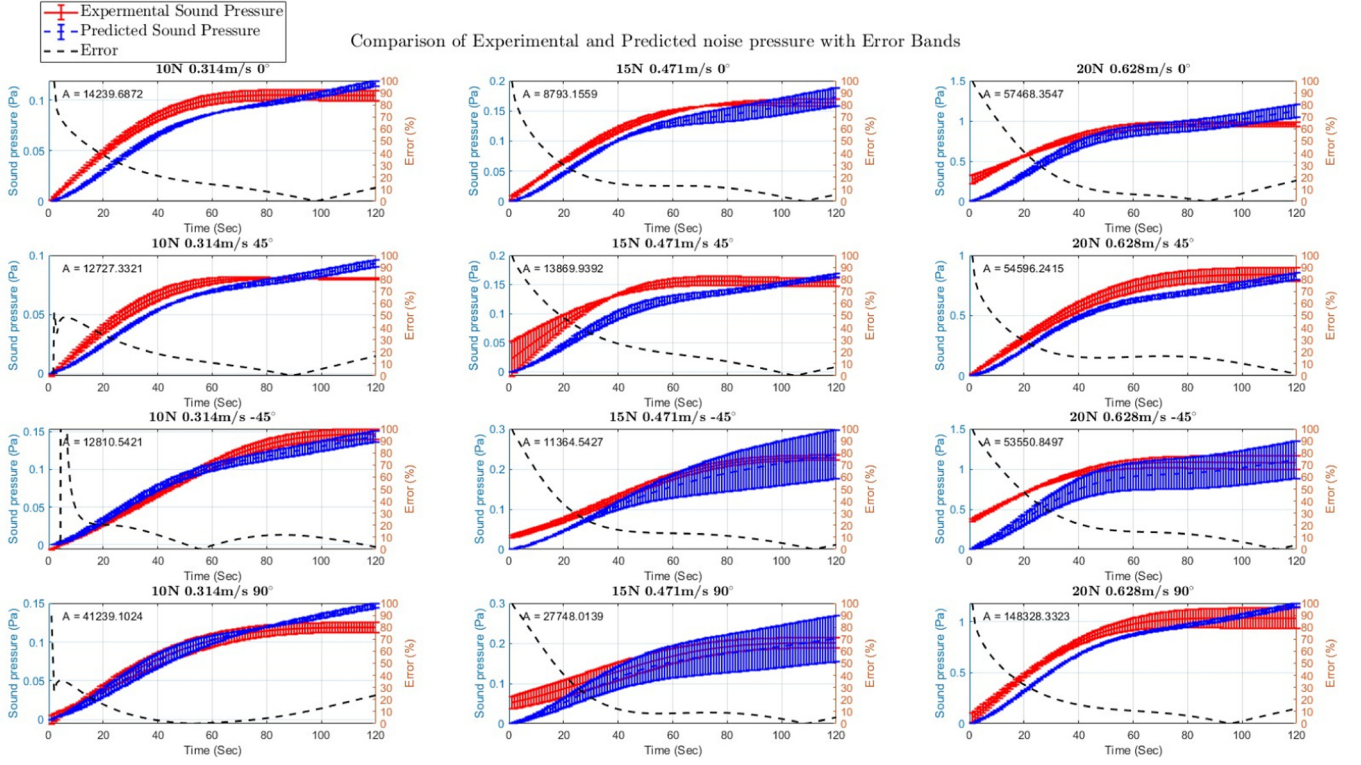


Fig. 17. Comparison between experimental and predicted noise.

