Frictional Anisotropy in Boundary Regime: Effect of Surface Texture*

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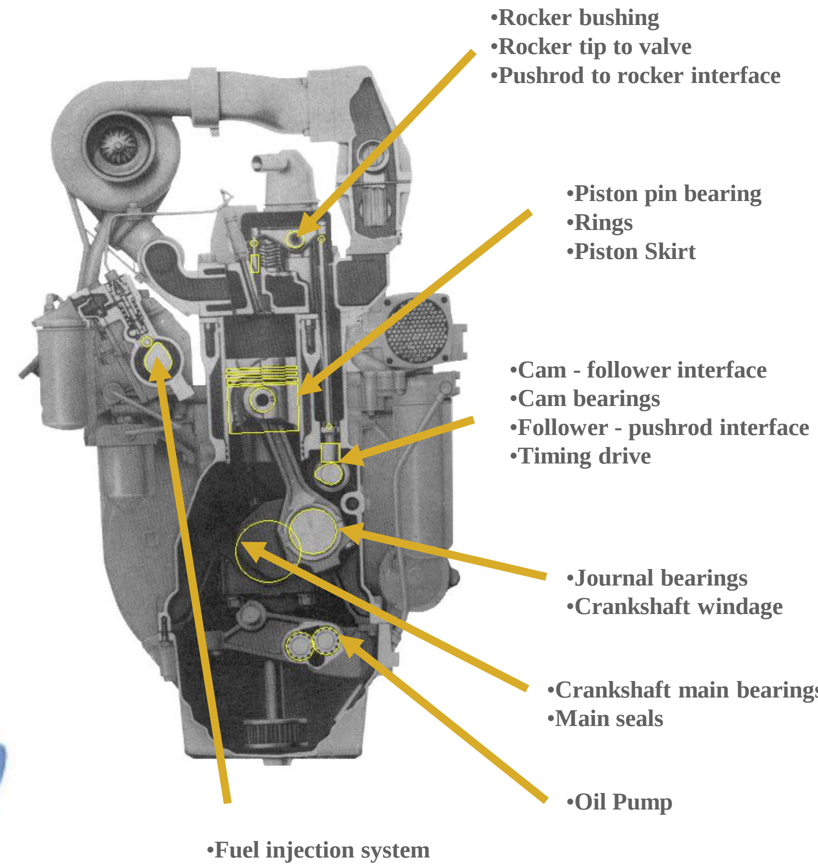
Outline

- Review some of our work leading to an unexpected finding of strong frictional anisotropy under boundary sliding
- Advance some ideas for the reason for frictional anisotropy
- Present detailed measurements of friction using various contact pressures, sliding speeds and materials

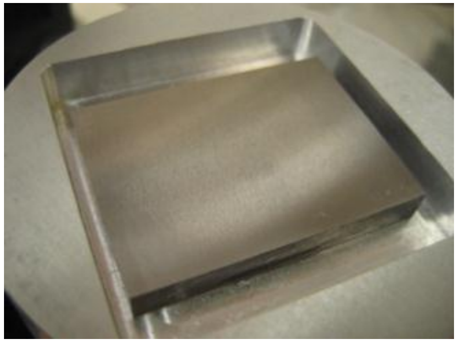
Goal - reduce frictional energy losses

- Study the friction in engine/vehicle systems, including axles, drives, and transmissions in heavy duty applications with the goal of reducing energy consumption
- Case-carburized gear steel lubricated with base stock and fully formulated oils



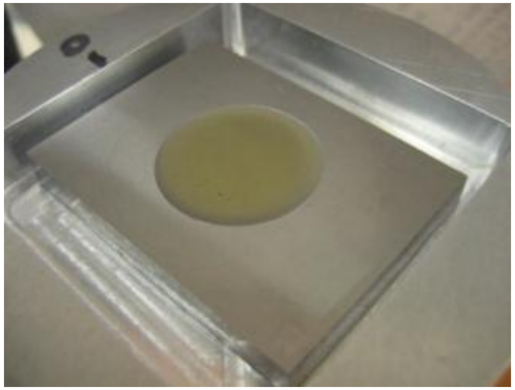
Task

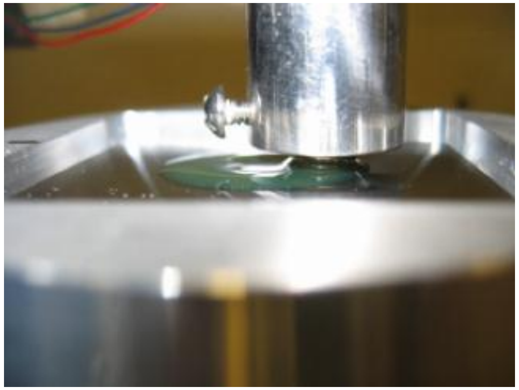
- Measure the friction and wear behavior of oil-lubricated steel surfaces
- Test
  - Ball-on-disk test geometry (CSEM)
  - Counterfaces:
    - Ground AISI 8620 case carburized Cr-Mo-Ni low alloy gear steel flat - 62 Rc (6.7 Gpa) hardness, roughness ≈ 0.4 µm Sa
    - 0.5” ø (13 mm) 52100 steel ball, 62 Rc
  - Load 5 N deadweight
    - Nominal Hertzian contact 0.46 GPa, 120 µm dia
  - Rotation speed adjusted to produce 1, 0.1, 0.02, 1, 5, 10 cm/s sliding speeds



Test Machine

- All tests were performed on same steel flat using different 52100 balls, by varying the track diameter in steps of 1 mm,
- Room temperature
- 2” x 1.5” (51 x 38 mm) rectangular flat was leveled to 0.002” (51 µm)
- Oil quantity was ≈ 130 mm^3
  - Oil pool was always present





Lambda ratio

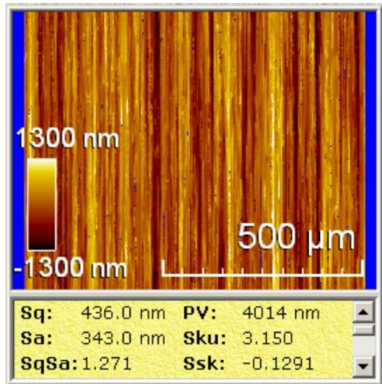
- Lubricant fluid film thickness / composite roughness
- Lambda ratio calculated for base stock used
- Base stock – 19 cSt at 40C
- 10W30 – 63 cSt
- Gear oils – various

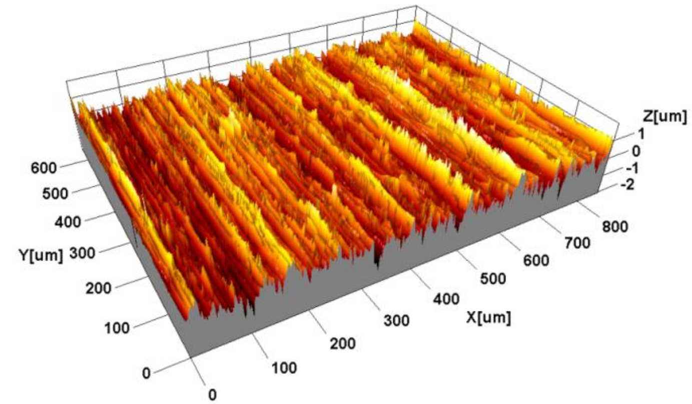
Sliding speed (cm/s)	Calculated lubricant fluid film thickness (nm) for PAO4	Calculated $\lambda = H/\sigma$
0.02	1.1	0.0069
0.1	3.5	0.021
1	17.9	0.106
10	92.0	0.544

Test Flat

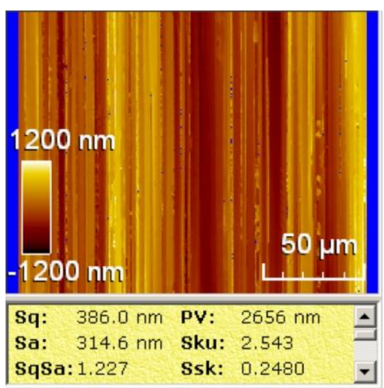
- Anisotropically ground 8620 gear steel
- Optical profilometry used to show surface topography using false color
- Surface has very parallel grinding marks
  - Surface RMS roughness around 400 nm

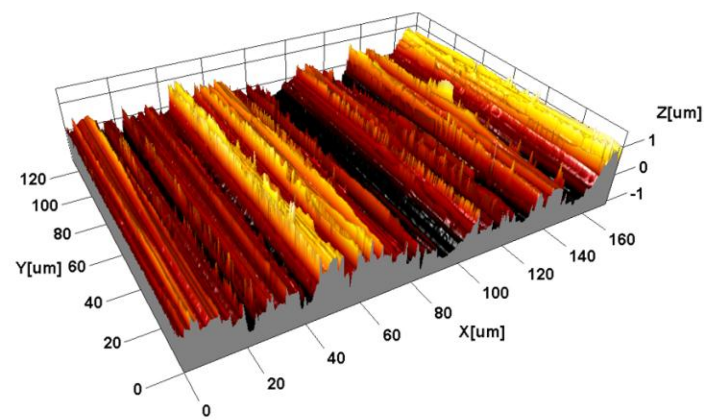
Low mag





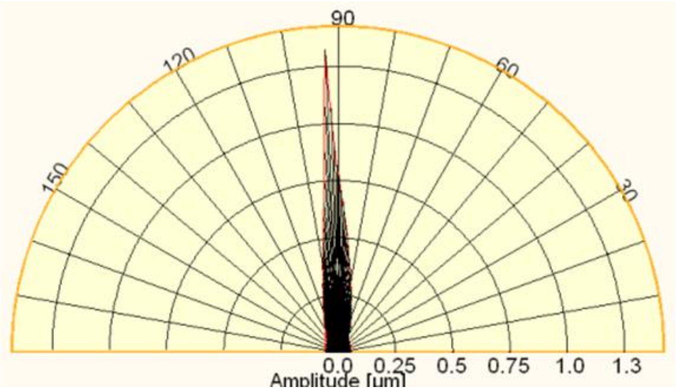
High mag



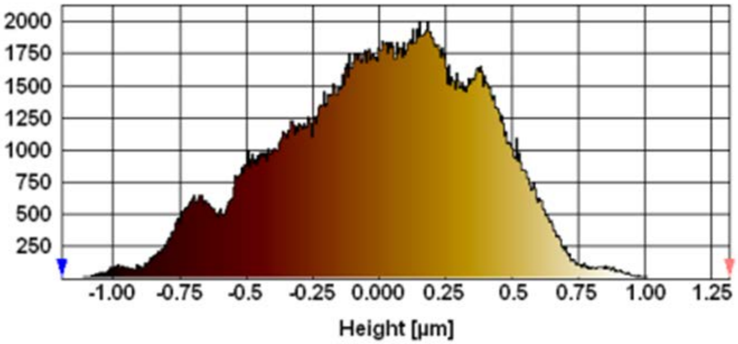


Test Flat

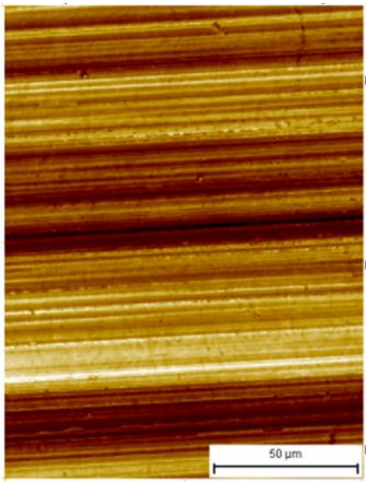
- Anisotropically ground 8620 gear steel
- Strong anisotropy



