# A Model for the Formation and Wear of Oxide Tribofilms on Aerospace Steels Under High-Speed Boundary Lubrication Conditions

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# Background: Motivation

## Main Gearbox Remains Helicopters' Achilles Heel

by Thierry Dubois and Mark Huber - January 1, 2013, 5:10 AM

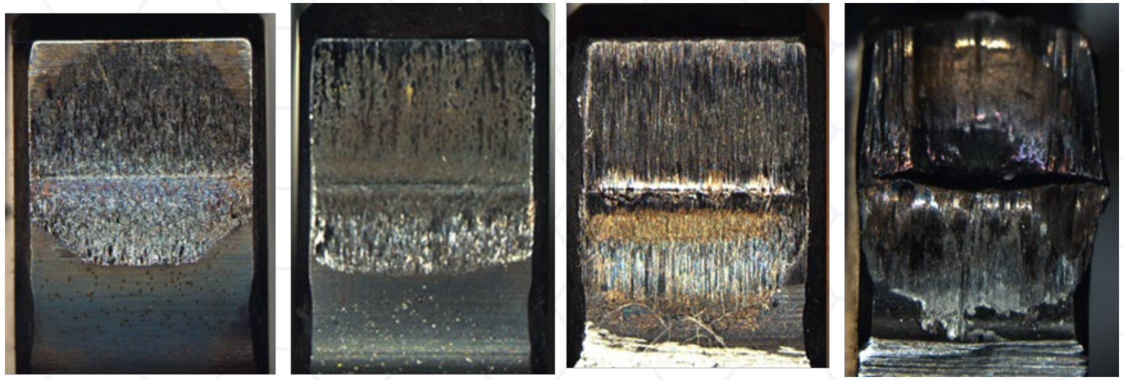
*Two Eurocopter EC225s have made controlled ditchings in the North Sea this year, due to main gearbox problems.*

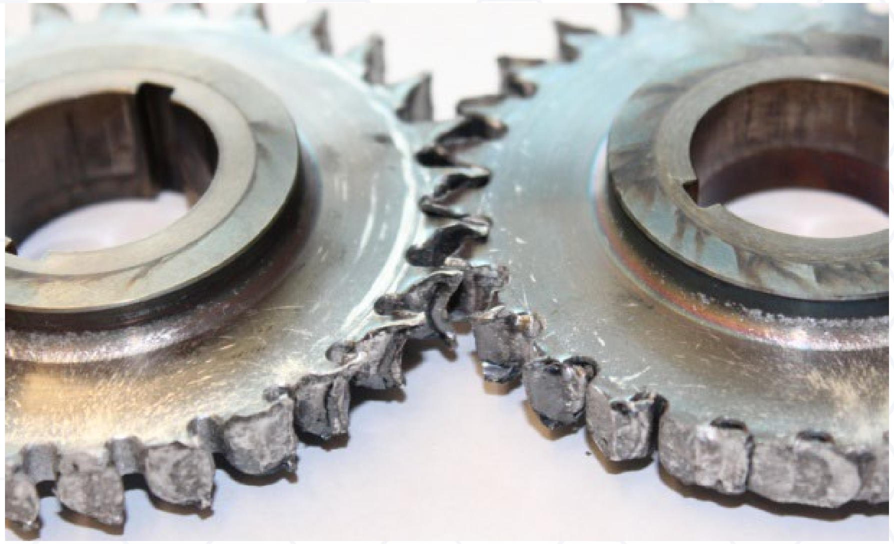
[T. Dubois and M. Huber. "Main gearbox remains helicopters' Achilles heel." *Aviation International News* 45.1 (2013)]

- Gearboxes are one of the few non-redundant vital helicopter components
- Gearbox failure is one of the leading non-human causes of helicopter crashes
- Failure due to loss-of-lubrication is one of the most common—and probably the least-understood—gearbox failure modes
- Loss-of-lubrication tests are expensive
- Extreme environment limits the potential for *in situ* experimental measurement
- Need to develop predictive modeling and simulation capabilities for loss-of-lubrication performance

# Background: Failure due to loss-of-lubrication

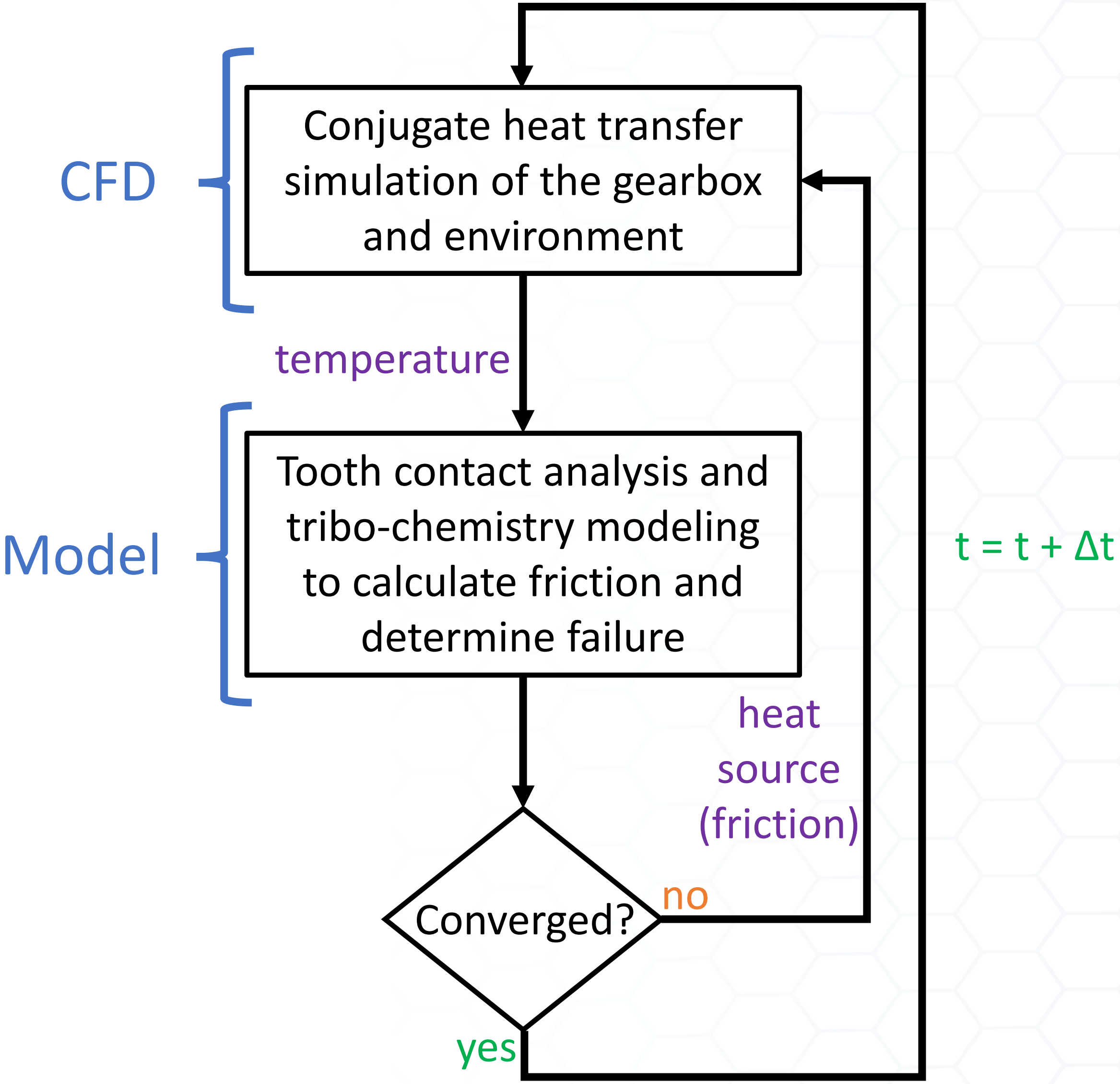
- In the context of loss-of-lubrication, gearbox failure means failure to transmit torque
- The only objective is landing safely; the gearbox will need to be replaced
- The gears in the bottom image to the right would result in a crash instead of a landing
- Landing with gear teeth looking like the top four images would be considered a success
- Barring other failure modes, ultimate failure occurs rapidly following a transition from mild oxidative wear to severe adhesive wear
- This transition is determined by a balance between oxide film formation and wear, as well as by high-temperature tempering effects