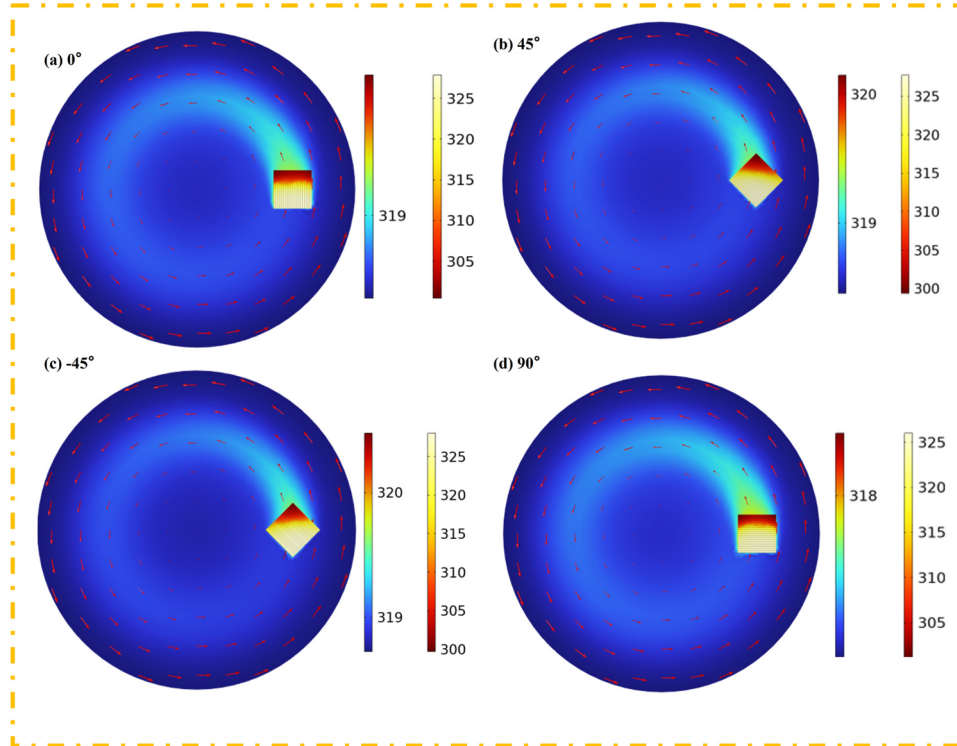


Fig. 18. Temperature distribution on disc and pin under 20N-0.628 m/s at 2 minutes unit: K (red arrows represent the disc rotating speed direction).

Compared to 0° and 90° , which reach a maximum temperature of 320 K, -45° and 45° show slightly lower maximum temperatures, ranging from 318 K to 319 K.

3.7. System stability analysis

The pin-on-disc tribological system is modelled as a lumped-parameter system with two degrees of freedom, as shown in Fig. 20. Notably, this dynamic model is not designed to replicate experimental