Angular spectrum of surface roughness



Histogram of height distributions

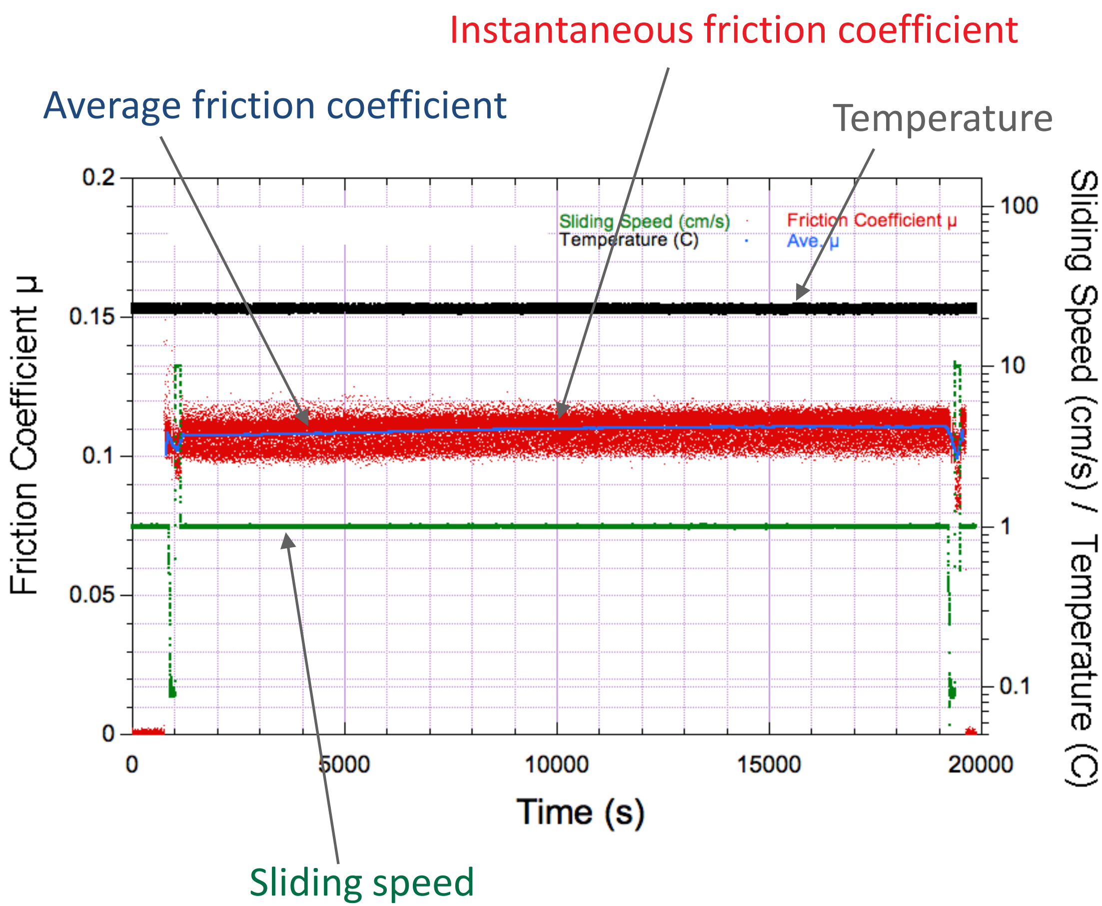


Test Procedure

- Test Procedure
  - Confirm that flat has been cleaned (acetone), proper oil added, new ball installed, and that track radius and starting position are correct
  - Start data acquisition
  - Record calibration using 5 N deadweight
  - lower ball and start sliding at **1, 0.1, 10 cm/s sliding speed**s for 2 minute periods
  - Continue run **~ 5 hours at 1 cm/s** (slower data rate)
  - Continue sliding at **0.1, 10, 1 cm/s** speeds for 2 minute period
  - Data stored
  - Obtain surface profilometer data on ball (no measurable track present)

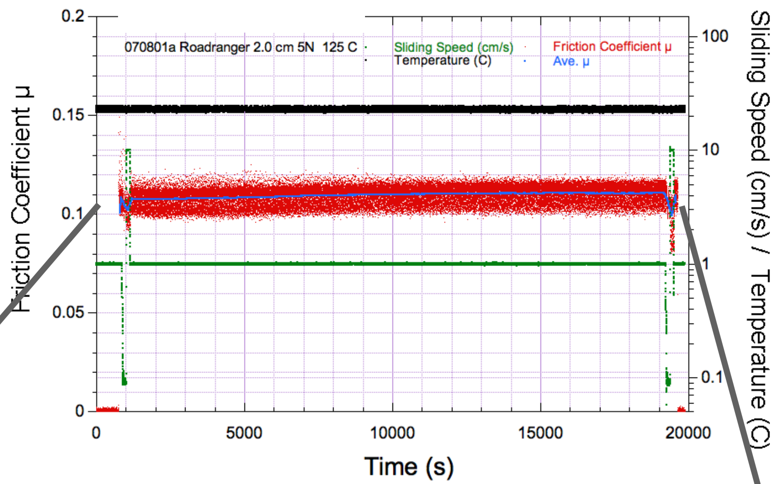
Typical sliding test results

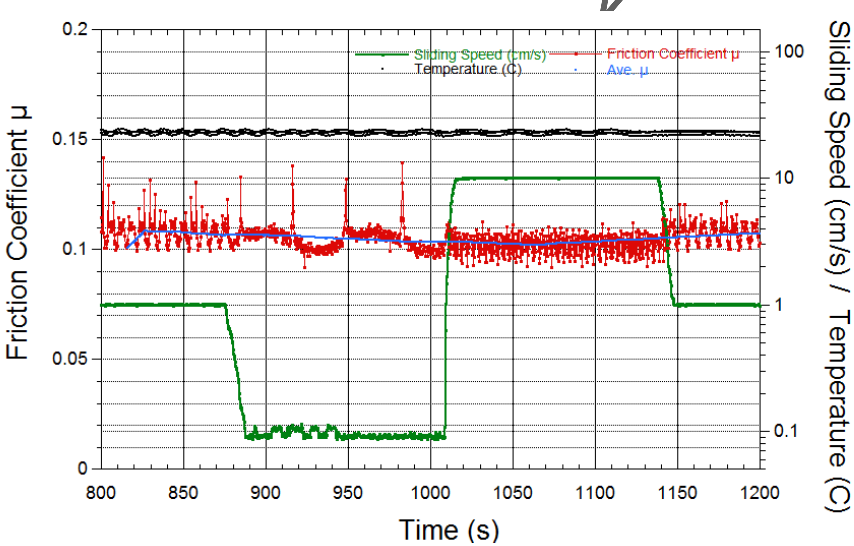
- 5 hr friction graph

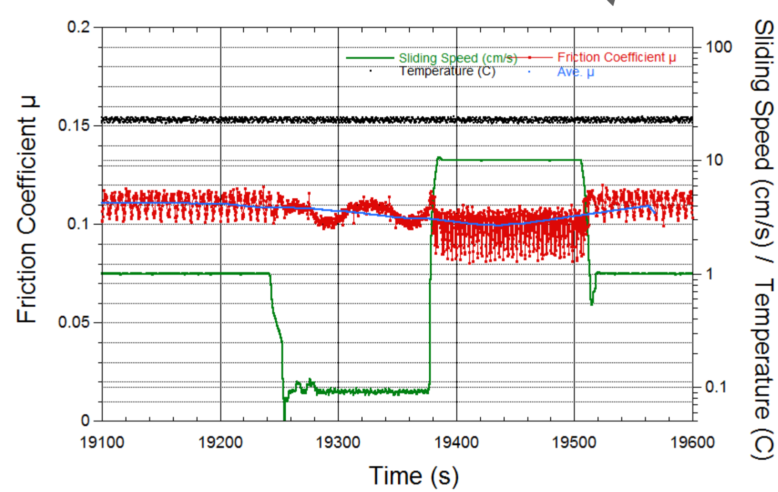


Details of pre-test and post-test sliding

- Essential to verify that sliding at 1 cm/s for 5 hours is always in boundary regime, nowhere approaching mixed lubrication
- Done by varying sliding speed 100:1 at beginning and end of test and noting essentially constant friction coefficient

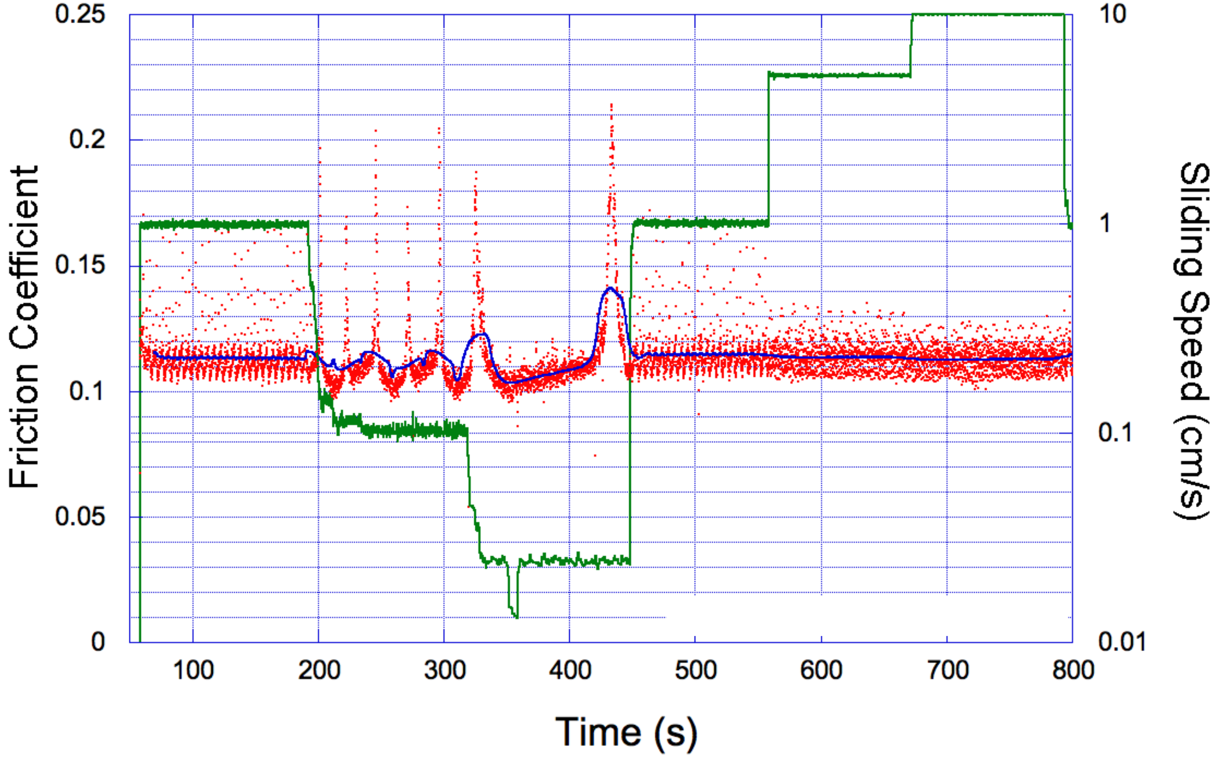






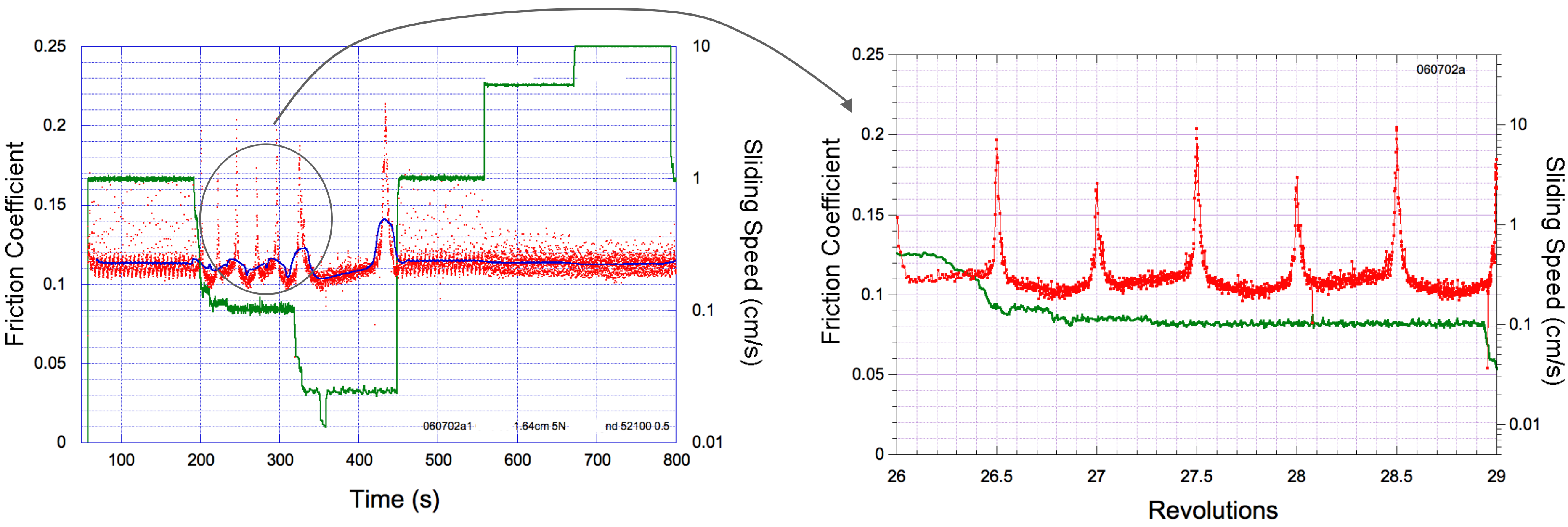
Anisotropic surface texture effect on friction

- Friction peaks or "spikes" at periodic intervals are observed
- Spikes are correlated with rotation of flat
- Momentary friction is up to 100% larger on "spikes"



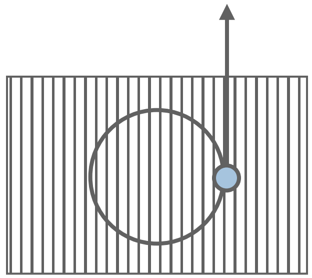
Anisotropy effect on friction

- Friction spikes are correlated with rotation of specimen
- Occur twice during every rotation and are non-sinusoidal