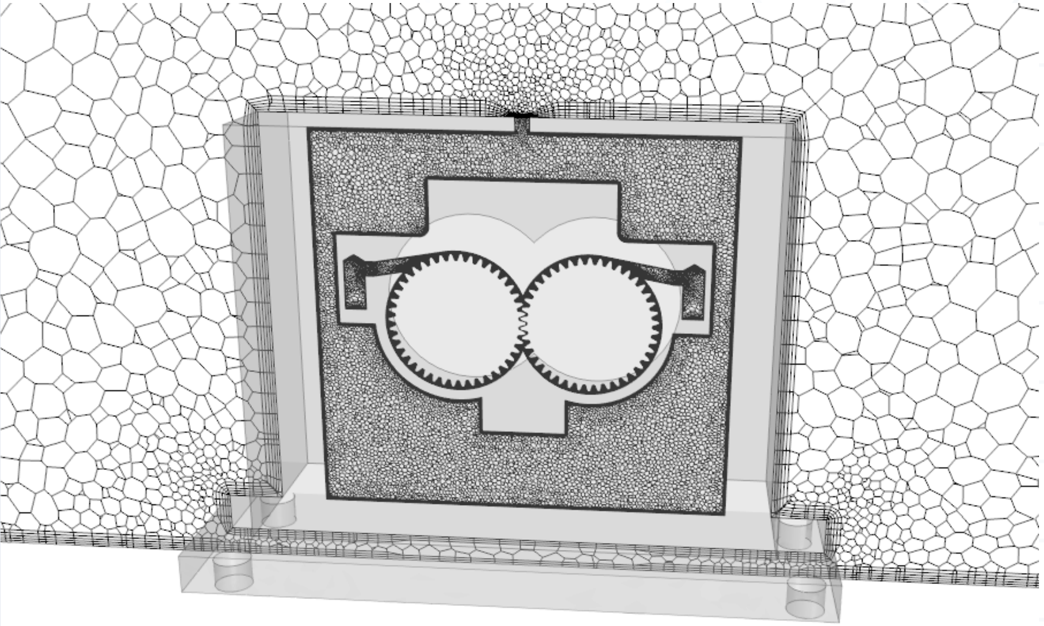
[S. Berkebile, K. Radil, N. Murthy, and M. Riggs, “Surface Finish and Phosphonium Ionic Liquid Additive to Postpone Scuffing During Starved Lubrication in High Speed Gears,” *Presented at ITS-IFToMM 2017 & K-TIS 2017*, Jeju, Korea (2017)]

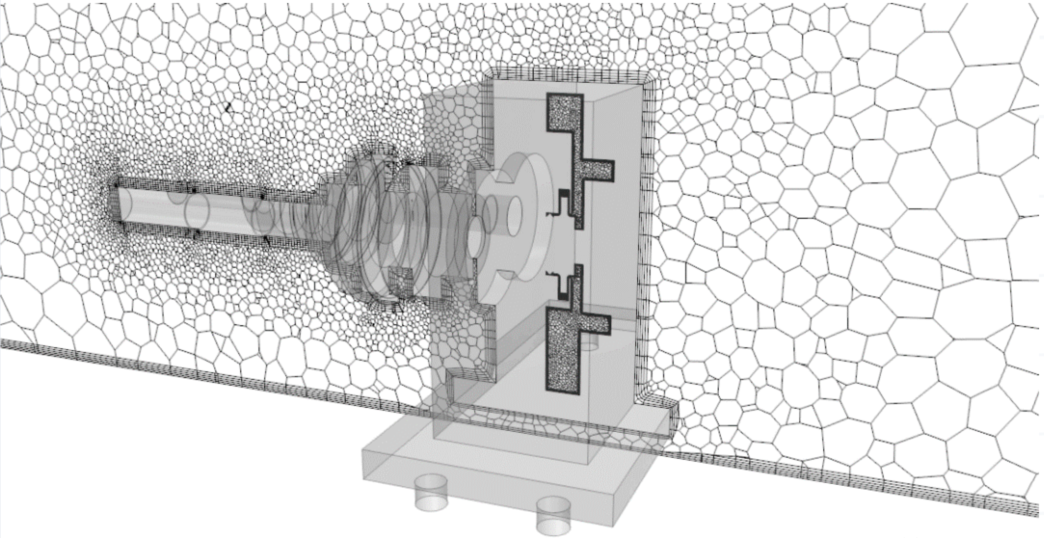
# Background: Modeling & simulation efforts to date

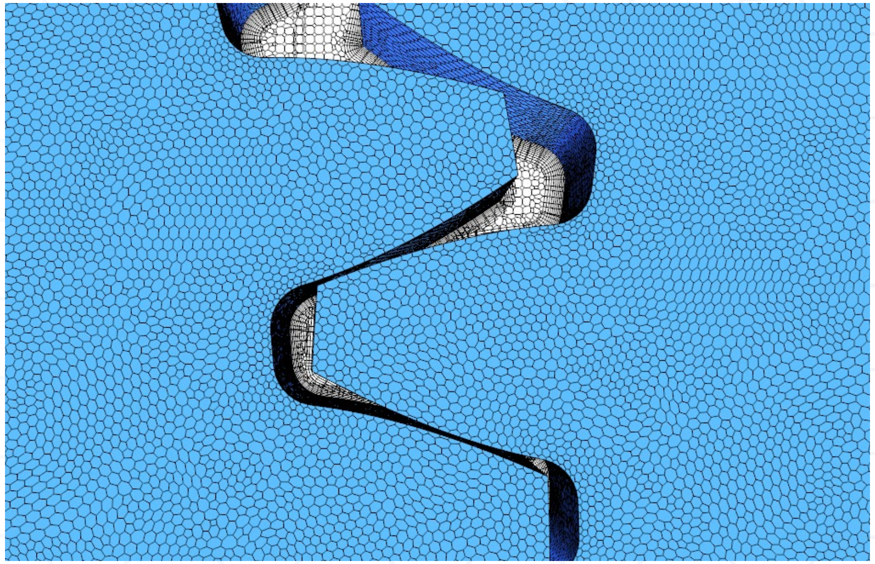


- Loss-of-lubrication failure is ultimately a tribo-chemistry/material failure mode
- Tribo-chemistry depends on the temperature and dynamics of residual lubricant flow, simulated with CFD
- Coupled CFD/gear tooth meshing friction model simulations of loss-of-lubrication have been carried out with a relatively simple meshing friction model
  - Blok criterion for lubricant failure
  - Blok criterion for tribofilm failure
- Initial results were promising, now need to make improvements to the tribo-chemistry and material modeling