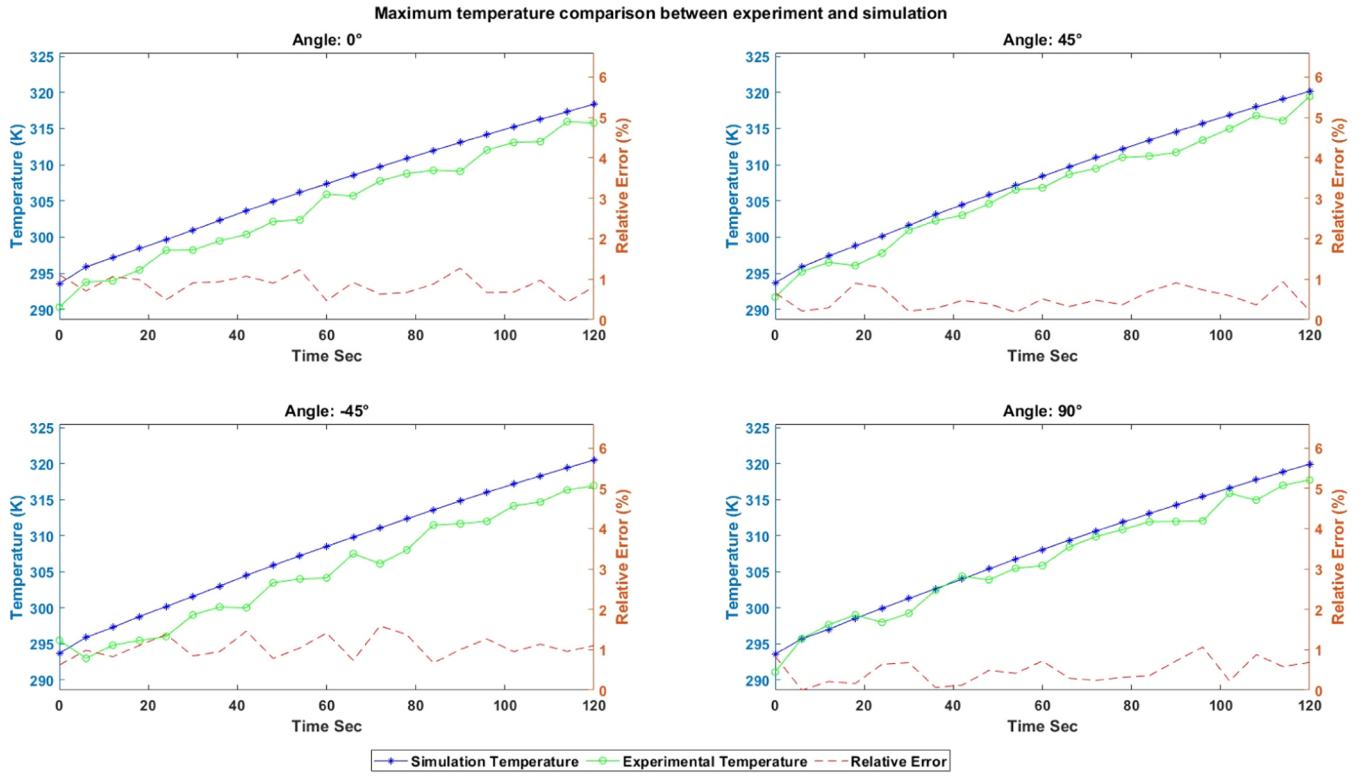


Fig. 19. Maximum temperature comparison between experiment and FEM simulation under 20N-0.628 m/s.

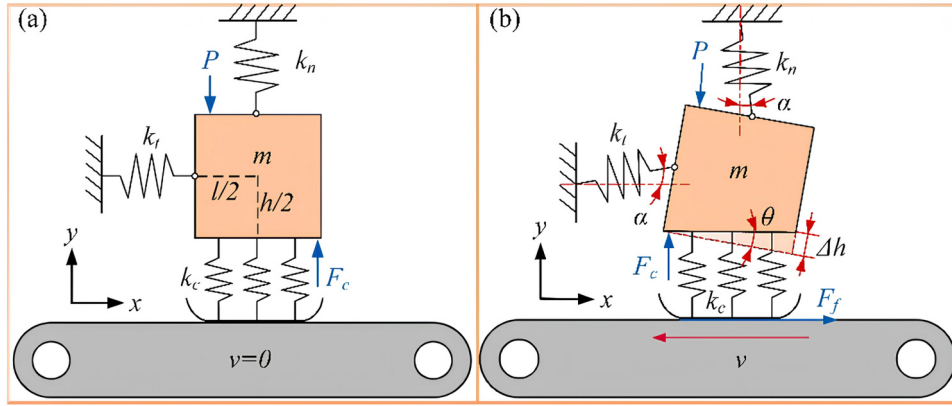


Fig. 20. Lumped-parameter system illustration: (a) the initial configuration and (b) the dynamic motion state of the tribological system (image retrieved from [53]).

outcomes but rather to enhance understanding of how wear depth (Δh) and the difference between static and kinetic friction coefficients ($\Delta\mu$) affect system stability and vibrational behavior. The model includes a friction block in contact with a moving belt with the velocity v . Uneven material loss creates an eccentric wear angle θ , described by Eq. (3), which changes contact dynamics and alters the contact stiffness k_c .

$$\tan \theta = \frac{\Delta h}{l} \# \quad (3)$$

where l is the tangential length of the friction pad.

The dynamic equations of the contact model, as shown in Fig. 20, are expressed as follows:

$$\begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix} \begin{bmatrix} \ddot{x} \\ \ddot{y} \end{bmatrix} + \begin{bmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix} = \begin{bmatrix} F_f \\ F_n \end{bmatrix} \# \quad (4)$$

where the stiffness matrix is:

$$\begin{aligned} K_{11} &= k_n \sin^2 \theta + k_t \cos^2 \theta \\ K_{12} &= K_{21} = k_n \sin \theta \cos \theta - k_t \sin \theta \cos \theta \# \\ K_{22} &= k_n \cos^2 \theta + k_t \sin^2 \theta \end{aligned} \quad (5)$$

