Wear Rate Reductions in Carbon Brushes, Conducting Current, and Sliding Against Wavy Copper Surfaces

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Abstract—Wear tests are presented in which a carbon brush, loaded by a constant force spring, conducts current and slides against smooth and wavy copper rotors. The wavy rotors possessed surface waves of tens to hundreds of microns. With brush current varying from 0 to 40 A, carbon brushes slid over the smooth and wavy rotors and wear rates (μg/s) were plotted versus rotor speed. Wear rates on the wavy rotor were generally less than wear rates on the smooth rotor, with and without current. Wear rates on the wavy rotor were considerably less than corresponding wear rates on the smooth rotor at certain rotor speeds.

Evidence suggests that wear rates were most reduced at those rotor speeds where surface waves on the wavy rotor passing beneath the brush caused the brush-stiffness-rotor system to resonate. Studies of contact voltage drop suggest that under these resonant conditions, the brush and rotor stayed connected. Also, no evidence of arcing or micro-arcing was found on the copper track. This study shows appreciable reductions (up to 50%) in wear rate possible on brush rotor systems by prescribing tiny surface waves on the rotor and running the rotor at speeds such that the surface waves induce microvibrations and resonance.

I. INTRODUCTION

When a brush slides over a mating surface, surface waviness/roughness in conjunction with friction can force the brush up and down, dynamically influencing instantaneous contact force, contact tractions, contact area, and surface films. This can influence friction and wear [1], [2].

Adhesive wear with low to moderate wear rates usually predominates at lower sliding speeds and forces; opposing asperities bond together and later break at different spots, generating tiny wear particles. If conditions intensify, more severe wear can be activated. With abrasive wear, hard particles from earlier wear events and/or the environment become entrapped within the sliding interface and cut or plow the surfaces. Large loads at high speeds may induce thermal mounds [3], [4] wherein heat generated by friction and electric currents (if present) elevate local temperatures and bulge material near the contact spots. Faster growing bulges separate less dominant neighbors, transferring and further concentrating loads. Temperatures, stresses, and thermal expansions increase, promoting loss of large particles. Thermal mounds have been found on brushes [5]–[7], brakes [8], [9], and seals [10], [11]. Simulations suggest that thermal cycling induces thermal mound wear [13]–[15]. Once initiated, thermal mounds form and detach within about 4–100 ms [5], [7]. On average, thermal mounds were observed on the face of a brush once every 25–30 s [7].

Surface undulations in the form of narrow grooves [16] or small pits cut into a rotor can appreciably reduce friction, contact resistance, and wear [17], [18]. Hard entrapped particles fall into depressions, reducing abrasive wear, and contact areas shift to other sites, disrupting the thermal cycles that form thermal mounds, as the slider passes over.

Vibrations can increase or decrease friction and wear, depending on conditions. Several studies on various material pairs noticed a reduction in friction induced by vibrations generated either externally or by surface roughness and sliding [19]–[24]. Studies of interaction between friction, wear, and system rigidity showed that changes in wear depended on normal load and sliding speed [25]–[28]: increased wear rates resulted from increases in the frequency of the applied load which enhanced surface fatigue.

Usually, contact vibrations increase brush wear [29]. Severe vibrations encourage arcing of brushes [30] and impact wear in sliders [31]. In machining, tool wear is accelerated and surface finish marred [32], [33].

Contact vibrations can also reduce wear. Ultrasonic vibrations applied normal to a slider drastically reduced steel pin on steel disk wear rates [34]. Others [35] found that vibrations sometimes increased and sometimes decreased wear rates, depending on the materials. In single point diamond turning of steel [36], ultrasonic vibrations reduced tool wear without harming surface finish; ultrasonic vibrations also extended carbide tool life ten to twenty-fold in the machining of glass [37]. Santini and Kennedy [9] realized 30% and 44% reductions in surface temperature and wear rate without a significant change in friction, after they cut a slot into a copper-based brake pad and ran it against a brake disk. They produced similar results by pressing the pad against the disk with a more compliant spring.