# Background: Previous modeling & simulation results







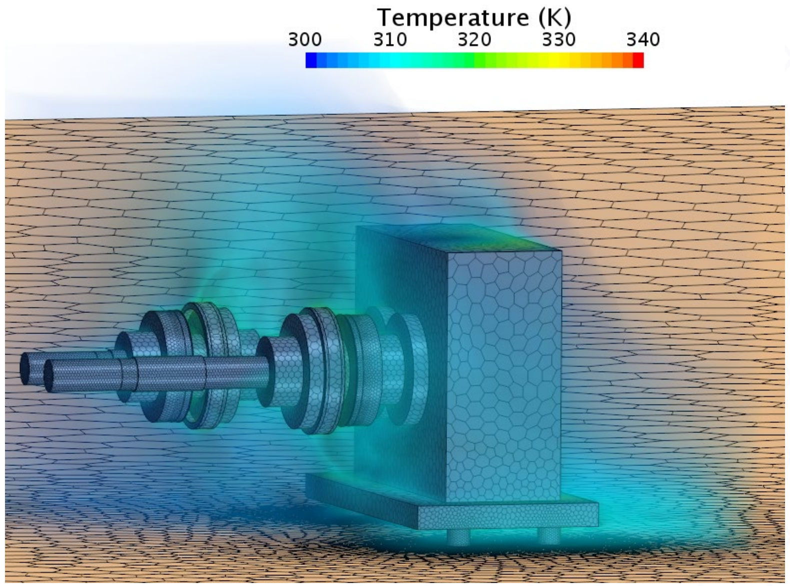
**Top left**: interior and exterior fluid domain grid slice through the axial mid-plane of the gears

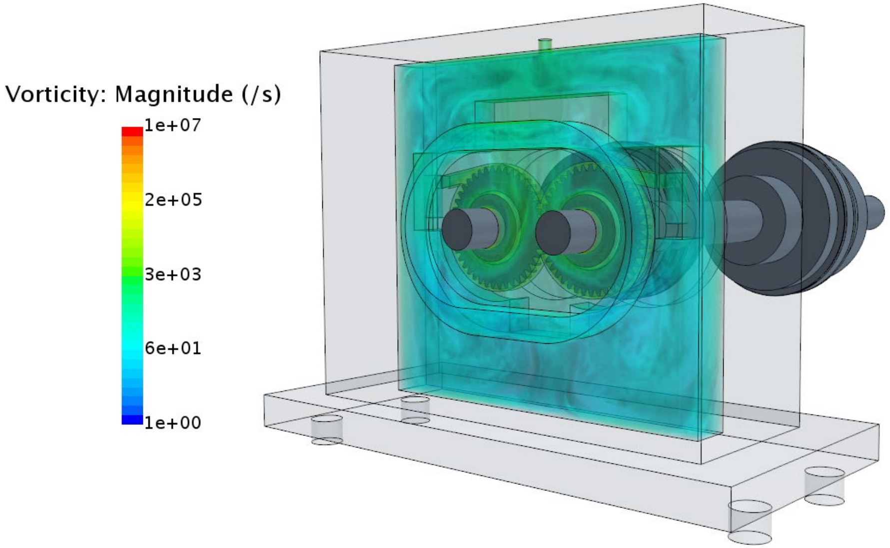
**Bottom left**: same as top left through the transverse mid-plane of the far gear

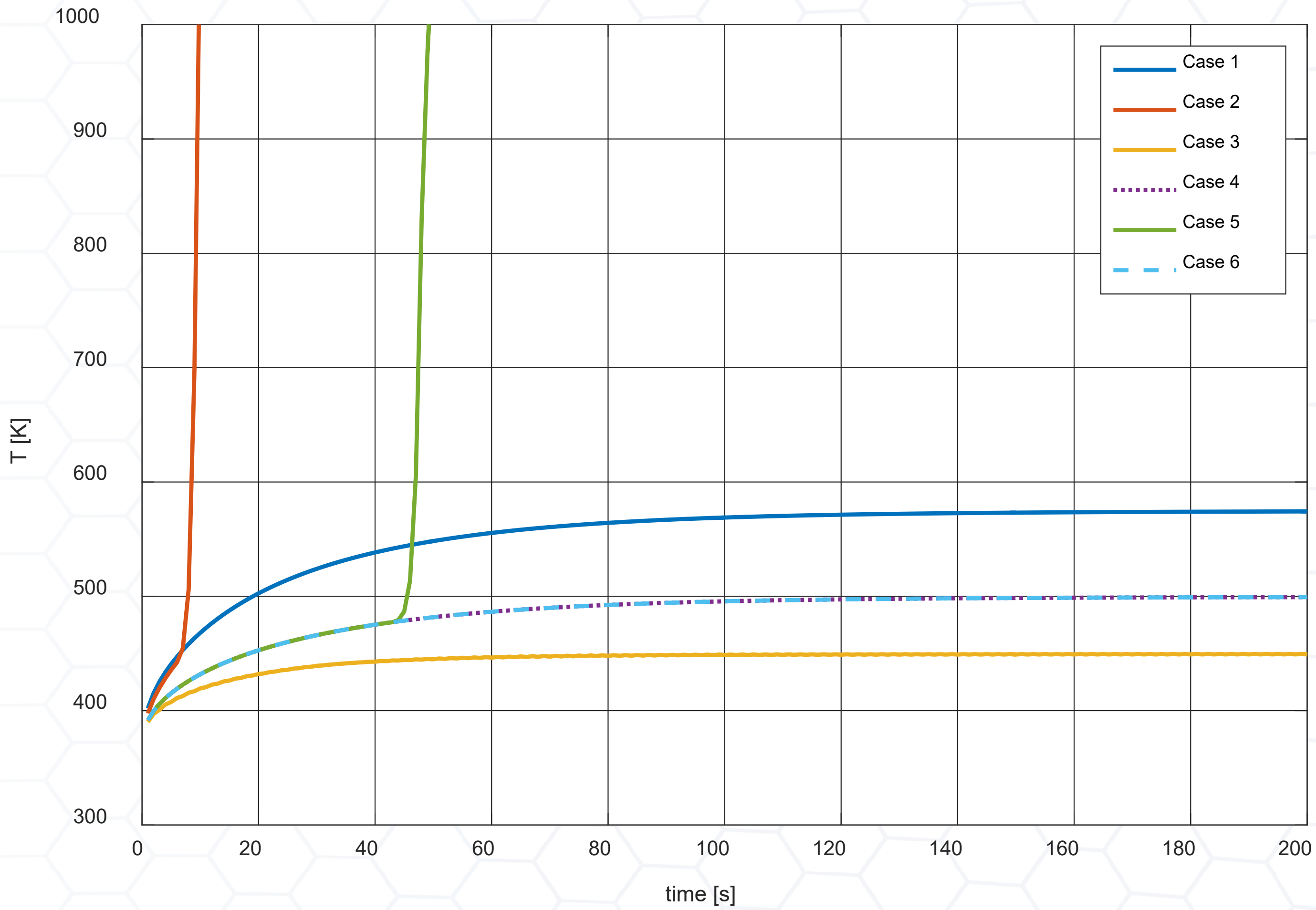
**Top right**: Surface grids on both gears where the teeth mesh, along with interior fluid domain grid slice at the axial mid-plane

- 35 million fluid cells inside gearbox
- 1 million fluid cells outside gearbox
- 12 million solid cells (conduction only)