The frictional force F_f is defined by Eq. (6).

$$F_f = -\mu F_c = -\mu k_c y \# \quad (6)$$

According to the Stribeck model, the coefficient of friction μ is a function of the relative velocity:

$$\mu = [\mu_k + (\mu_s - \mu_k) e^{-\tau |\nu_r|}] \text{sign}(\nu_r) = [\mu_k + \Delta\mu e^{-\tau |\nu_r|}] \text{sign}(\nu_r) \# \quad (7)$$

Here, ν_r denotes the relative velocity between the friction pairs, and τ represents the exponential decay between the kinetic and static COFs.

The support force F_n is defined as:

$$F_n = -P + F_c \# \quad (8)$$

The normal contact force F_c is defined as $F_c = k_c y$, P is the applied external load.

The contact stiffness k_c be expressed as follows [53,54]:

$$k_c = E(A_r/I)^\# \quad (9)$$

Here, E represents the elastic modulus, I indicates the plateau height on the pin surface, and A_r denotes the real contact area. Measuring the actual contact area and contact plateau height is challenging, but they can be computed using an FEM model, as shown in Fig. 21. A_r and I change due to wear development, suggesting a relationship between A_r , I , and Δh . If such a relationship is established, Δh can be used to represent the alteration of k_c , enabling an investigation into how Δh influences the lumped parameter system stability. The contact pressure distribution affects both the wear rate and the development of the contact area. As wear progresses, surface topography changes can cause the contact pressure to redistribute, resulting in non-linear variations in the contact area over time [55].

Fig. 22 and Fig. 23 present the relationship between A_r , I , and Δh . It can be seen that contact plateau height I linearly decreases with the Δh max wear depth (Eq. (10)), and contact area A_r is third power function of max wear depth Δh (Eq. (11)).

$$I = a\Delta h \quad (10)$$

$$A_r = a_1 \Delta h^3 \quad (11)$$

The stability condition of Eq. (4) is given by:

$$\begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix} \begin{bmatrix} \ddot{x} \\ \ddot{y} \end{bmatrix} + \begin{bmatrix} K_{11} & K_{12} + \mu k_c \\ K_{21} & K_{22} + k_c \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \quad (12)$$

By defining $Y_1 = x$, $Y_2 = y$, $Y_3 = \dot{x}$, and $Y_4 = \dot{y}$, the differential form of Eq. (12) is expressed as $\dot{Y} = JX$, where $X = \text{diag}(x, y, \dot{x}, \dot{y})^T$, the Jacobian matrix is written as:

$$J = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -\frac{K_{11}}{m} & -\frac{K_{12} + \mu k_c}{m} & 0 & 0 \\ -\frac{K_{21}}{m} & -\frac{K_{22} + k_c}{m} & 0 & 0 \end{bmatrix} \quad (13)$$

The Jacobian matrix J , as shown in Eq. (13), of the developed model displays asymmetry, potentially leading to vibrational instability in the system.

Based on complex mode theory, stability is maintained when the eigenvalue's real component does not exceed zero. However, if this value becomes positive, the system transitions into an unstable state.

The key parameters for assessing the stability of the 2-DOF system are assumed as follows: $m = 0.015$ kg, $k_n = 1000$ N/m, $k_t = 500$ N/m, $h = 0.15 \times 10^{-9}$ to 600×10^{-9} m, $l = 6 \times 10^{-3}$ m, $v_r = 0.628$ m/s, $P = 20$ N, $\mu_k = 0.4$, and exponential decay constant τ of the kinetic and static COFs is 0.2, gaps between the kinetic and static COFs $\Delta\mu = 0.002$ to 0.15.

Fig. 24 shows how the eigenvalues of the friction system change with wear depth Δh and the difference between the kinetic and static COFs $\Delta\mu$. According to Fig. 24 (a), even when $\Delta\mu$ is zero or near zero, the friction system still becomes unstable as wear develops. When $\Delta\mu$ is constant, as the wear depth increases, the system alternates between stable and unstable states. When Δh is constant, increasing $\Delta\mu$ causes the system to experience more unstable states during stick-slip transitions, while lower $\Delta\mu$ improves system stability.