For carbon samples sliding against wavy copper surfaces without conducting current, Bryant and Lin [38] reported reductions in wear rate (in μg/s) compared to wear rates for sliding against smooth surfaces. Wear rates for the smooth and wavy rotors were identical at some speeds, but at certain
profiles of wavy rotor a. The surface height $Z \leq 100 \mu m$ is exaggerated; the highest point (100 $\mu m$) is the coordinate reference. The circumferential distance from the reference point $X \leq 88$ cm, and the radial distance from the inner ring $Y \equiv (r - r_{inner}) \leq 2.54$ cm. In Fig. 2, peaks and valleys form ridges that zigzag across the sliding track (profile width) with radial position $Y$ not constant; i.e., $Y = Y(X)$. Similar profiles for wavy rotor $b$ can be found in [7].

Probe measurements of axial (perpendicular) motions targeted a metal screw attached to the brush (see Fig. 1). At slow speeds (quasi-motionless) axial motions mirror the overall waviness profile, producing a $Z$ versus $X$ trace roughly related to the maximum heights across the track width $Y$. Fourier analysis produced the spectrum of displacement amplitudes (in $\mu m$) versus mode number shown in Fig. 3; in these figures the first mode is due to rotor tilt while the second mode is a flatness error caused by out of plane bowing. Higher modes are surface features.

In Fig. 3(a), the wavy polycarbonate-backed surface used in the $a$ experiments, shows modes 5–17 enhanced compared to Fig. 3(c), the smooth steel-backed surface used in the $a$ experiments. Here, mode 3 is very large, mode 5 large, with slightly smaller modes 4, 6, and 11 about the same size; modes 7–10 and 12–13 are yet smaller; and after 13 only mode 17 is appreciable. Wavy polycarbonate-backed surface $b$ in Fig. 3(b) (used in the $b$ experiments) shows modes 9–16 enhanced compared to Fig. 3(c): the relative heights of modes 3 and 4 are about the same; modes 5–7, 10, and 11 are slightly smaller than 3 and 4, with 6 and 10 the tallest of this group; modes 8 and 9 are yet smaller, and after 11 only mode 16 is appreciable. The smooth rotor used in the $b$ experiments was similar to the smooth $a$ rotor.

C. Brush Holder, Load Spring, and Holder Support Arm

A constant force spring (weight 0.039 kg) affixed to a brush holder pressed the brush against the rotor along the top of the trailing edge (see Fig. 1). To maintain a constant force over a large displacement range, constant force spring stiffnesses were weak. Spring loads were constant: against rotors $a$ the force was 24.4 N, and against rotors $b$ the force was 20 N.
For the a experiments, anodized aluminum plates insulated the brush holder and a piece of paper separated and insulated the load spring from the brush. Resistance between the brush and machine frame measured about 30 MΩ. For the b experiments, the insulation was not effective and the resistance was about 0.6 Ω. Clearances between the brush sides and inner surfaces of the holder were less than a millimeter.

The holder support arm was a 37.5-cm long beam (2.54 × 2.54 cm cross section) cantilevered to the machine frame. For experiments a, the lowest resonant beam frequency was about 110 Hz; for experiments b, the lowest resonant beam frequency was about 170 Hz. This frequency changed from a to b experiments because masses (brush and spring) and stiffnesses were different.

D. Brushes

Brushes were NECC brush grade 634, with material properties listed in Table I and geometries in Table II. Brushes with slider geometries B and C (Table II) possessed different face dimensions to include nominal contact geometry as a test parameter; mass and other dimensions were similar. All brushes weighed about 0.085–0.092 kg. Millimeter level clearances between the brush and holder allowed normal and small lateral movements of the brush within the holder.

III. MEASUREMENTS

Measured were wear rate, contact connect/disconnect, rotor speed, brush displacements (via capacitance gauge) perpendicular to the rotor, contact forces (via strain gauges mounted on the holder arm), brush current, and temperature (via embedded thermal couples). Finally, dynamic displacements were Fourier-analyzed for spectral content.

A. Wear Rate

Wear rate \( WR = \frac{(W_i - W_f)}{\Delta t} \) was measured as the weight lost while sliding during time \( \Delta t \). The nonstandard wear rate measure (ug/s) avoided counting revolutions to record distance slid, and simplified comparing wear of brushes with different brush face surface areas. Initial and final brush weights \( W_i \) and \( W_f \) were measured with 0.1 mg resolution. A test ended when 10–25 mg had worn away\(^1\). Tests usually lasted 24 h. Before weighing, the brush was allowed to stabilize to ambient conditions for at least 20 min.