# Background: Previous modeling & simulation results



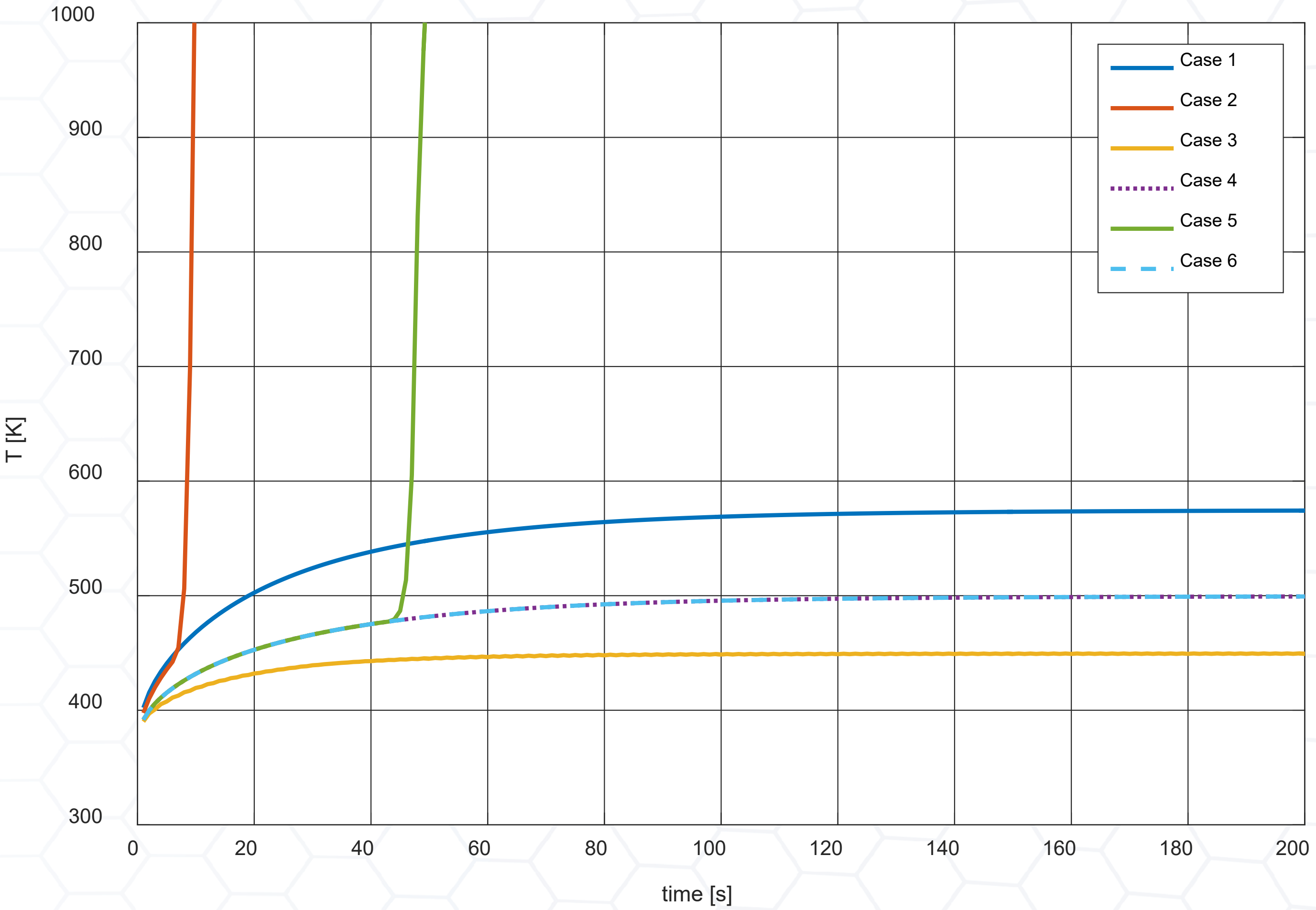




[S. McIntyre, A. Isaacson, and R. Kunz, "Loss-of-Lubrication Simulation of Spur Gears Using a CFD-Based Multi-Scale Technique and Gear Meshing Tribology Model," *Presented at the 73rd Annual STLE Meeting*, Minneapolis, MN (2018)]

# Background: Previous modeling & simulation

- Blok total contact temperature failure
  - Early failure or no failure
  - If temperature levels out, does not predict failure occurring
- Need to accommodate thermally-activated tribofilm chemistry and wear
  - Not as important in low-speed sliding
  - More important at high speeds
- Model competition between oxide formation and tribofilm wear



# Background: Experimental observations

**Ball-on-disk Tribometer**

- 5 Tests per sliding speed (Us)
- 6 Sliding speeds: 1,2,4,8,12,16 m/s
- Constant entrainment speed (Ue) = 16 m/s
- Target temperature = 393 K
- Constant Load = 100 N
- Maximum Hertzian Contact Pressure = 1.3 GPa

Loss of Lubrication Test Procedure

1. Run-in specimen for 10 min under full lubrication at Ue = 16 m/s and Us = 16 m/s
2. Change to desired sliding speed
3. Turn off oil supply
4. Wait for scuffing to occur
5. Stop test 30 s after scuffing (1 out of 5 tests)

Chemical and Topographical Analysis

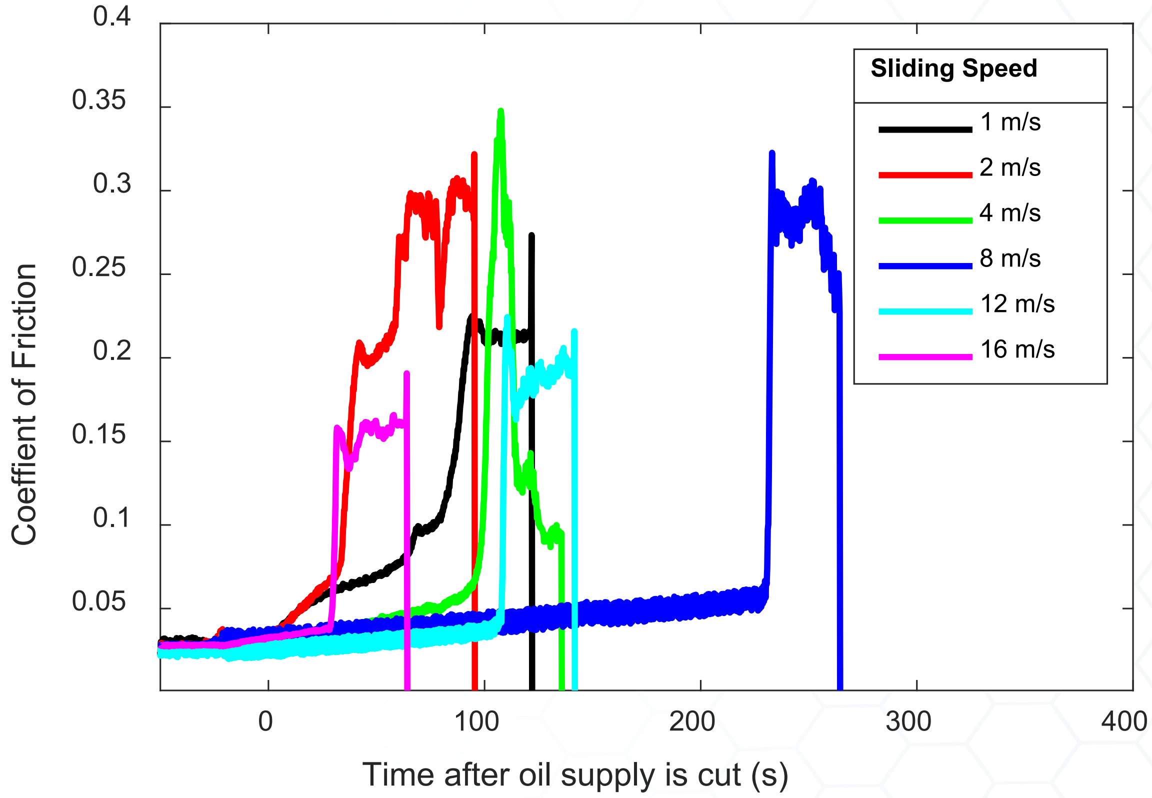
Laser Scanning Confocal Microscopy (LSM)
Scanning Electron Microscopy (SEM)
Energy Dispersive Spectroscopy (EDS)