As shown in Fig. 24 (b), $\Delta\mu$ has no effect on the vibration frequencies of the two system modes. In contrast, Δh significantly impacts the frequencies, which rise rapidly as Δh increases and eventually stabilize at a certain value. Thus, wear development not only causes system instability but also increases vibration frequencies. Meanwhile, a larger difference between the kinetic and static COFs leads to system instability

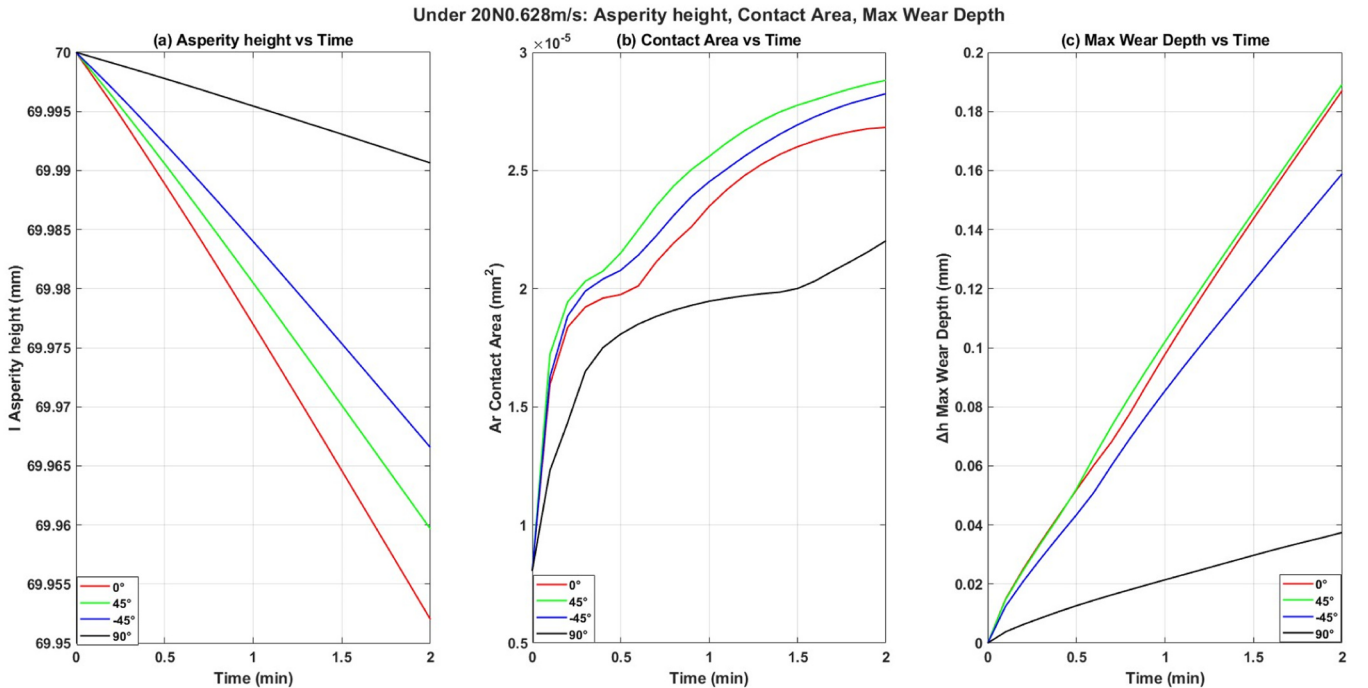


Fig. 21. (a) contact plateau height I variation, (b) A_r contact area variation, (c) Δh contact plateau height variation.