B. Contact Connect/Disconnect

In the a experiments, currents flowed from the sample to the return brush (see Fig. 1) through the path: power supply–test

\(^1\)On a 0.1 mg resolution scale, this produced a repeatable and reliable measurement.
TABLE III
ROTOR GEOMETRIES AND LOADING CONDITIONS USED IN THE EXPERIMENTS

<table>
<thead>
<tr>
<th>exp</th>
<th>prominent waviness modes</th>
<th>spring force (N)</th>
<th>beam natural frequency (Hz)</th>
<th>friction coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>11, 17</td>
<td>24.4</td>
<td>110 - 160</td>
<td>0.37</td>
</tr>
<tr>
<td>b</td>
<td>9, 16</td>
<td>20</td>
<td>150 - 175</td>
<td>0.21</td>
</tr>
</tbody>
</table>

brush–slip ring–rotor–shaft–return brush. Static and dynamic contact resistance through this circuit measured about 0.1 Ω. The resistance between the test brush and the machine frame was about 30 Ω.

In the b experiments, currents flowed through the 0.1-Ω circuit: power supply—test brush–slip ring–rotor–shaft–return brush path, and also through an inadvertent shunt: brush–holder arm–machine frame–bearings–shaft–return brush. Static and dynamic contact resistance through the shunt measured about 0.6 Ω.

For the a experiments, the brush current waveform and the contact voltage drop were displayed on 100 MHz or 5 GHz bandwidth oscilloscopes to check for instantaneous disconnects of the brush from the rotor. An abrupt change (low to high) in the disconnect signal indicated a contact disconnect. During tests without current, the power supply was (sometimes) a 4 V dc source in series with a 220Ω resistor. Voltage drop across the brush–slip ring–rotor–shaft–return brush circuit was of ten millivolt order (low) when the slider contacted the rotor. When disconnected, the voltage drop approached the 4 V source (high).

C. Friction Coefficient

The coefficient of friction was calculated as the ratio of the tangential force at the beam root (measured by strain gauges) to the spring force, which was assumed constant and equal to the spring load. Beam vibrations affected strain gauge measurements of the normal force; consideration of free body diagrams associated with the holder arm–spring–brush mass system suggest that the normal force measured at the beam root is not representative of the normal contact force: large inertial forces generated by beam vibrations and brush mass sum with the normal contact force transmitted down the beam to contaminate readings. The constant force spring, with weak stiffness and poor vibration transmissibility, is between the brush and beam. Only tiny vibrations of the brush mass can contaminate the readings.

IV. RESULTS

A. Procedure

Brushes slid against the smooth and wavy rotors at 500–1500 rpm with increments usually about 100–250 rpm.

Two sets of experiments, a and b were conducted, each with different conditions and rotors; Table III summarizes the results. Set a involved smooth and wavy rotors a, while set b involved smooth and wavy rotors b. At the beginning of each experiment, the normal force acting on the brush was adjusted with negligible tangential force to 24 N (a) or 20 N (b). Each test ran for at least 20–24 h. The coefficient of friction was measured over a short interval for every test. Experiments were run with dc currents of 0, 10, 20, and 40 A. The test brush was the anode.

B. Wear Studies

1) Wavy Rotor/Smooth Rotor Comparison without Current: Wear rates $W_R_{aw}$ for sliding against the wavy polycarbonate rotors a (Fig. 3(a)) and b (Fig. 3(b)) and $W_R_s$ for sliding against the smooth steel-backed (Fig. 3(c)) copper surfaces versus speed are plotted in Fig. 4. No currents were conducted. Solid lines denote the a experiments, and dashed lines pertain to the b experiments. The brushes possessed geometries B and C in Table II. Shaded symbols pertain to $W_R_{aw}$, open symbols to $W_R_{aw}$. Markings or superscripts a and b denote wavy or smooth rotors a and b with experimental conditions a and b. All data points were measured at least twice (some several times) to verify curve shape and repeatability. Plotted data were derived from the longest duration tests (typically 20–24 h). Shorter duration tests, not reported here, demonstrated repeatability.