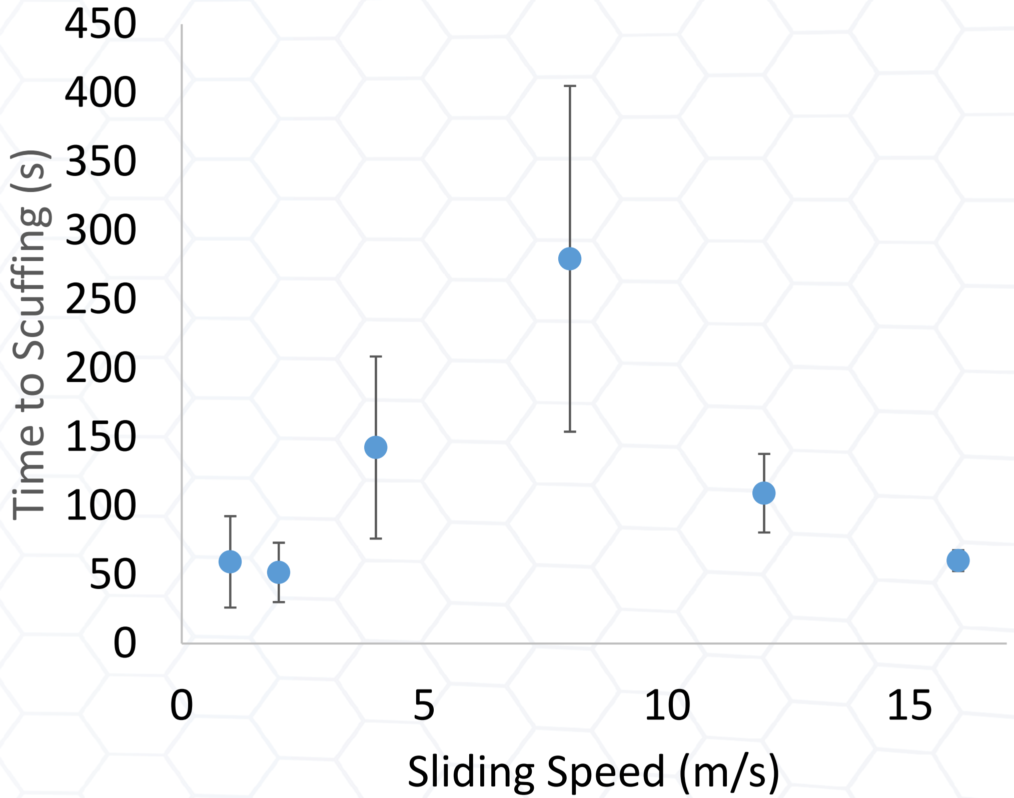
[N. Murthy and S. Berkebile, “Contact condition dependence of scuffing modes in gear steels following loss-of-lubrication.” (2018)]

# Background: Experimental observations

After oil is cut the following events occur:

1. The coefficient of friction (COF) gradually increases
2. Scuffing occurs resulting in a sudden spike in COF (and audible sound)
3. The COF stabilizes when the scuff spreads to the whole track
4. The load is relieved at the end of the test resulting in a sudden drop in COF

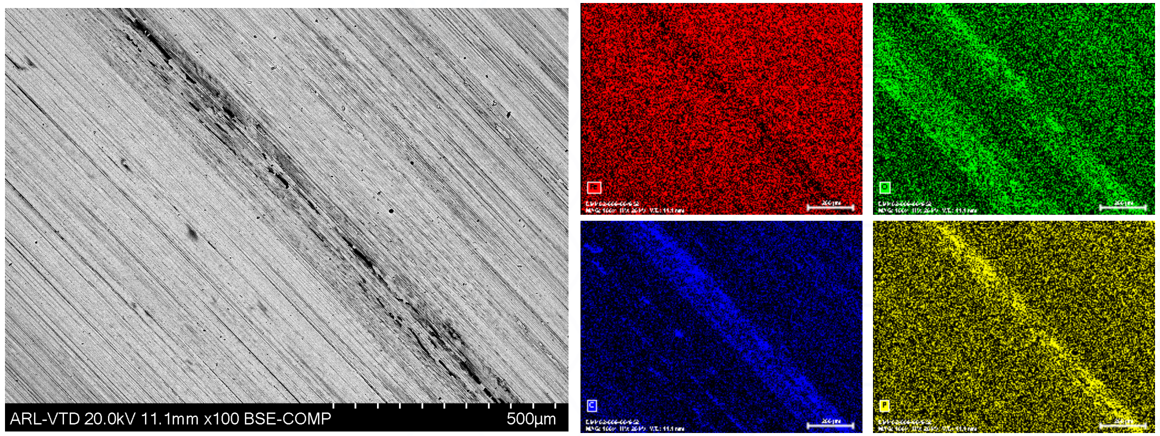




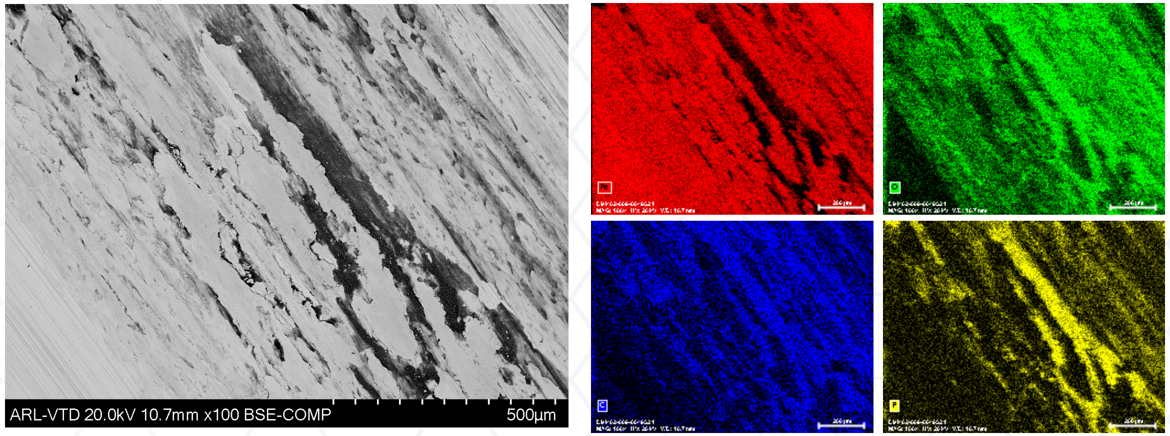
[N. Murthy and S. Berkebile, “Contact condition dependence of scuffing modes in gear steels following loss-of-lubrication.” (2018)]