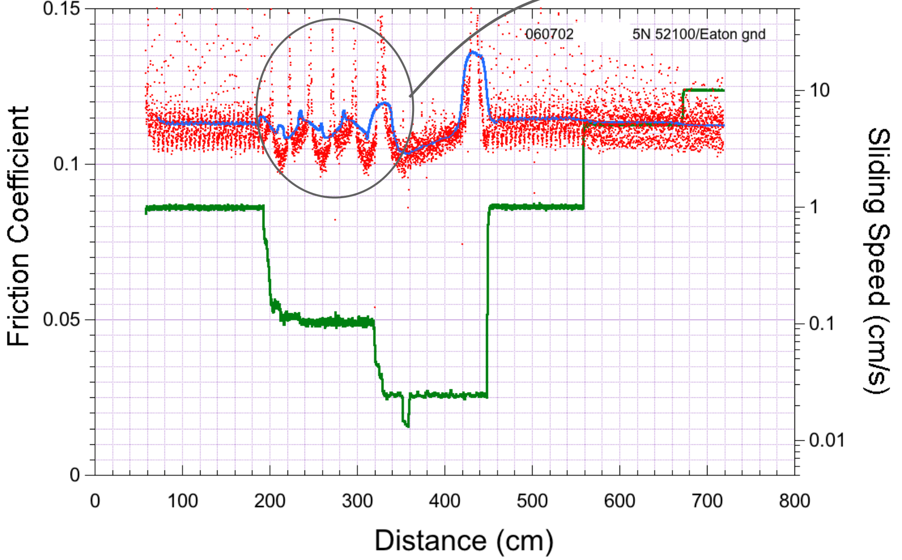


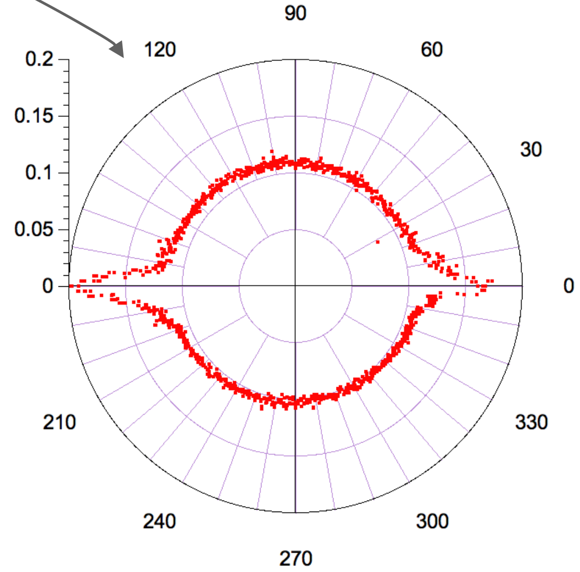
Anisotropy effect on friction

- Polar graph shows that friction spikes are synchronized to direction of sliding
  - Friction quite constant except when ball is sliding *parallel* or *near-parallel* to grinding lay



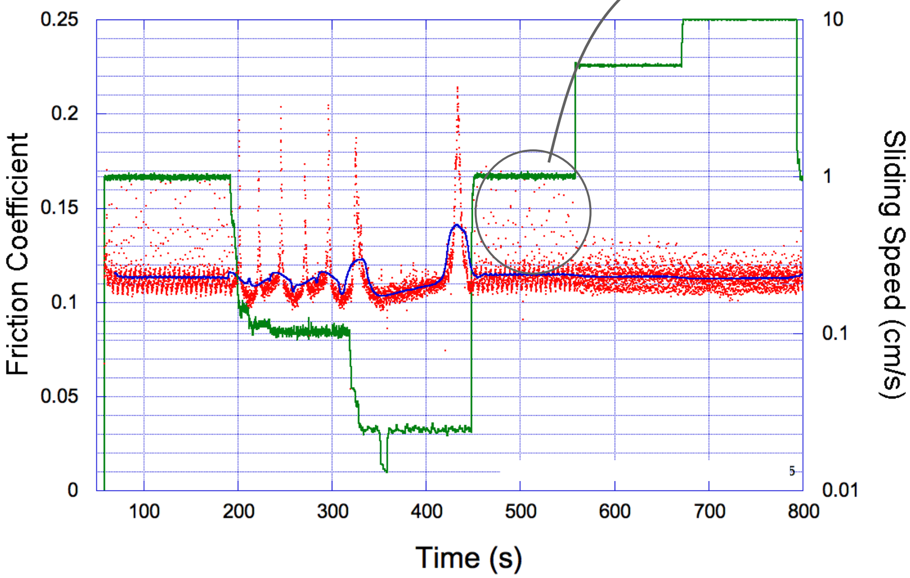
At 0° ball is sliding parallel to grinding lay

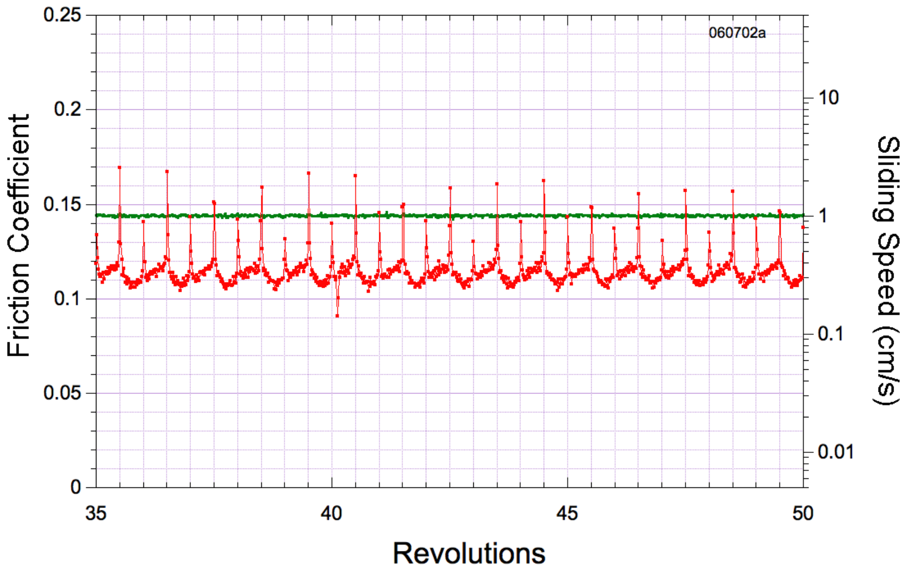




Anisotropy effect on friction

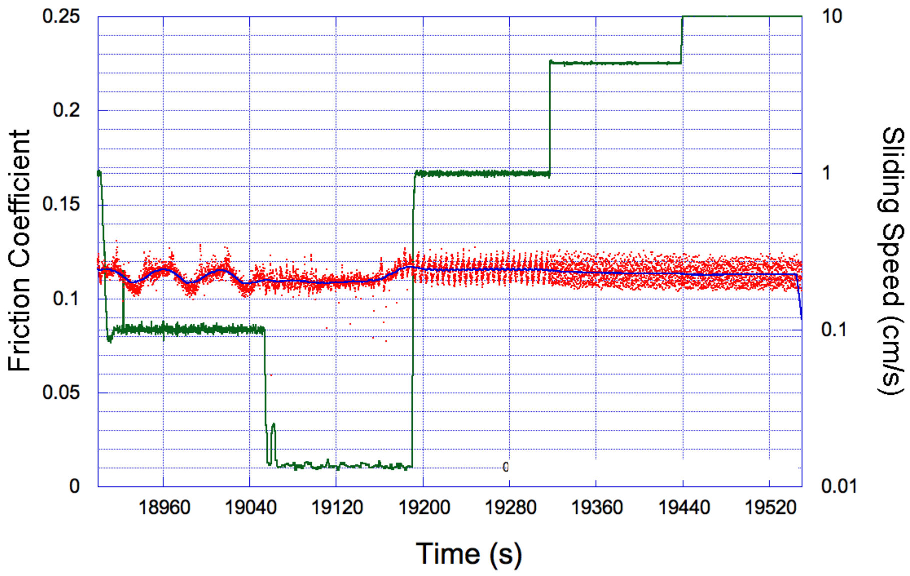
- The same effect is observed at a sliding speed of 1 cm/s





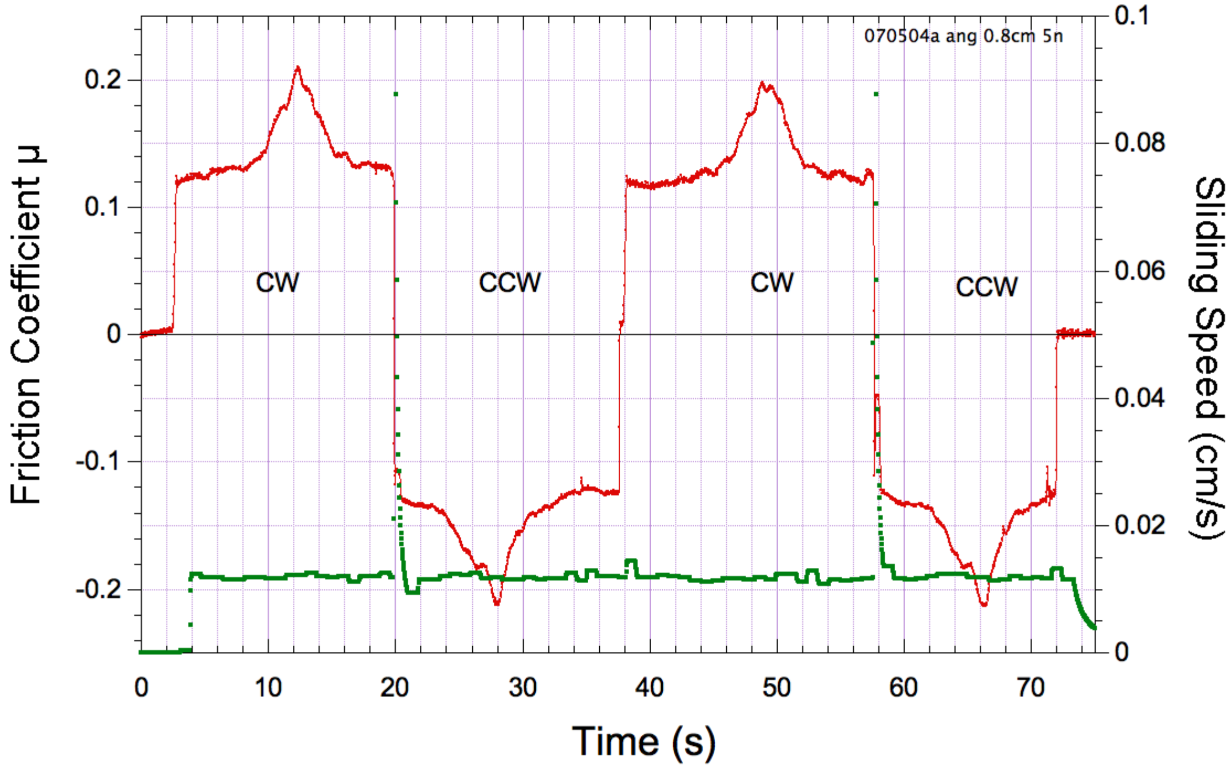
Anisotropy effect on friction

- At the end of test the “spikes” have largely disappeared



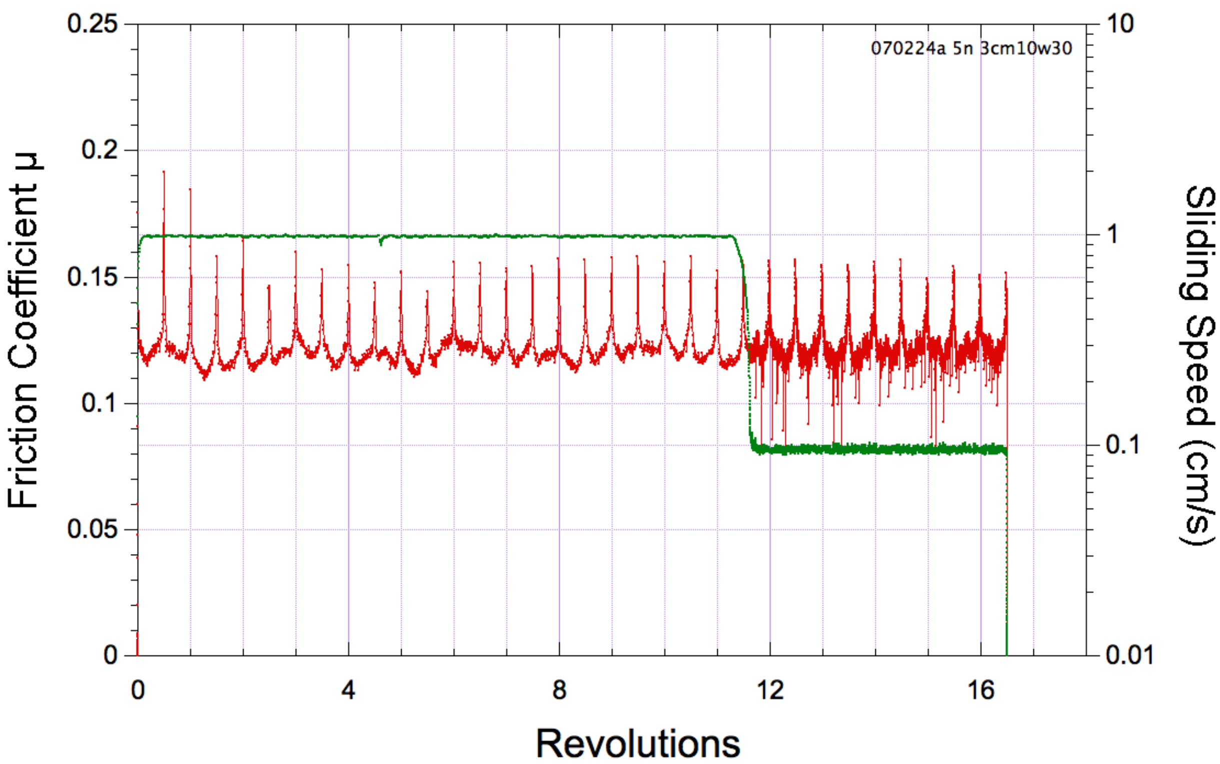
Anisotropy effect on friction

- In this test the flat was slowly motored ≈ 30° CW and CCW
  - Spikes repeatable, but not perfectly symmetrical with respect to sliding direction