All curves tend to increase with speed. Excepting the tail of the $a W_R_s$ curve, the $W_R_s$ curves monotonically increase over the entire speed range. Below 1000 rpm the $W_R_s$ curves are nearly linear with moderate slope; above 1000 rpm slopes steepen. The $W_R_{aw}$ curve, always less than or equal to the corresponding $W_{R_s}$ curve, increases and decreases with speed.

2) Wear Rates with Current: Fig. 5 contains wear rate versus speed for brushes conducting 10, 20, and 40 A and sliding against the smooth (curves with solid symbols) and wavy rotors (curves with hollow symbols). Here slider geometry B was used.

The wear rates for the wavy rotors a and b were generally lower than the wear rates for the corresponding smooth rotors a and b. For a current of 10 A, experiment a curves showed appreciable drop in $W_R_{aw}$; experiment b was not conducted with 10 A. For currents of 20 and 40 A, experiment b curves showed a appreciable drop in $W_R_{aw}$. In experiment a for $W_{R_s}$ was lower than $W_R_{aw}$ for higher speeds while at 40 A at almost all speeds $W_R_{aw}$ was lower than $W_R_{aw}$.

3) Slider Motions (Vibrations) Normal to Rotor Surface: Fig. 6 is an amplitude (μm) spectra versus frequency (Hz) of displacements normal to the sliding surface, derived from sliding against wavy rotor a at 1000 rpm. In the figure the
absissa ranges from 0 to 400 Hz, and the first twenty or so peaks—equally spaced on the abscissa axis—are prevalent. Peaks represent harmonics of vibrations activated by sliding. The fundamental near 17 Hz is consistent with the rotor speed of 1000 rpm activating the first waviness mode.

Peaks at higher frequency should be induced in part by surface waves of higher modes passing beneath the slider. Fig. 6 is a spectrum of dynamic motions of the brush on the wavy rotor $a$. Fig. 6 exhibits a relative ordering of peak heights qualitatively similar to Fig. 3(a), except that in Fig. 6 peaks 8 and 9 have grown relative to their neighbors (compare these peaks to their respective counterparts in Fig. 3(a)). Peaks 7–10 fall within a band extending from 110 to 160 Hz; this frequency band is close to the first resonant frequency of the brush mass–spring–beam holder arm system $a$.

Relative amplification of peaks for modes 7 and 8 in Fig. 6 suggests a resonant frequency near this range (100–120 Hz). Peaks 7 and 8 have grown relative to peak 9 in Fig. 6 (compared to the same peaks in Fig. 3(a)).

4) Measurement of Slider Forces and Friction: Measured normal and tangential forces yielded frictional coefficients independent of speed with rapid fluctuations about 0.37 for the $a$ experiments, and 0.21 for the $b$ experiments. Fig. 7 shows traces taken at 1000 rpm while running against the wavy rotor $a$. Fluctuations at 1000 rpm range from 0.29 to 0.57. Traces on the smooth rotors were similar to traces on the wavy rotors.

C. Circuit Continuity

1) No Current: The 4 V supply in series with the 220-Ω resistor induced currents of tens of milliamperes when applied across the aforementioned Fig. 1 circuit (see Section III.B) beginning with the test brush and ending with the return brush.

Oscilloscope traces of this contact disconnect signal (voltage drop across the circuit from test brush to return brush) taken at 750, 1000, and 1525 rpm revealed small spikes at intervals of about 800 ps. A typical spike was about 25 mV, the highest was less than 50 mV.

2) Slider Surface Inspection: After all runs with current, the copper sliding track of the wavy rotor was carefully inspected. No evidence of arcing or micro-arching (melt zones, pits, craters, etc.) was observed.

V. DISCUSSION

Figs. 4 and 5 imply that a tiny amount of surface waviness (usually 10–100, and up to 300 μm) on a rotor may reduce brush wear. This was observed at all speeds and currents for $b$ experiments, and for most speeds and currents for the $a$ experiments.