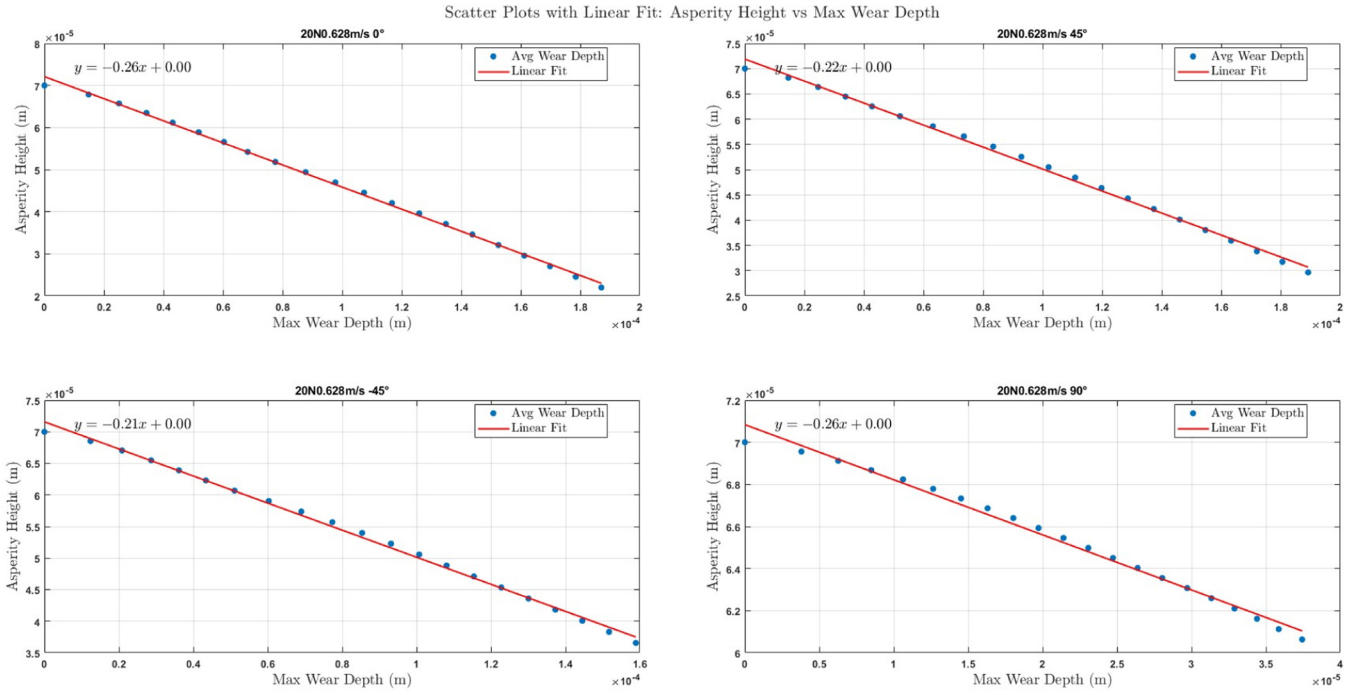


Fig. 22. Fitting the relationship between plateau height I and max wear depth Δh .