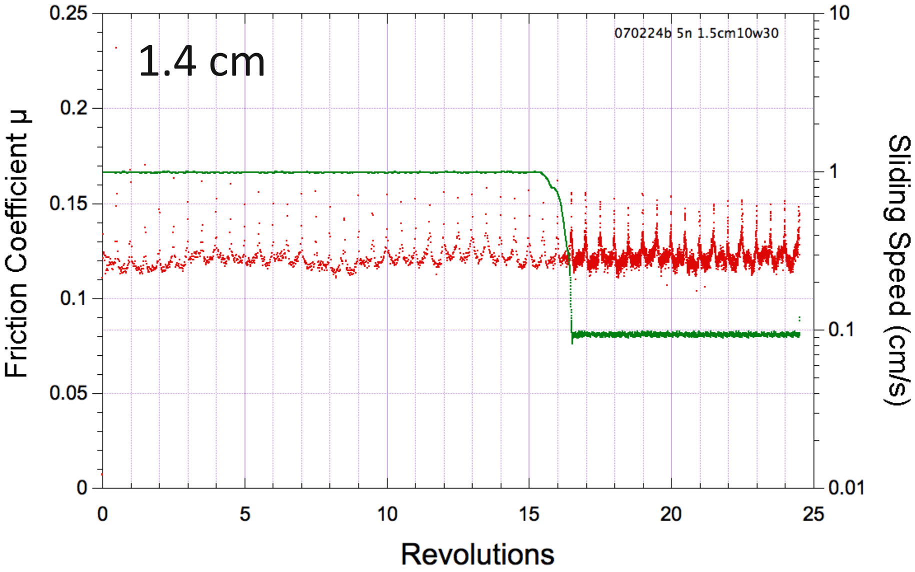
Anisotropy effect on friction

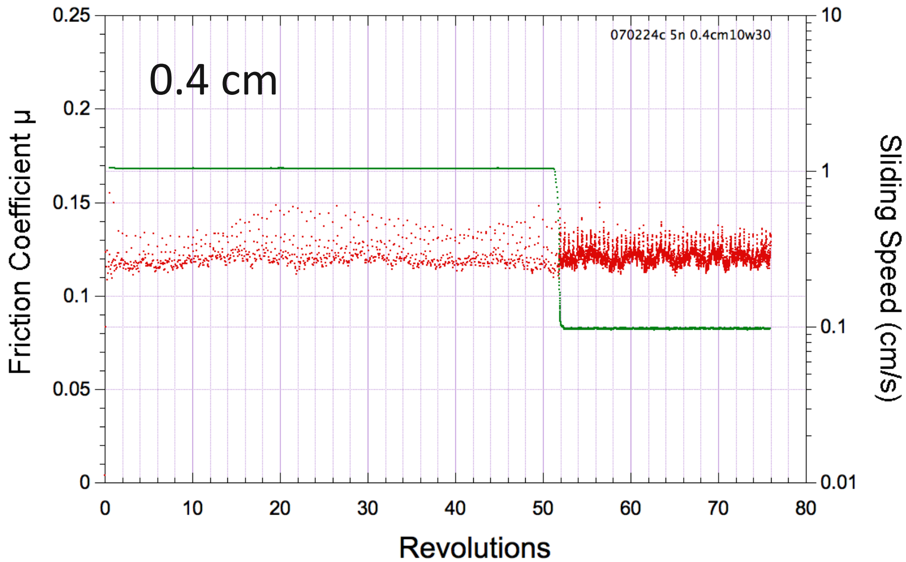
- Magnitude of effect insensitive to sliding speed (1 or 0.1 cm/s) for 10W30 test oil



Anisotropy effect on friction

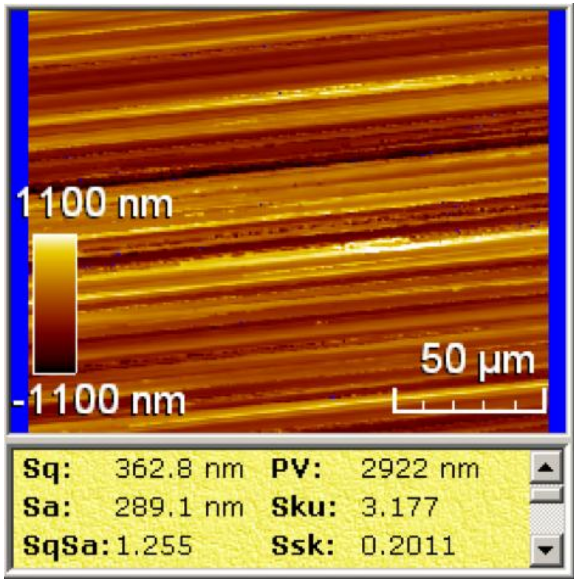
- Magnitude of effect is slightly sensitive to wear track radius (1.5 vs 0.4 cm)

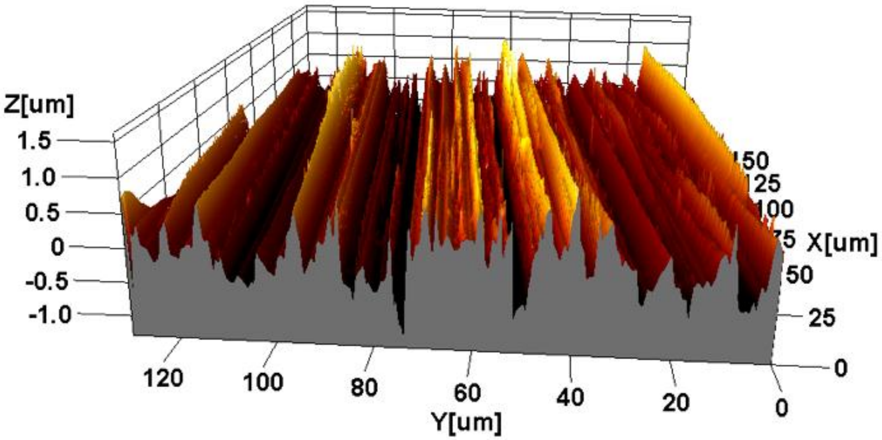


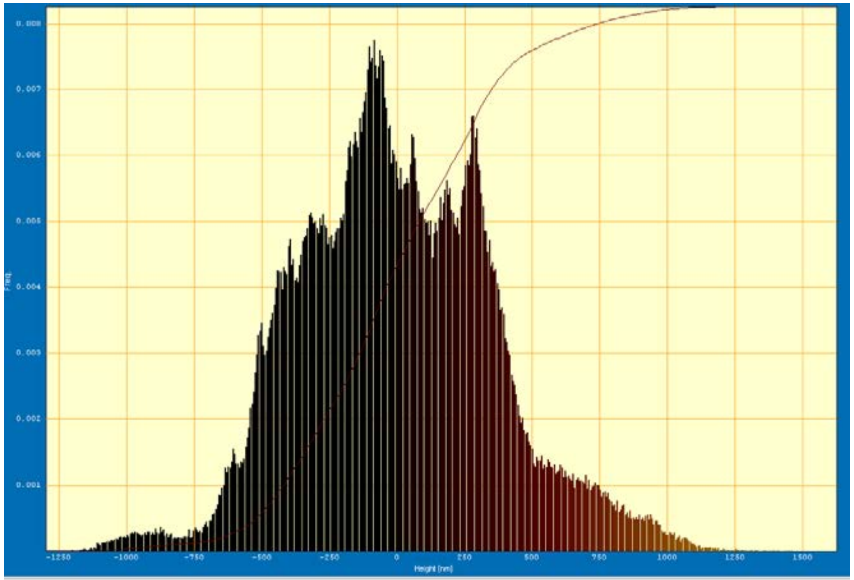


Unworn area

- Optical profilometry
  - Sa = 289 nm, kurtosis > 3, and skew = positive
  - indicates shallow valleys and pointed hills

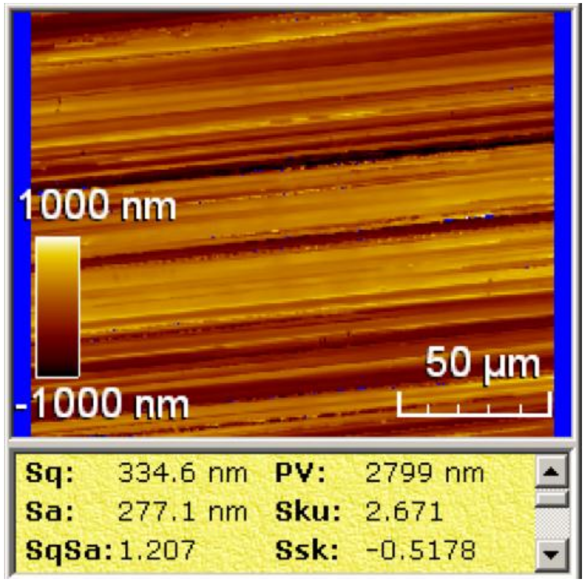


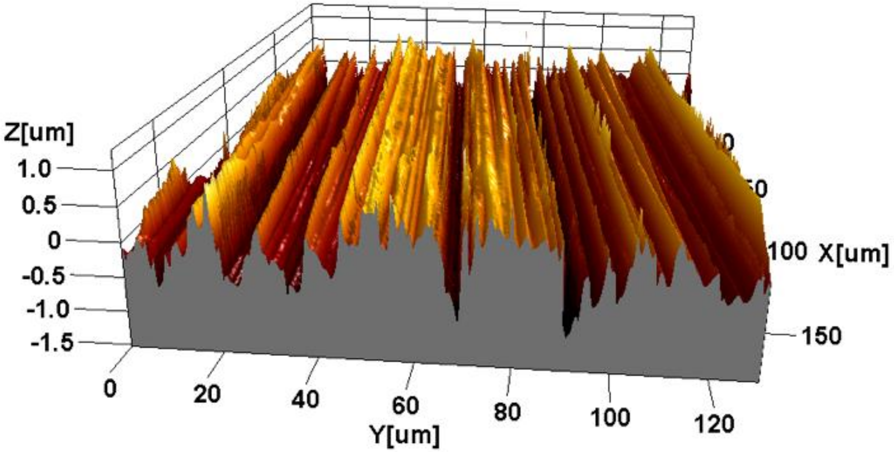


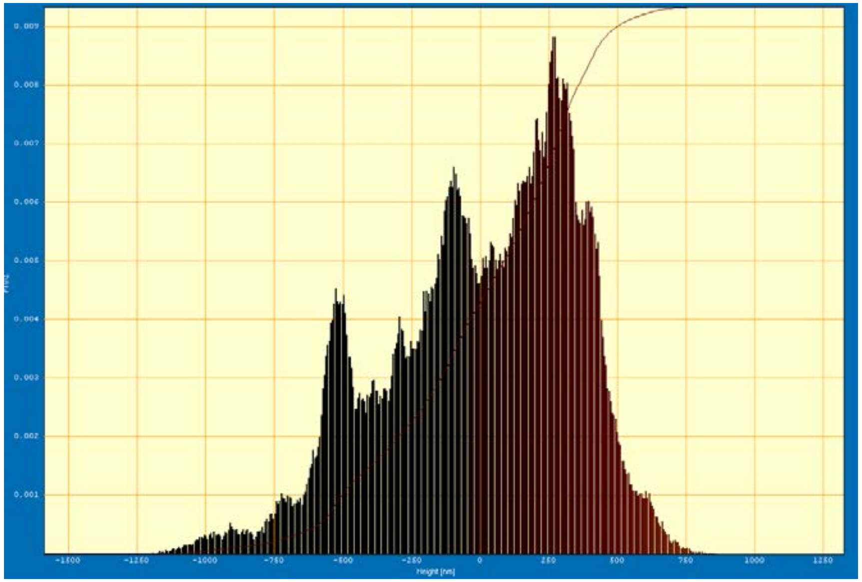


Worn wear track

- Worn area not obviously “flat” on top of asperities
  - Sa = 277 nm, kurtosis < 3, and skew = negative
  - Flattened hills with valleys





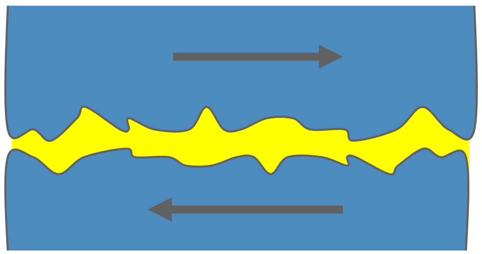


Friction Literature

- Analytical studies have concluded that the fluid film thickness between sliding contacts is thinner when sliding is in the direction parallel to grinding lay, than across grinding lay.