# Background: Experimental observations

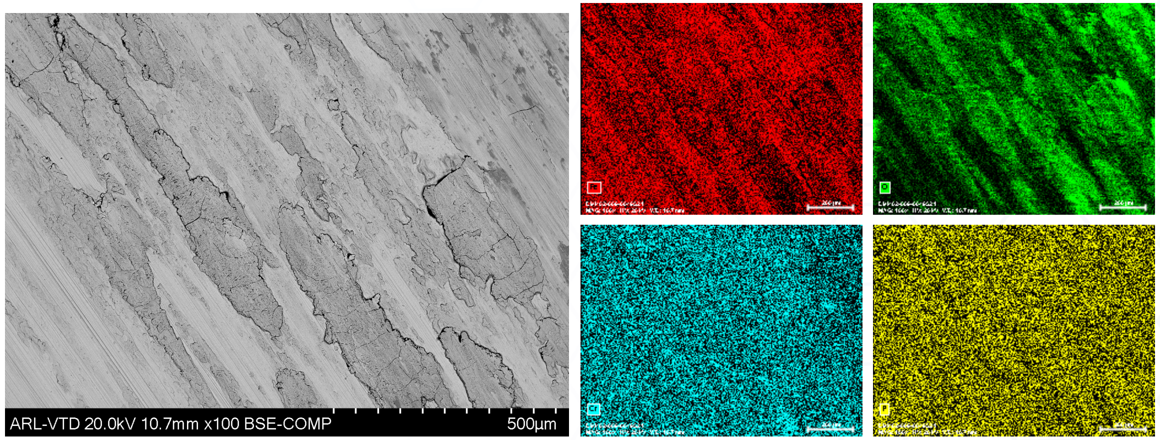
Sliding Speed = 1 m/s



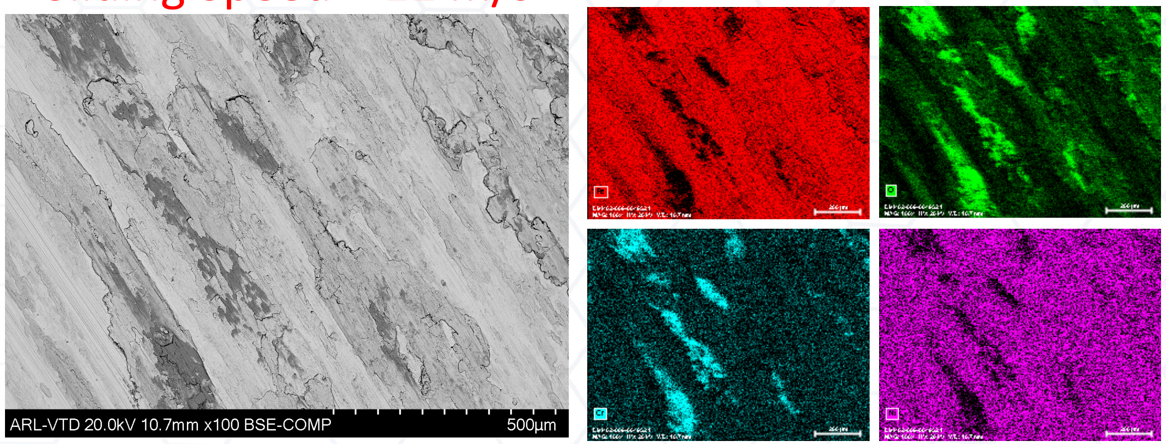
Sliding Speed = 4 m/s



Sliding Speed = 8 m/s



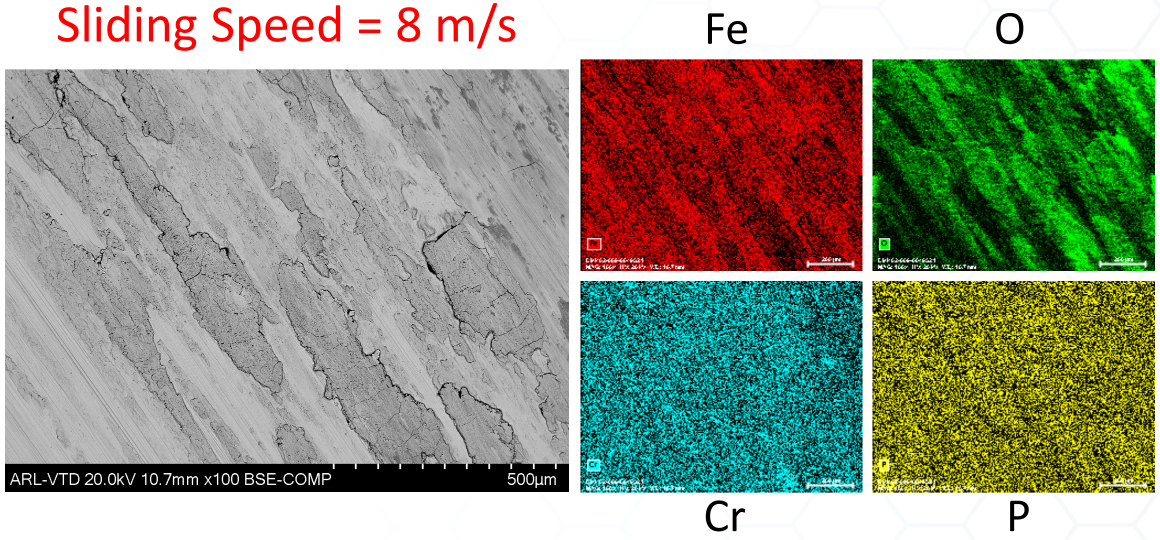
Sliding Speed = 12 m/s



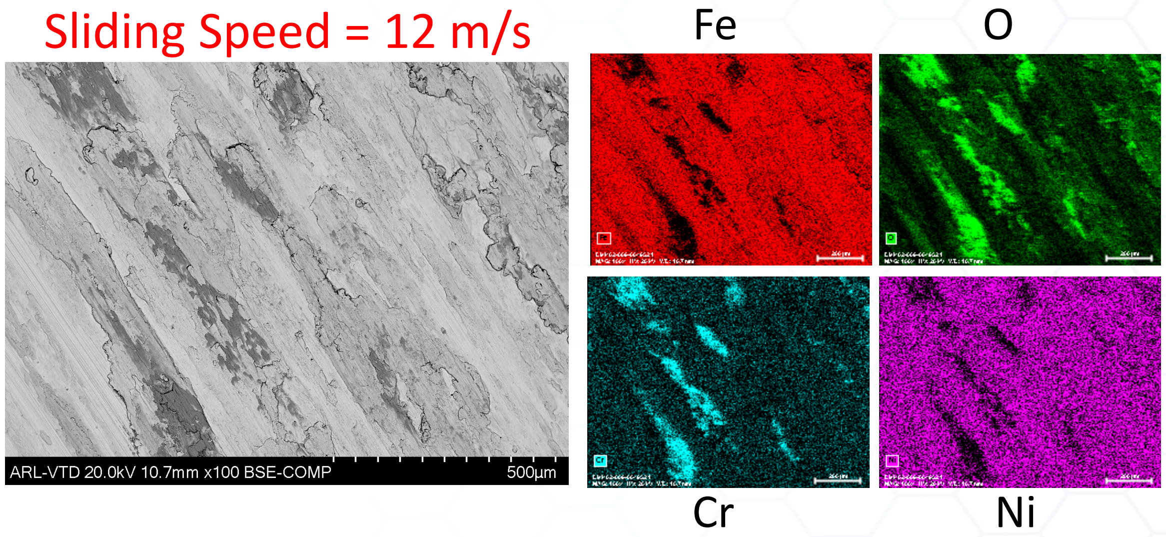
[N. Murthy and S. Berkebile, “Effect of sliding speed variation on scuffing following loss-of-lubrication,” (2018)]