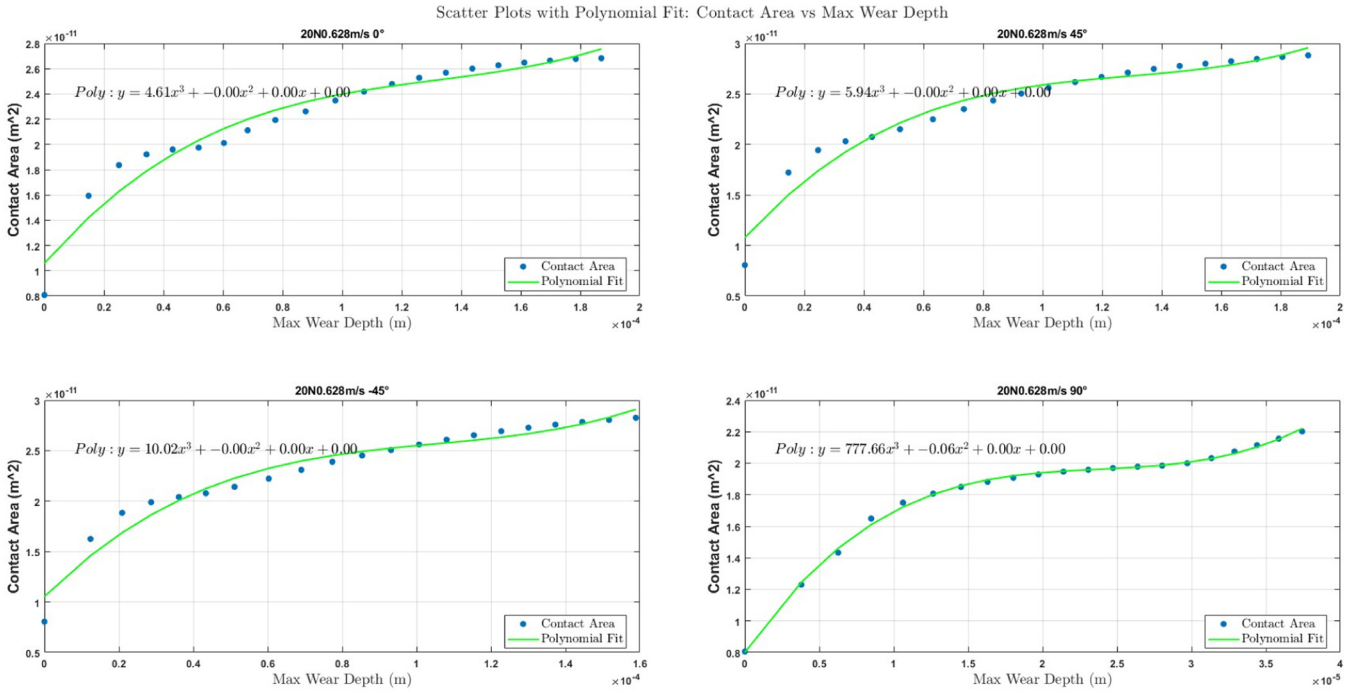


Fig. 23. Fitting the relationship between contact area A_r and max wear depth Δh .

but does not affect vibration frequencies.

4. Conclusion

This study provides a comprehensive investigation into the effects of the orientation between layer lines and relative sliding direction on the tribological performance, noise generation, thermal behaviour, and system stability of 3D-printed ABS materials in flat-flat sliding contact scenarios. Key findings from both experimental and FEM simulations highlight the complex interplay between orientation, friction, wear,

noise, temperature, and dynamic system behaviour under varying loads and sliding speeds.

Friction and Wear performance: Layer orientation significantly affects the coefficient of friction (COF) and wear performance. The 90° orientation shows the highest COF, caused by mechanical interlocking and ploughing effects as asperities obstruct the sliding motion, while also exhibiting the lowest wear rate due to reduced shear stress. In contrast, the 0° orientation has the lower COF but the highest wear rate. This is because the sliding direction aligns with the layer lines, minimizing asperity engagement and ploughing effects, which lowers sliding