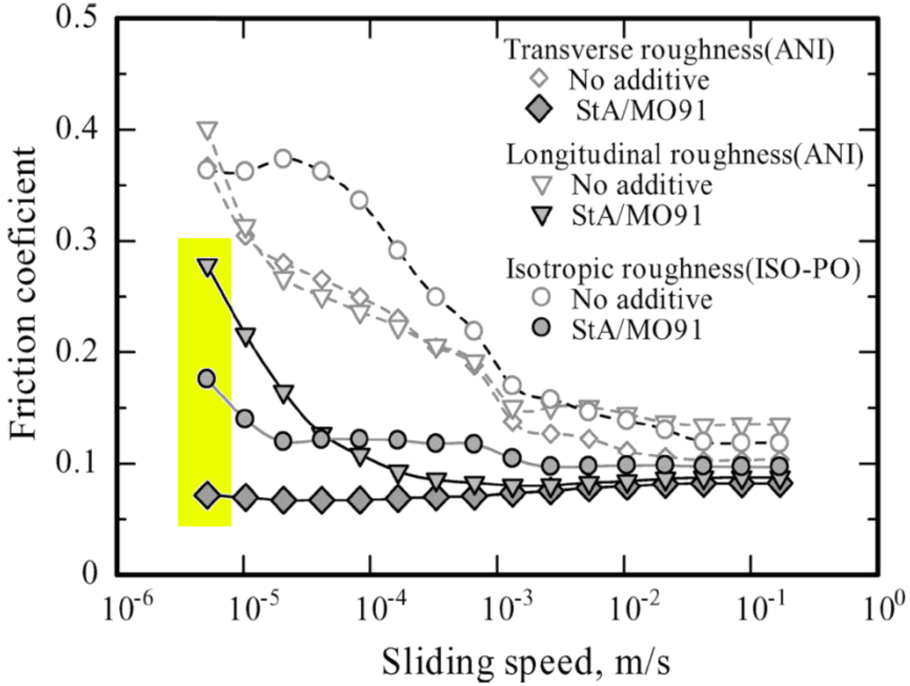
- Mechanisms that influence friction between sliding surfaces include:
  - Counterface mechanical properties
  - Interaction of asperities
  - Presence or absence of reaction or chemical boundary films
  - Rheological properties of the lubricant
  - Sliding speed and load

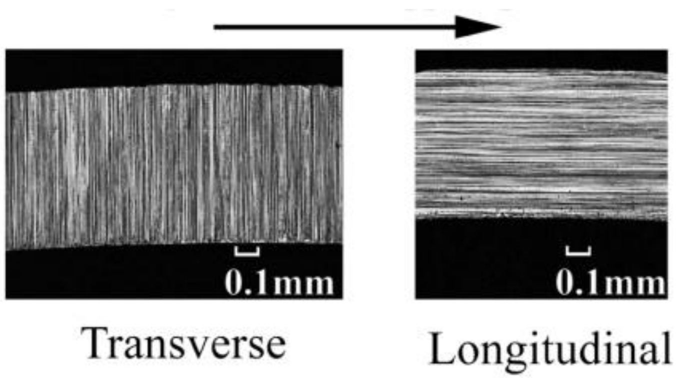
- Few experimental studies of texture effect on friction, for different materials



Reports

- Steel roller-on-ring test with base stock and formulated oils
- No data about sensitivity of coefficient of friction to exact parallelism of sliding

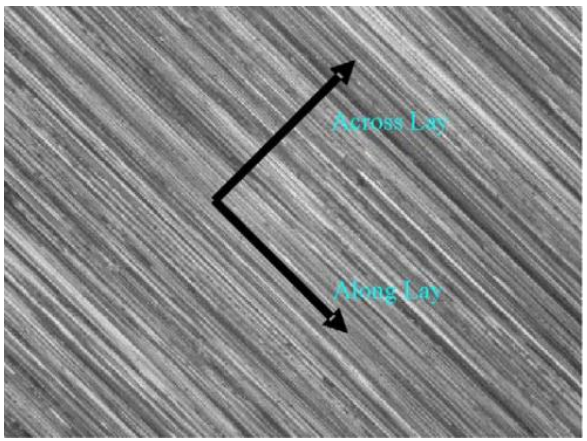


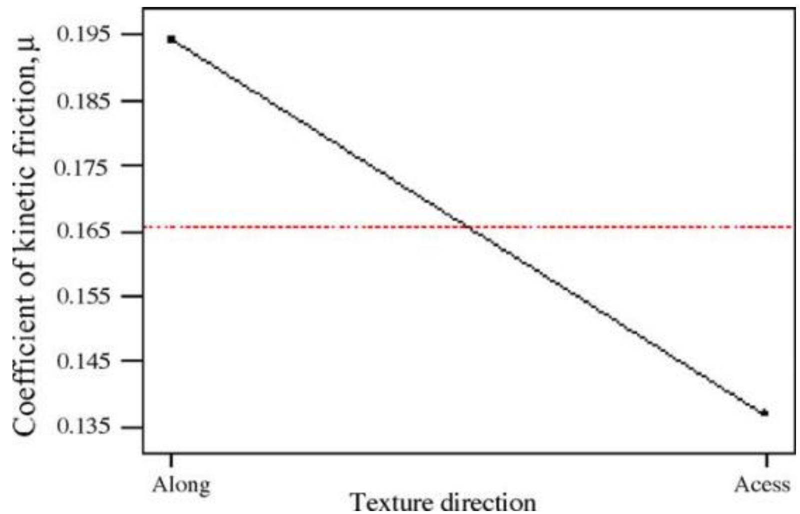


Masabumi Masuko, Saiko Aoki and Akahito Suzuki
Tribology Transactions, 48: 289-298, 2005

Reports

- Steel ball-on-flat test with 5W30 formulated oil at 2.34 GPa Hertzian contact pressure
- COF 0.195 when sliding parallel to ridges
- COF 0.135 when sliding across ridges



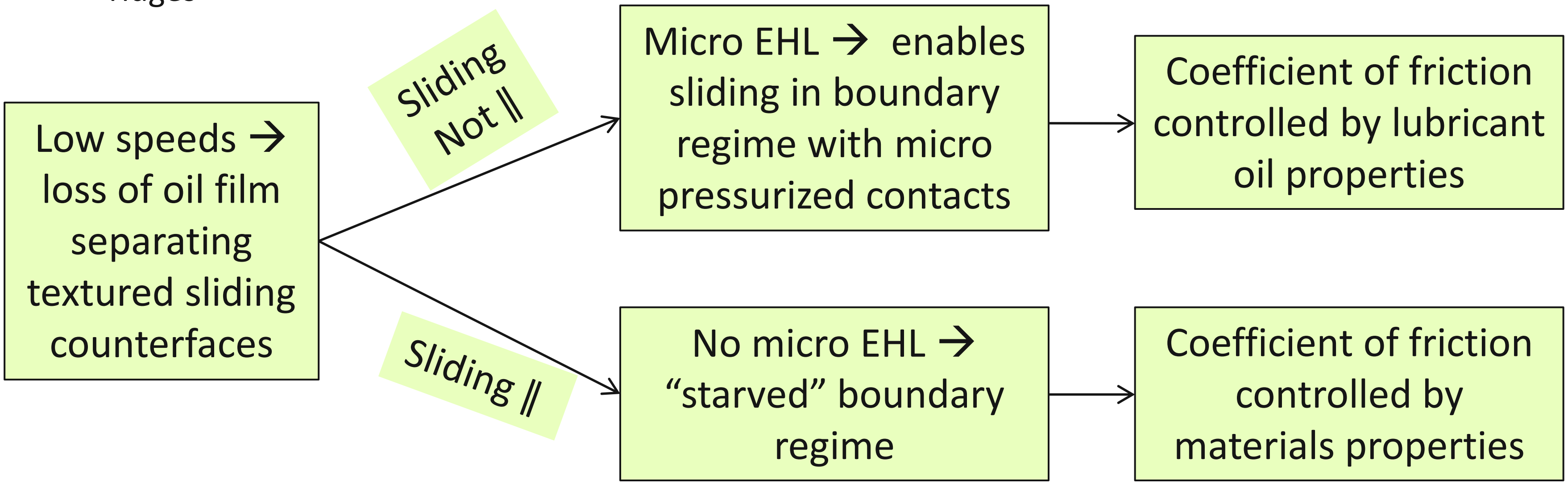


R Singh, S. Melkote, F. Hashiomoto, Wear 258 (2005) 1500-1509

Hypothesis

- Lambda ratio = thickness of lubricant film/composite roughness
- λ ratio for sliding contacts influenced by inlet geometry, surface texture, geometry, oil rheology, load, and speed
- With high λ ratio, sliding will be in mixed or hydrodynamic regime, friction due primarily to shearing of lubricant film - COF is low
- It is hypothesized for pure longitudinal sliding that micro EHL is inhibited because of two factors; rapid side leakage of oil from ridges, and that long ridge provides no adequate convergent inlet zone for oil entrainment

  …But that micro convergent inlet zones are still present for sliding non-longitudinally across ridges



Consequences

- Because "spike" effect controlled by 1) presence of longitudinal ridges, and 2) consequent presence or absence of micro EHL.
  1) Soft textured counterface material will lose "spike" effect quickly as plasticity blunts the ridges
  2) Hard textured counterface material will retain "spike" effect indefinitely
  3) Counterface pairs with intrinsic low unlubricated COF will have small spike magnitude (unlubricated , e.g. air or N2)
  4) Counterface pairs with intrinsic high unlubricated COF will have large spike magnitude as asperities interact
  5) Spike effect should be enhanced for high loads and slow sliding speeds

Results

- A number of tests with PAO10 basestock were conducted to explore variations in load, speed and track diameter on the behavior of the frictional anisotropy
- Four contact pressures were used, with two of them similar (0.58 vs 0.68 GPa) but used different loads
- One test repeatedly reverse the disk direction such that the ball slid repeatedly back and forth over the same area