# Background: Experimental observations

Sliding Speed = 8 m/s



Sliding Speed = 12 m/s



- EP additives appear to be active at low speed
- By 8 m/s (longest life), little EP additive effect
- 8 m/s case appears to depend on the dynamics of Fe-O films, little alloying element migration
- At higher speeds, alloying elements migrate, need to consider Fe-Cr-O (though short life here)
- Will model Fe-O tribofilms first since they appear to have the largest effect here
- Disclaimer: largest effect for AISI 9310 under these contact conditions, maybe different otherwise

# Modeling wear due to sliding

- Modeling wear due to asperity fatigue (mild-moderate)
- Not modeling severe wear due to bulk plastic flow
- Archard’s model for the wear volume is relevant
  - $V$ is the volume of the worn material ($\mathrm{m}^3$)
  - $p$ is the material flow pressure (Pa)
  - $L$ is the applied normal load (N)
  - $d$ is the sliding distance (m)
  - $k$ is the wear coefficient (dimensionless)
- Need material flow pressure (difficult material property)
- Need to model wear coefficient–friction coefficient relation

$$V = k\,\frac{L\,d}{p}$$

[J. Archard and W. Hirst. “The wear of metals under unlubricated conditions,” *Proc. Royal Soc. London. Series A*. 236 (1956) 397-410]

# Modeling wear coefficient vs. friction coefficient

- Very few existing models
- Using Beheshti and Khonsari model
- Uses continuum damage mechanics to predict number of load cycles to failure for a given asperity
- Wear coefficient is 1/3 the probability that an asperity forms a wear particle
- Probability $P$ of an asperity forming a wear particle is the reciprocal of $N$ the number of cycles to failure
- Calculates $N$ by integrating the damage dynamics until the damage $D$ exceeds the critical damage $D_C$. $N$ is the number of cycles when that occurs
- Considers failure only due to shear stress (sliding), not due to normal stress (rolling contact fatigue)

$$k = \frac{P}{3} = \frac{1}{3N}$$

$$\frac{dD}{d\epsilon} = -\frac{\sigma_\infty}{\psi_D}$$

$$\Delta\epsilon = \underbrace{\frac{\Delta\sigma}{E(1-D)}}_{\text{elastic}} + \underbrace{\left(\frac{\Delta\sigma}{K(1-D)}\right)^M}_{\text{plastic}}$$

[A. Beheshti and M. Khonsari. "A thermodynamic approach for prediction of wear coefficient under unlubricated sliding condition," *Tribol. Lett.* 38 (2010) 347-354]

# Calculating the rate of wear

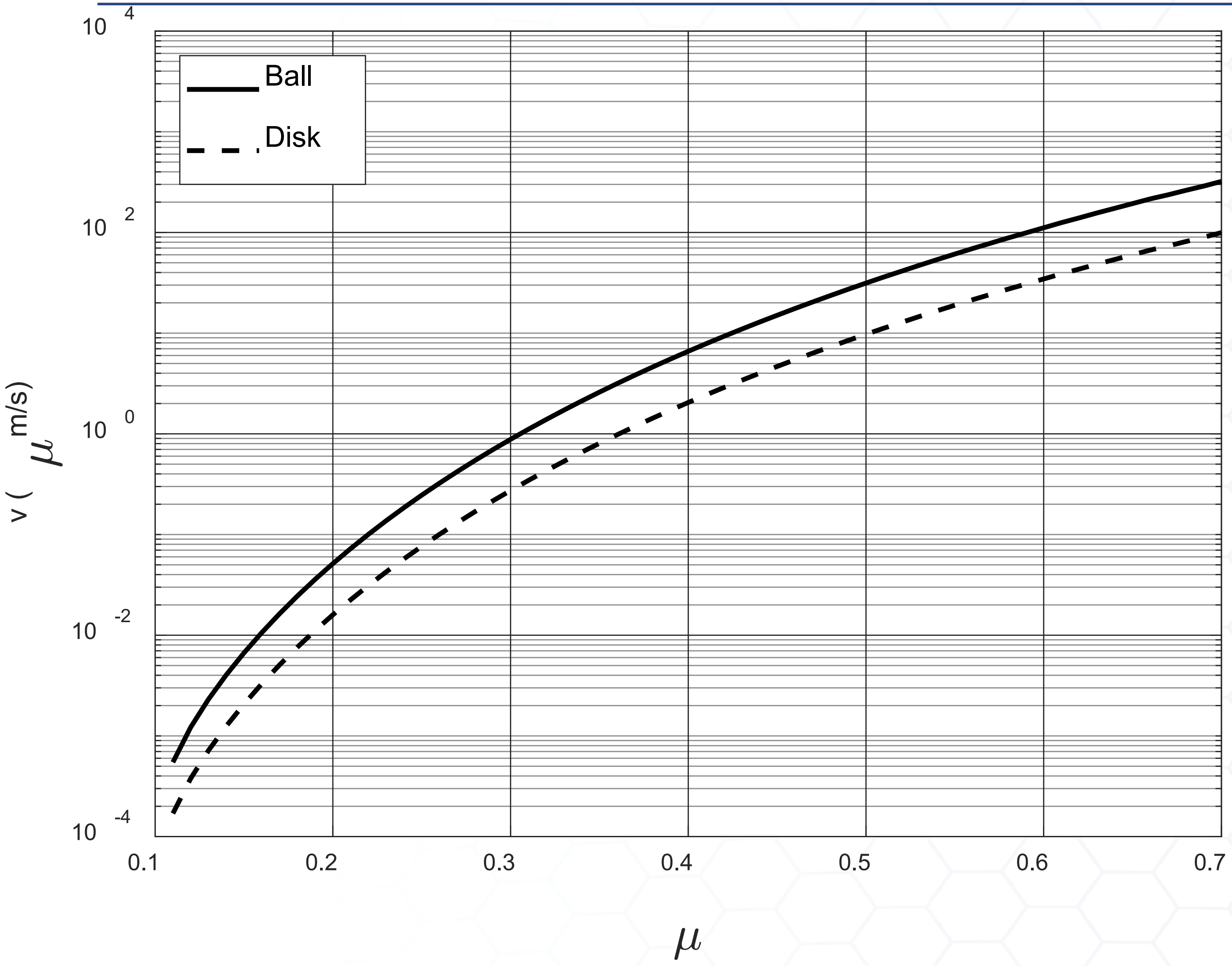
- Hertzian contact mechanics for contact area $A$ and radius $R$
- For each pass through contact, sliding distance is twice the contact radius
- Given load $L$, wear coefficient $k$, and material flow pressure $p_f$:

$$V_{pass} = k\,\frac{2LR}{p_f}$$

$$\Delta x_{pass} = \frac{V_{pass}}{A}$$

- The surface recession rate of the ball is $\Delta x_{pass}$ times its rotation rate in rev/s
- The surface recession rate of the disk is $\Delta x_{pass}$ times its rotation rate in rev/s

# Wear rate vs. friction coefficient results



Predicted wear rate vs. coefficient of friction for the WAM
16 m/s entraining velocity, 8 m/s sliding velocity

- Picked reasonable steel properties
- Plotted rate of surface wear (micron/second) for ball and disk vs. friction coefficient
- Predicts zero wear below $\mu \approx 0.1$
  - Properties should probably be a blend of steel & oxide
  - Model doesn’t consider rolling contact fatigue of asperities
  - How valid is Archard’s assumption that all asperities contact at the flow stress for these conditions?
- These results look reasonable (maybe a bit high); we’ll use them