For brushes sliding against wavy copper surfaces without current, Bryant and Lin [38] reported reductions in wear rate and hypothesized that resonance-induced microvibrations changed interfacial physics. Fig. 8 from [38] contains the $b$ curves of Fig. 4 and the fractional difference $FD = (WR_s - WR_0)/WR_0$ (dashed curve). A local maximum in $FD$ corresponds to a local minimum in $WR_s$ relative to $WR_0$. At 600 and 1000 rpm where $FD$ maximizes, products of speed and prevalent mode number (10 and 16, see Fig. 3(b)) on wavy rotor $b$ are related: 600 rpm × 16 ≈ 1000 rpm × 10. These products in mode number–revolutions per second (160 Hz and 167 Hz, respectively) are close to the resonant frequency of the brush holder support arm (150–175 Hz) for setup $b$. 
The microvibrations seem to reduce wear, but not by disconnecting the brush from the rotor. The resonant frequency of the holder arm beam for the $a$ set of experiments was approximately 110–130 Hz. If the brush/rotor contact disconnect voltages (discussed in Section III.B) were caused by rotor waviness-induced beam resonance, disconnects would have appeared at time periods dictated by the inverse of the beam resonant frequency, roughly every 10 ms. The observed small disconnects appeared about every 800 ps, suggesting no relationship to the resonant beam vibrations. In addition, a complete disconnect would demand a disconnect voltage over 200 times the largest of the observed disconnect voltages.

Fig. 8 suggests that the waviness retarded severe wear. At about 1000 rpm the slope of the smooth rotor curve $WR_o$ abruptly steepens (more than triples), indicating a more severe wear mechanism [1], [2] such as thermal mounding or abrasive wear [1]–[4]. In general, $WR_o < WR_a$. At higher speeds, wear rates $WR_o$ on the wavy rotor approaches the dotted line, which extrapolates the low slope portion of $WR_o$ to higher speeds. Since $WR_o$ follows the low speed slope, low speed wear mechanisms are likely to be operative on the wavy rotor at higher speeds, whereas on the smooth rotor, more severe wear mechanisms reign.

A small amount of surface waviness (Fig. 3(a) and (b)) seems to discourage severe wear and limit moderate wear. Without disconnecting the contact, perpendicular vibrations could limit wear only by modulating the contact forces. Rocking vibration modes coupled to the perpendicular modes could disrupt interfacial physics without disconnecting the contact by rapidly transferring the contact to new locations within the slider–rotor interface, as the rocking causes the interface to touch at new sites. This could disrupt thermal evolutions needed to form thermal mounds and/or open clearances in the contact interface, allowing abrasive particles to escape.

Wavy rotor $b$ uniformly yielded lower wear rates than wavy rotor $a$. Differences between the two systems include the following: higher contact forces for the $a$ (24.4 N) experiments than for the $b$ (20 N) experiments; different resonant frequen-
cies (about 110 and 160 Hz); different waviness profiles (wavy rotor $b$ had only two prominent higher modes, whereas wavy rotor $a$ had several); and for experiments with current, the presence of an inadvertent shunt in $b$. Although the shunt likely influenced wear whenever rotor $b$ conducted current, it is interesting that wear rates on the $b$ wavy rotor were still generally lower than wear rates on the $b$ smooth rotor.

Finally, estimates of aerodynamic lifting forces between slider and rotor [30] were very small. Self-induced chatter vibrations [30] with kilo-frequencies are far higher than the frequency range 100–175 Hz. These factors are probably unrelated to the wear rate reductions.

VI. SUMMARY AND CONCLUSION

This study suggests that in brush applications it might be possible to prescribe a small amount of surface waviness onto the rotor surface (to excite microvibrations) that reduces wear rates without compromising device functionality. The following can be concluded:

1) Brushes sliding against rotors having different surface waviness amplitudes can exhibit appreciably different wear rates. The wear rate of the brush–rotor sliding system can be minimized by prescribing a small amount of surface waviness of a certain mode on the rotor, and then sliding the brush over the rotor at speeds such that the frequency at which a certain waviness mode passes beneath the brush is close to the brush–spring–rotor resonant frequency.

2) Surface waves on the rotor can excite brush vibrations and reduce the wear rate without disconnecting the contact circuit.

3) For carbon brushes acting as anodes, appreciable reductions in wear rate appear with and without current.

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REFERENCES