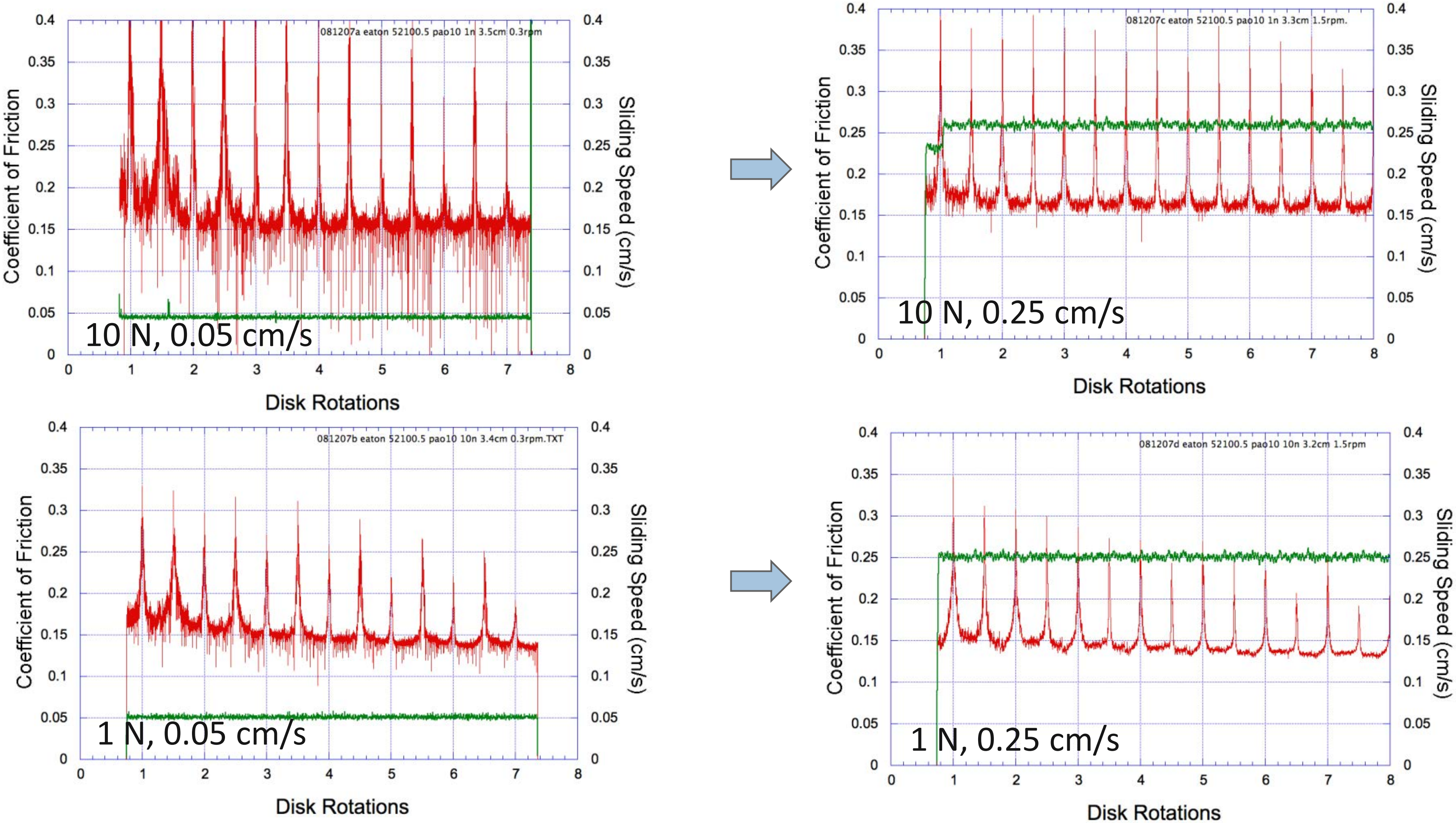
Ball dia (mm)	Load	Mean P	Contact dia
12.7 mm	10 N	0.58 GPa	148 µm
12.7 mm	1 N	0.27 GPa	68 µm
3.175 mm	10 N	1.45 GPa	94 µm
3.175 mm	1 N	0.68 GPa	44 µm

Results

- A, Large ball, low load, large track, slow
- B, large ball, high load, large track, slow
- C, large ball, low load, large track, fast
- D, Large ball, high load, large track, fast
- E, Large ball, low load, small track, fast
- F, Large ball, high load, small track, fast
- G, small ball, low load, small track, fast
- H, small ball, high load, small track, fast
- I, Small ball, low load, large track, fast
- J, Small ball, high load, large track, fast
- M, small ball, high load, large track, fast, reversals

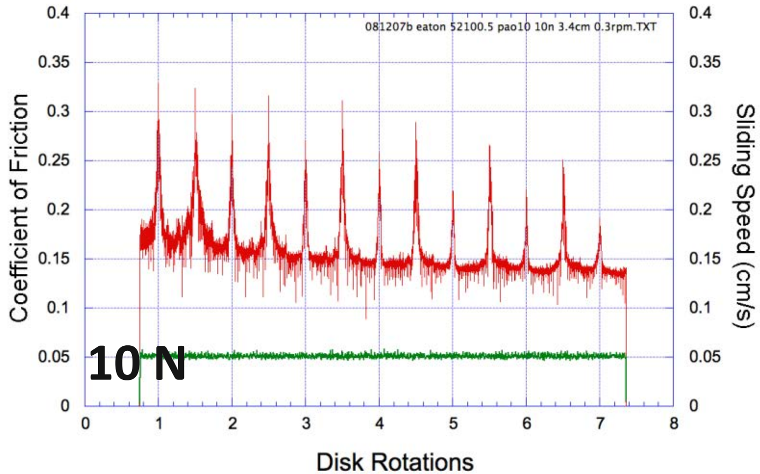
Results

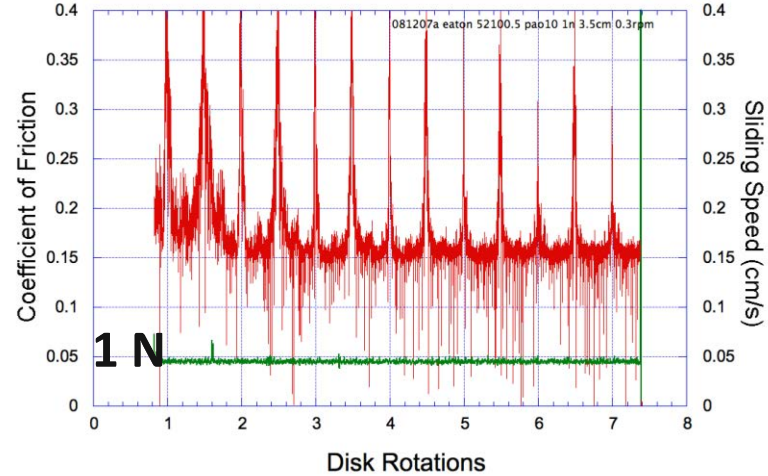
- Slightly smaller spikes were measured at higher sliding speeds for both 1 N and 10 N load

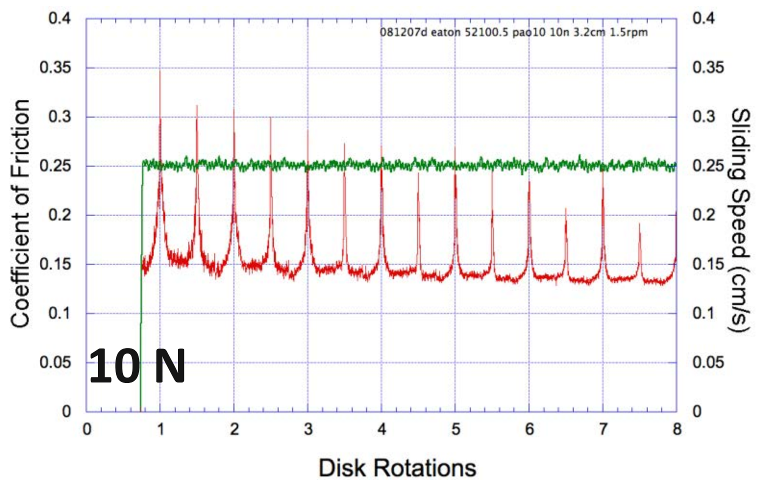


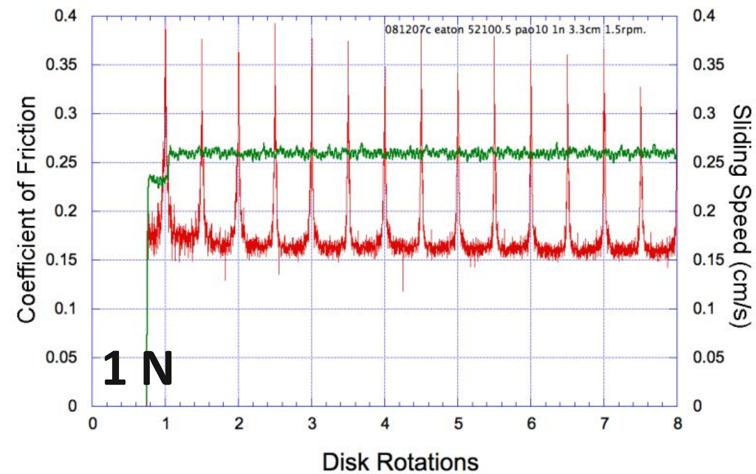
Results

- Sliding at lower load produced larger friction spikes – not expected



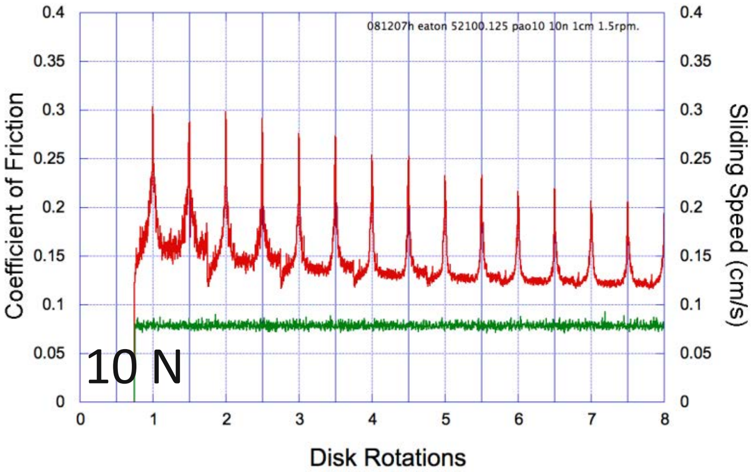


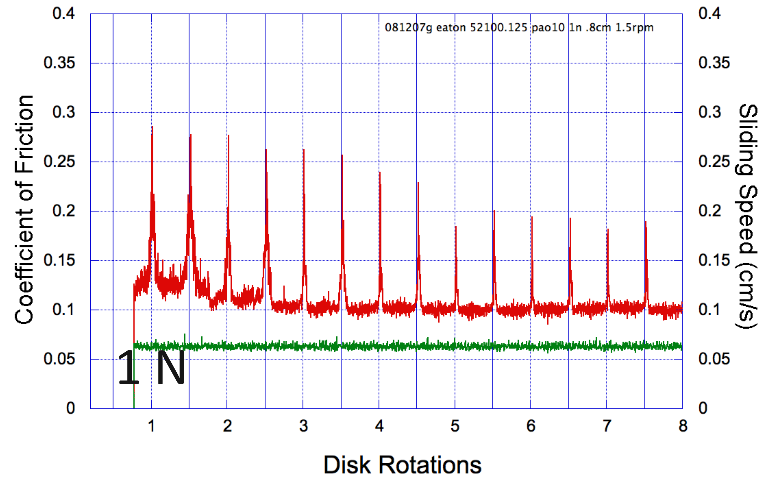


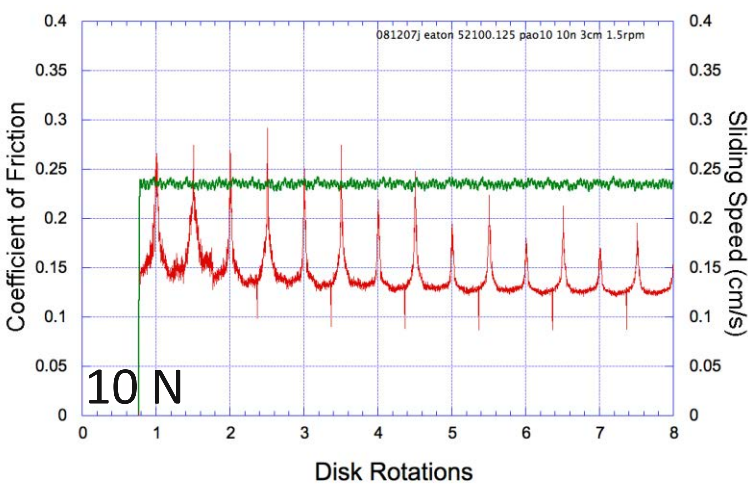


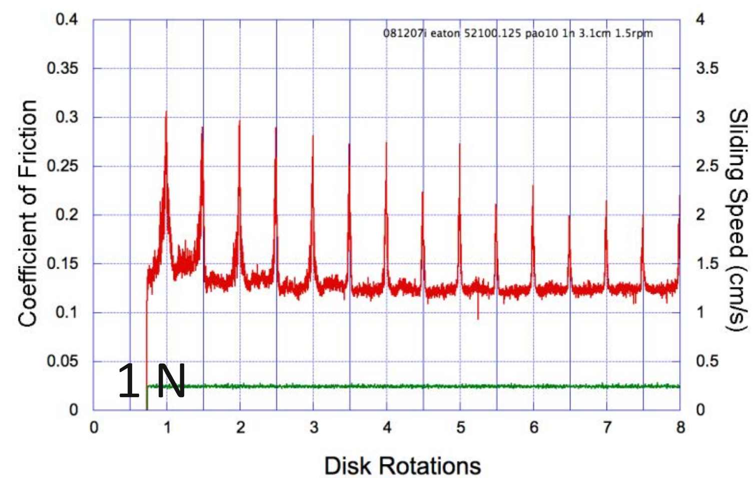
Results

- Low load produces larger spikes (small ball)