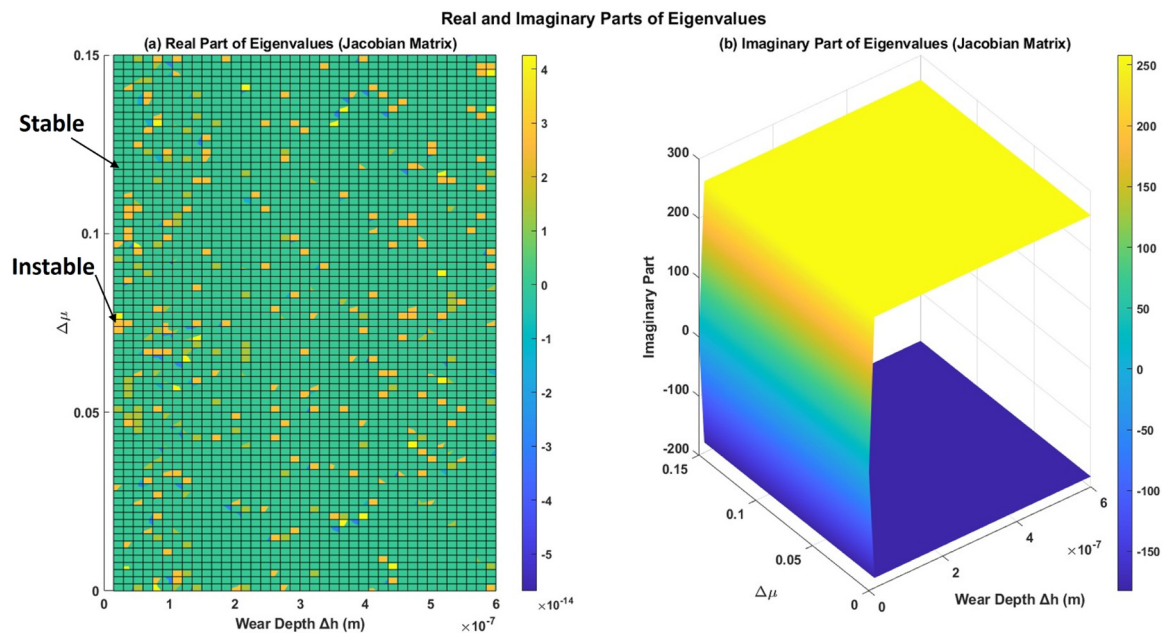


Fig. 24. (a) The real part and (b) the imaginary part of the eigenvalues of the friction system changes with wear depth Δh and the kinetic and static COFs $\Delta \mu$.

resistance but promotes material removal. However, this alignment accelerates material removal, causing greater material loss. The $\pm 45^\circ$ orientation strikes a balance, reducing both friction and wear through smoother diagonal sliding. This demonstrates its potential to lower energy dissipation and enhance tribological performance.

Noise Generation: The orientation angle between the sliding direction and 3D-printed layer lines significantly affects cumulative noise and stick-slip phenomena. The -45° orientation produces the highest cumulative noise due to trapped wear debris, which induces strong stick-slip behaviours and amplifies high-frequency vibrations. The 90° orientation exhibits the second-highest noise level, driven by mechanical interlocking of ridges and valleys, as well as a larger difference between kinetic and static COFs ($\Delta \mu$), which enhances vibration and noise. In contrast, the 45° orientation results in the lowest noise level, attributed to smoother sliding, minimal trapped debris, and the lowest $\Delta \mu$, showcasing its potential for reducing noise and improving tribological performance. After validation, the noise prediction model shows relatively high errors compared to experimental results, exceeding 15 % in some cases but remaining below 12 % in most instances.

Thermal Insights: The FEM simulations accurately predict the temperature distribution and evolution during sliding, with relative errors below 6 % compared to experimental measurements. The distinct comet-shaped temperature pattern on the disc, consistent with prior studies, highlights radial heat dissipation from the contact centre. Orientations of -45° and 45° exhibit slightly lower maximum temperatures (318–319 K) compared to 0° and 90° (320 K), suggesting that orientation angle insignificantly impact thermal behaviours under frictional sliding.

Dynamic stability analysis: The system stability analysis demonstrates that wear depth (Δh) and the difference between kinetic and static COFs ($\Delta \mu$) significantly influence the dynamics of the pin-on-disc tribological system. Increasing Δh causes the system to alternate between stable and unstable states, while larger $\Delta \mu$ makes the system more prone to instability during stick-slip transitions. Moreover, Δh affects vibration frequencies, causing them to rise and stabilize, whereas $\Delta \mu$ has no impact on frequency but plays a critical role in system stability. These findings highlight the intricate relationship between wear, friction, and stability, providing valuable insights for optimizing tribological system design and performance.

CRediT authorship contribution statement

He Feiyang: Methodology. **Zheng Bohao:** Data curation. **Khan Muhammad:** Supervision, Methodology, Conceptualization. **Tian Yang:** Writing – original draft, Software, Methodology, Investigation, Data curation, Conceptualization.

Statement of originality

We hereby declare that the manuscript titled “Influence of Layer Orientation on Wear, Noise, and Stability in 3D-Printed ABS Components” is an original work of the authors and meets the following conditions:

The research presented in this manuscript has not been previously published in part or whole in any format or language.

The manuscript is not currently under consideration for publication elsewhere, nor will it be submitted to any other journal or publication unless it is formally declined by Tribology International.

All data, text, and figures included in this manuscript have been generated by the authors or appropriately cited from published sources.

The manuscript does not contain any material that infringes upon the copyright or intellectual property of others.

The authors have adhered to ethical standards in conducting the research and reporting the results.

By submitting this manuscript, we affirm our commitment to maintaining scientific integrity and ensuring compliance with the policies of Tribology International.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

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