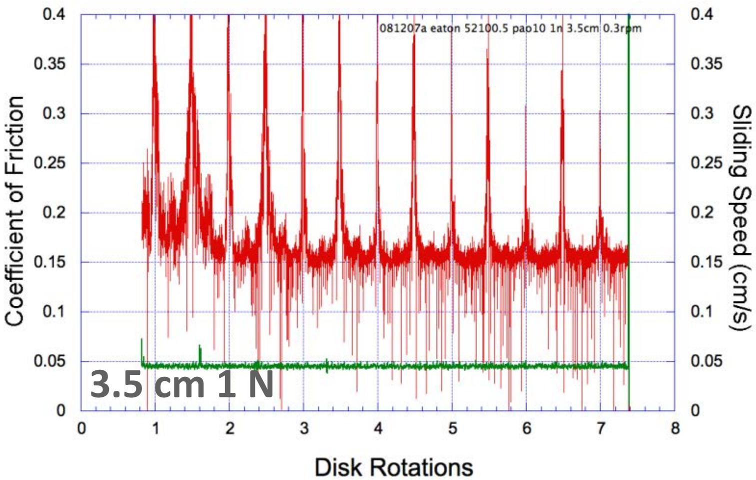


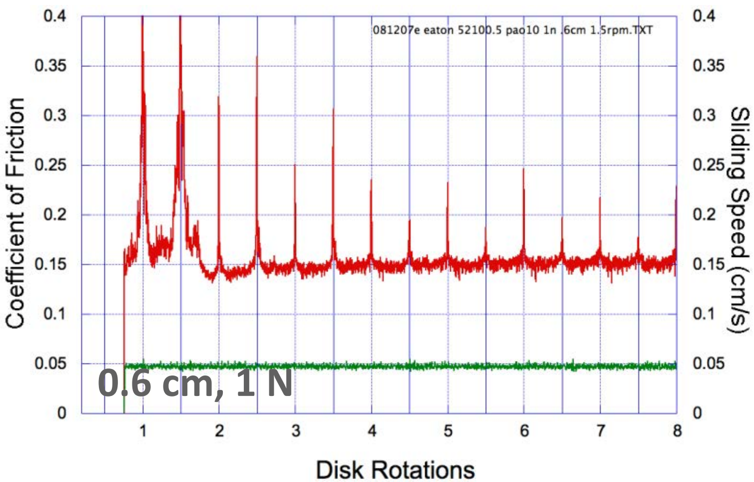


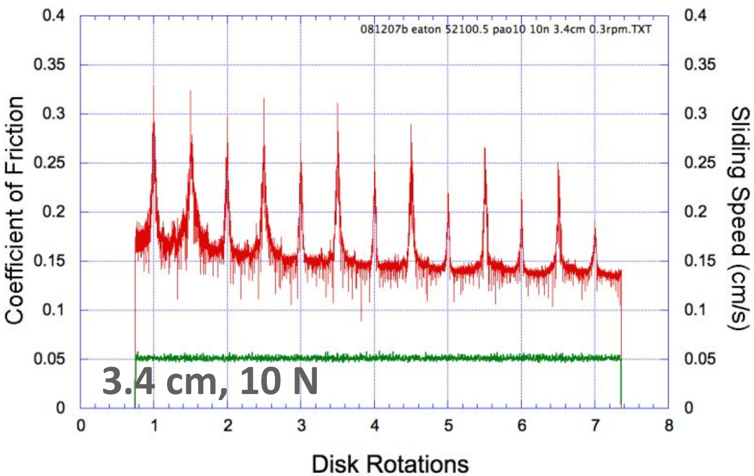


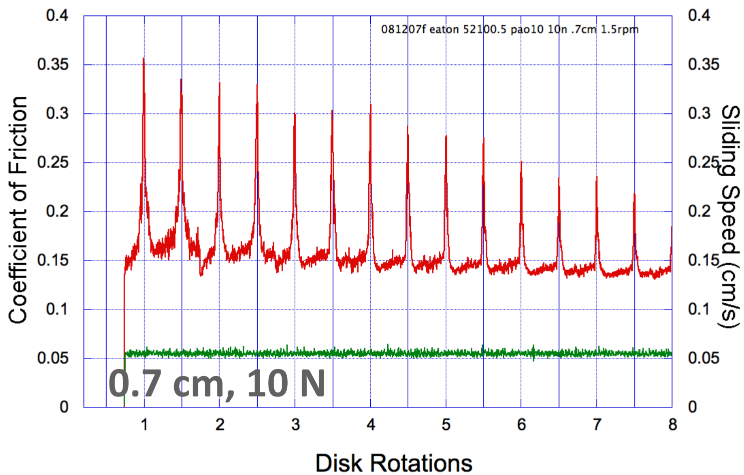
Results

- Effect of track diameter on friction is inconsistent



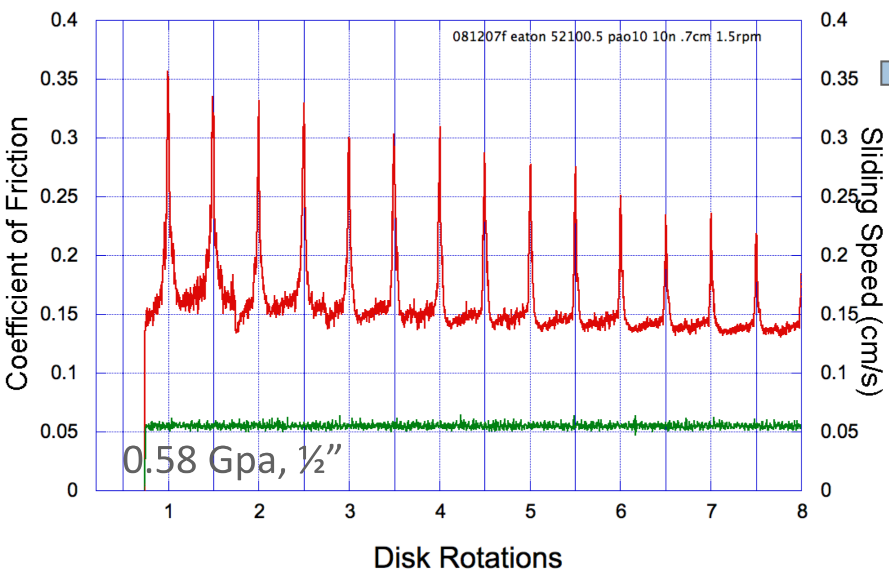


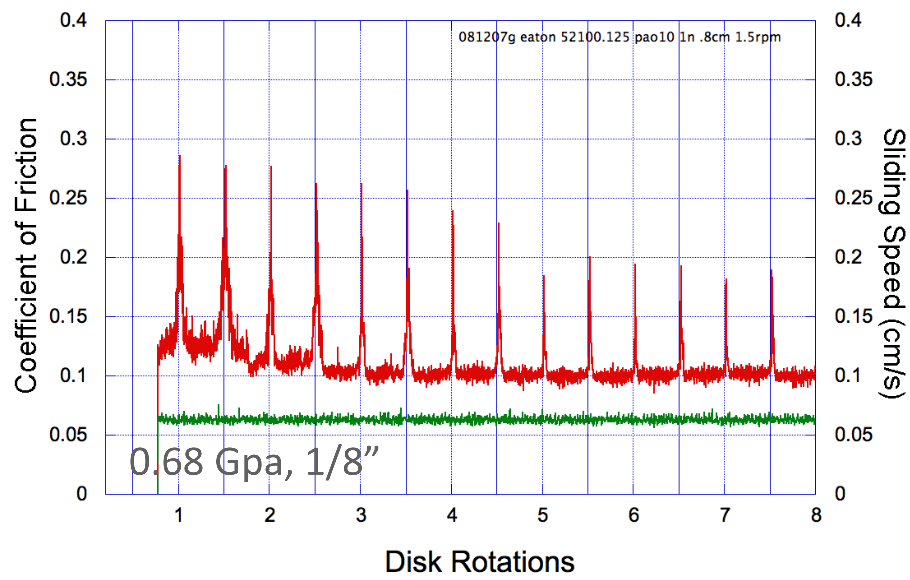




Results

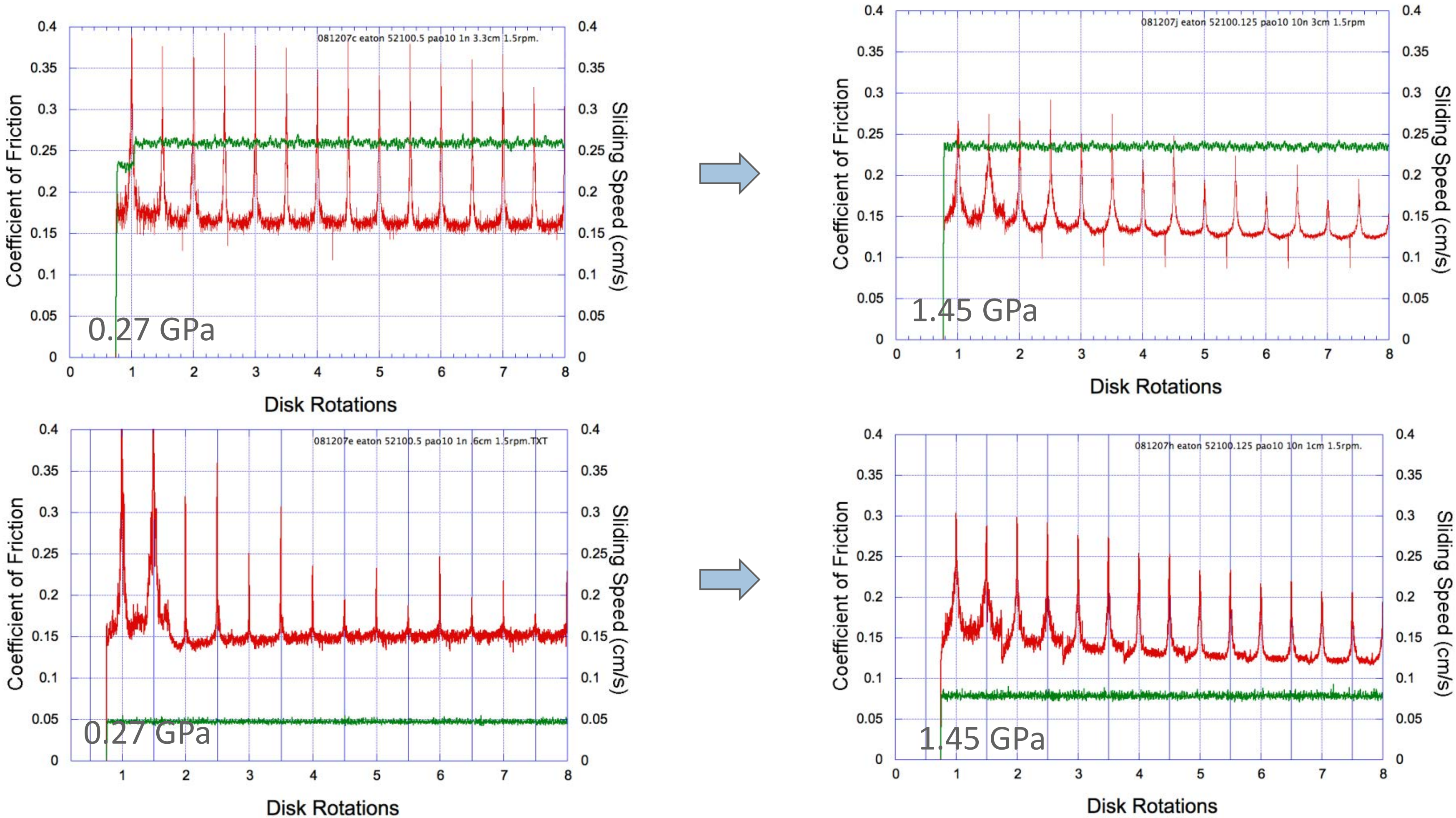
- Near identical Hertzian pressure conditions gives near identical behavior regardless of ball size
- Poor lubricating properties of PAO10 causing a rapid blunting of ridges and rapid reduction in spike size





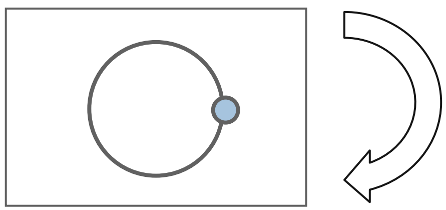
Results

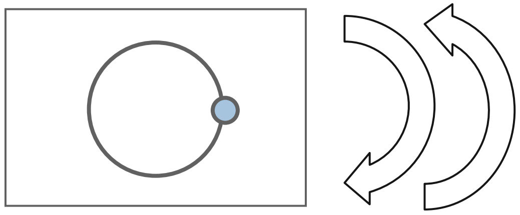
- Different Hertzian contact pressures show no clear trend

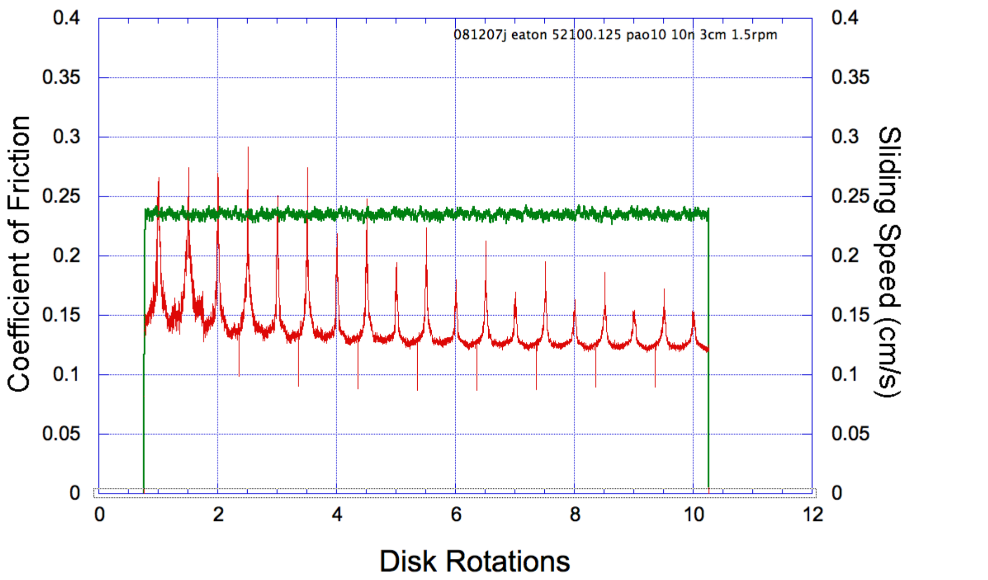


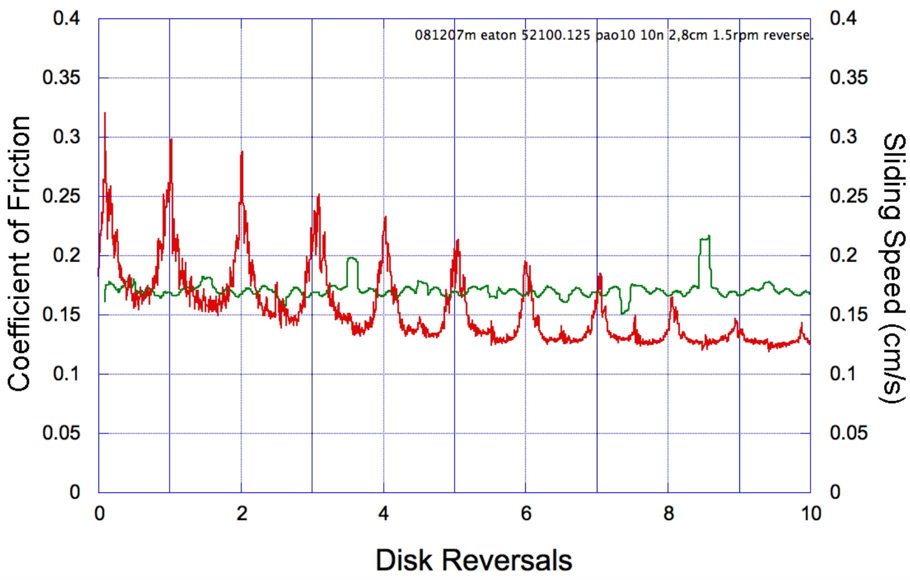
Results

- Oscillating test caused more rapid frictional spike loss than unidirectional
- Attributed to ridge blunting caused by fatigue









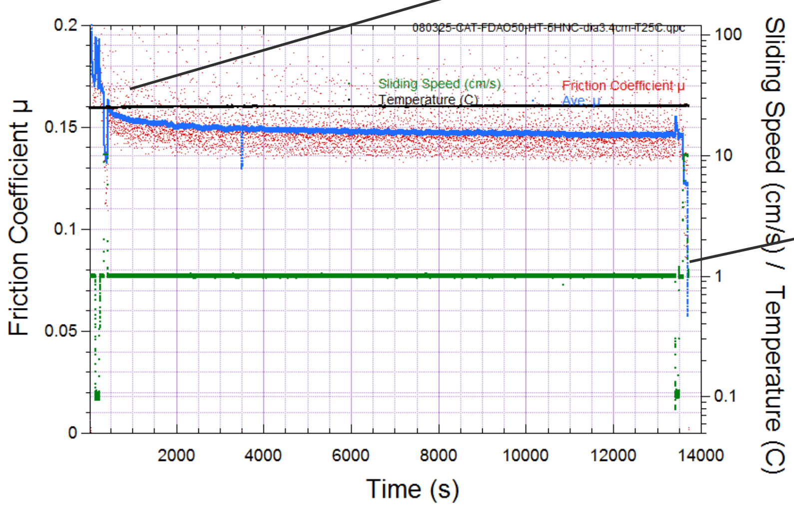
Effect of material comparison

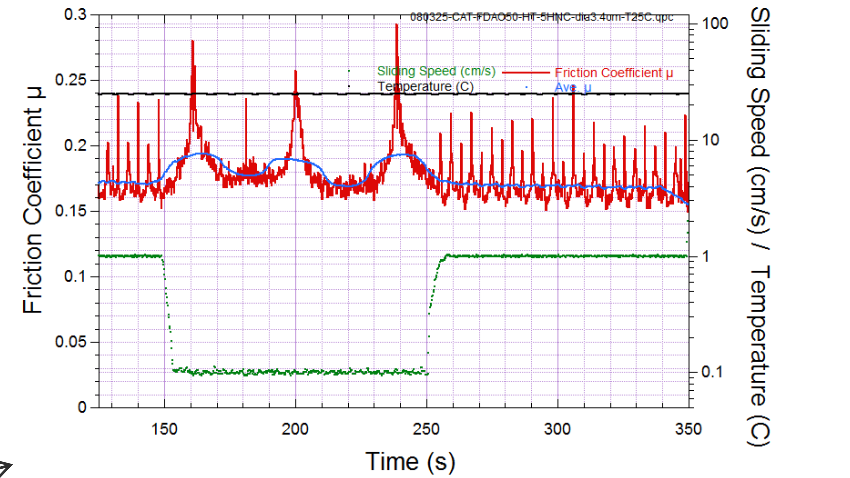
- If spike effect is caused by loss of micro EHL and consequently greater intimate asperity contact, then spike amplitude should trend with the intrinsic friction of counterface materials in the absence of oil lubrication, i.e., “dry”
- Extremely hard counterface will be resistant to ridge blunting and exhibit persistent spikes

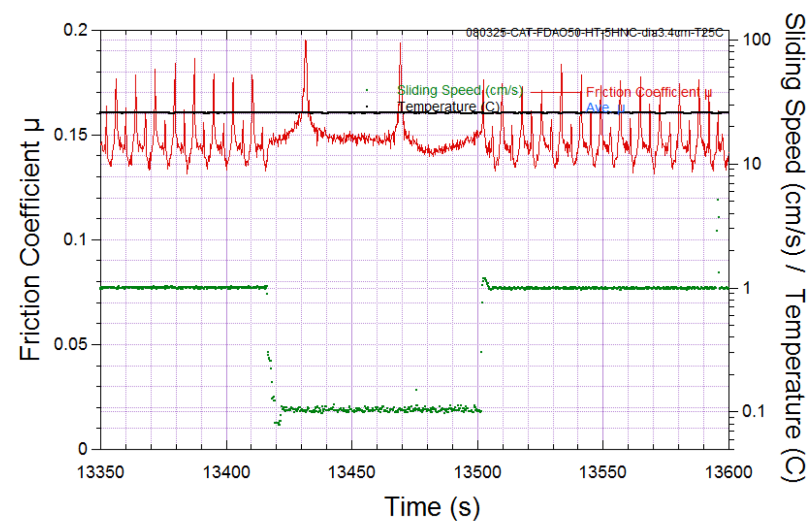
- Ground flats coated with
  – Hard wear resistant nitride coating that has COF near 0.4 in dry N_2 or air
  – Hard wear resistant hydrogenated DLC coating that has COF near 0.02 in N_2 and 0.1 in air

Comparison - Mo nitride coating

- Extreme hardness of coating resists blunting of ridges, maintaining frictional spike through end of test

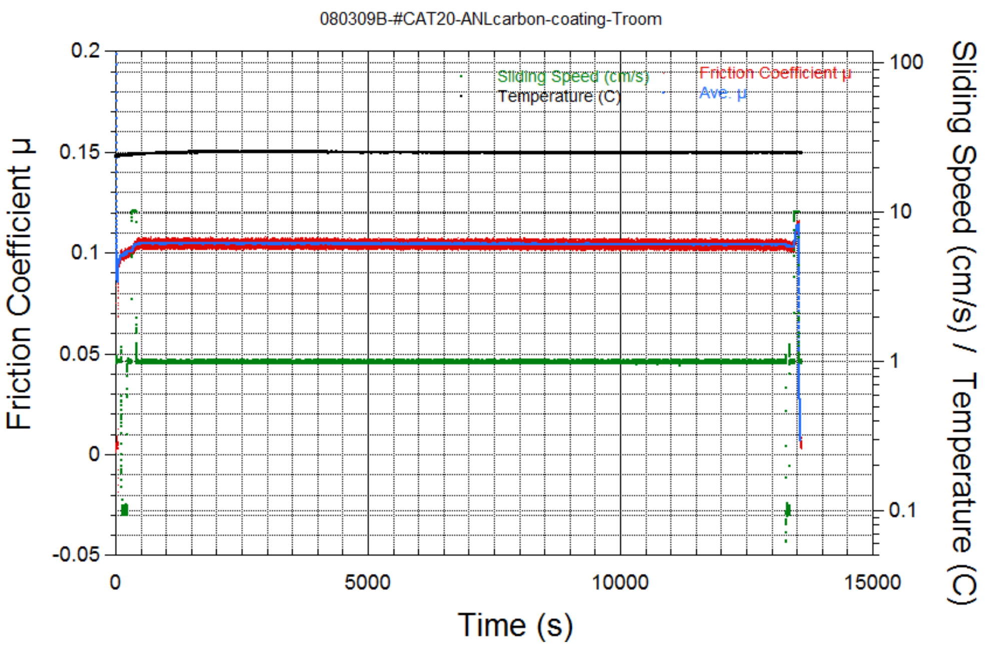


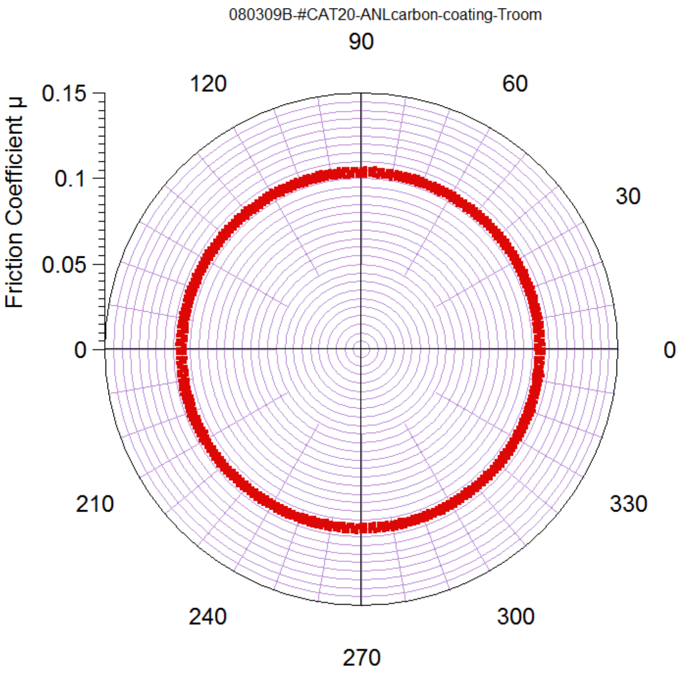




Comparison - hydrogenated DLC

- No frictional spikes
- Attributed to intrinsic low friction of the unlubricated sliding system, which maintains low friction during asperity contact in absence of micro EHL





Summary

- Measured the friction of type 52100 ball sliding against directionally ground 8620 flat using ball-on-disk sliding motion
  - average COF in oils was nearly constant at 0.12 - 0.15
  - COF largely insensitive to oil type and sliding speed
  - Higher friction occurred when ball was sliding very near parallel to grinding lay for steel
  - As much as 150% increase in COF on “spike”
  - Spike effect restricted to ± 5° off exact parallel
  - Friction spike is persistent for sliding against hard wear-resistant surface coating
  - Friction spike is absent for sliding against hard wear-resistant surface coating that has intrinsic low unlubricated COF
  - Magnitude of friction spike exhibits some variability depending on load/speed/sliding track radius/pressure of contact
  - White light interferometry showed no obvious visual changes between worn and unworn tracks, but surface metrology suggests blunting of ridges on worn specimens
- Friction spike is thought to be due to loss of micro EHL as a result of side leakage and presence of long ridges which provide no adequate convergent inlet zone for oil entrainment
- General results agree with trend observed by others of friction increase when sliding parallel to texture direction

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- *Blank slide*