# Modeling oxide tribofilm formation

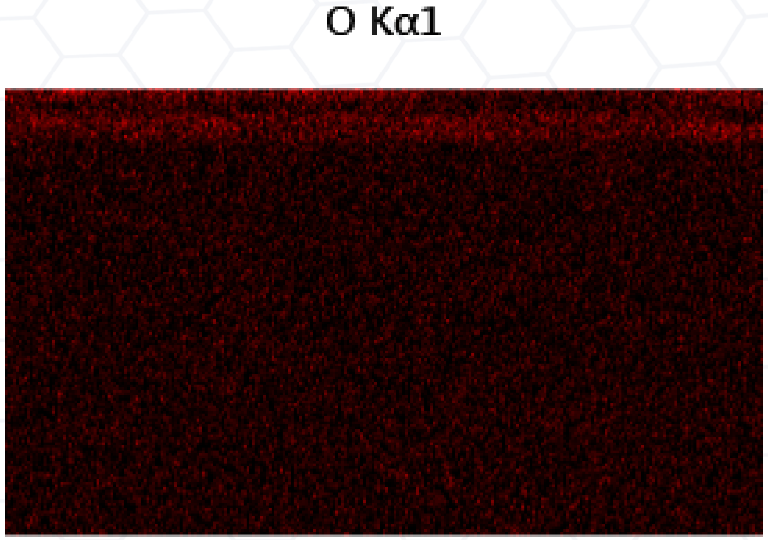
- Expect oxide scale thickness about 1 μm
    - From what we've seen on scuffed samples
    - For more evidence, Cody held Pyrowear 53 at 400° C for 15 minutes, did SEM/EDS
    - Found 1-2 μm scale thickness
- For this range, oxide formation rate is limited by the diffusion of oxygen through the scale
- Using Wagner model:

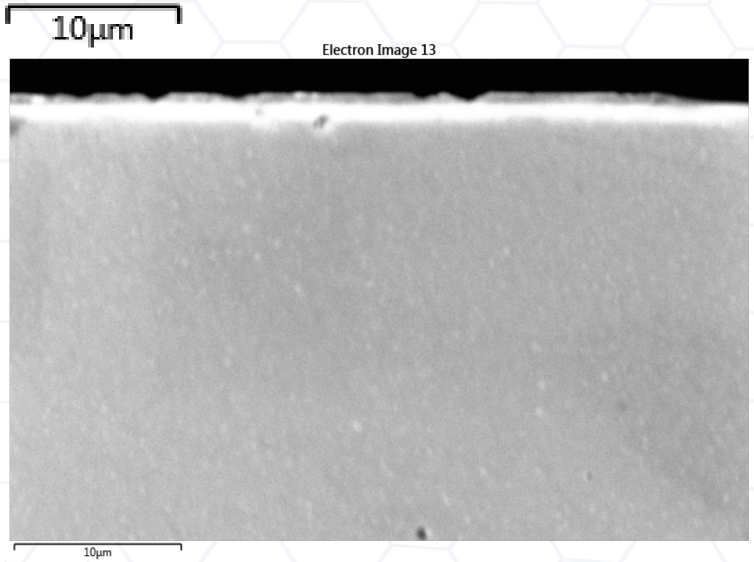
$$X^2 = 2\,k\,t$$

$$\frac{dX}{dt} = \frac{k}{X}$$

$$k = k_0 \exp\left(\frac{-Q}{RT}\right)$$

- [1]Using Q = 164000 J/(mol-K), $k_0$ = 1.0701 $cm^2/s$
- Net oxide growth rate is baseline rate due to operating temperature plus the increment due to each pass through contact at the contact temperature times the rotation rate





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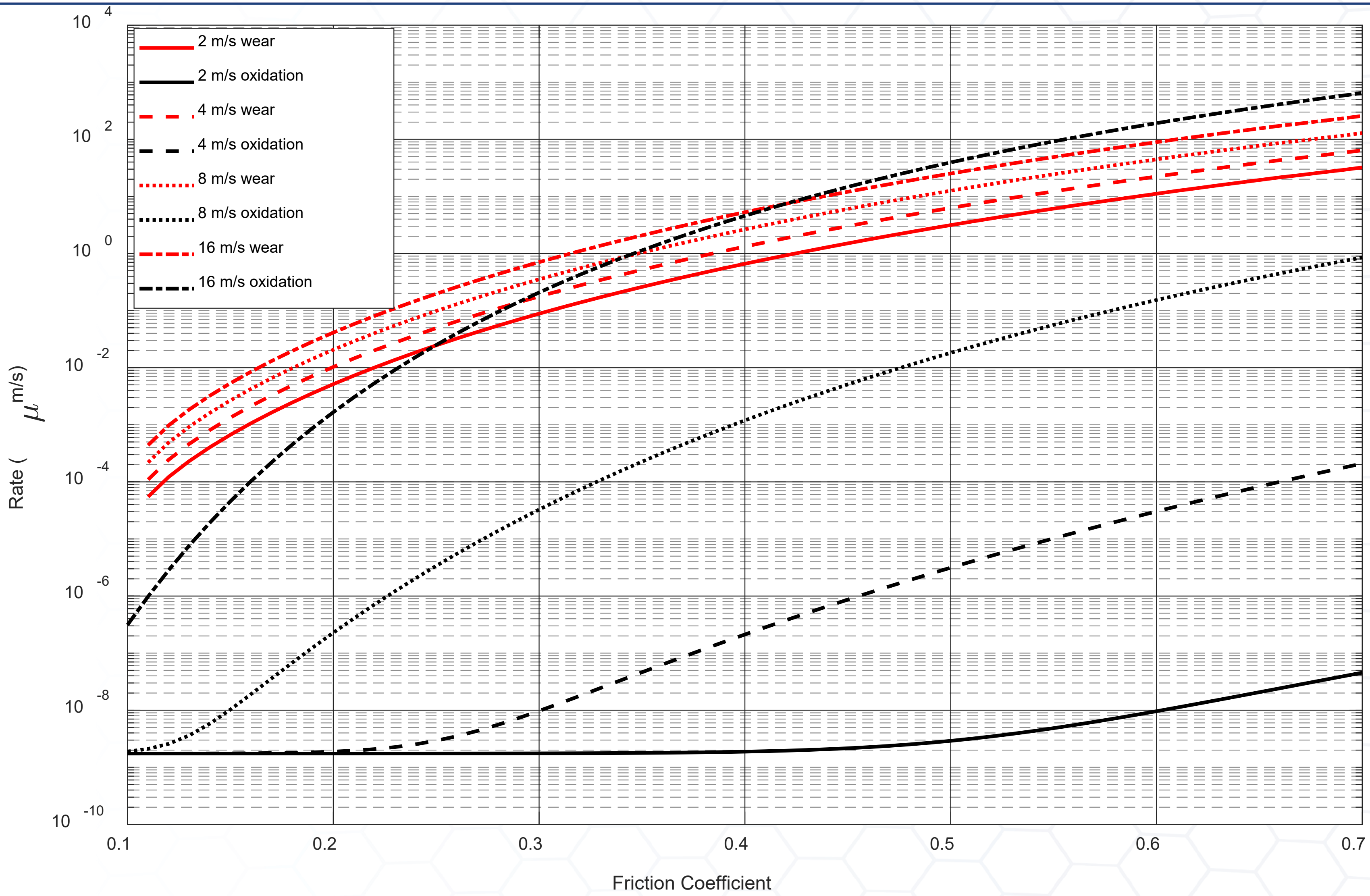
[1] [D. Young. *High Temperature Oxidation and Corrosion of Metals*. Elsevier (2016).]

# Modeling oxide tribofilm formation



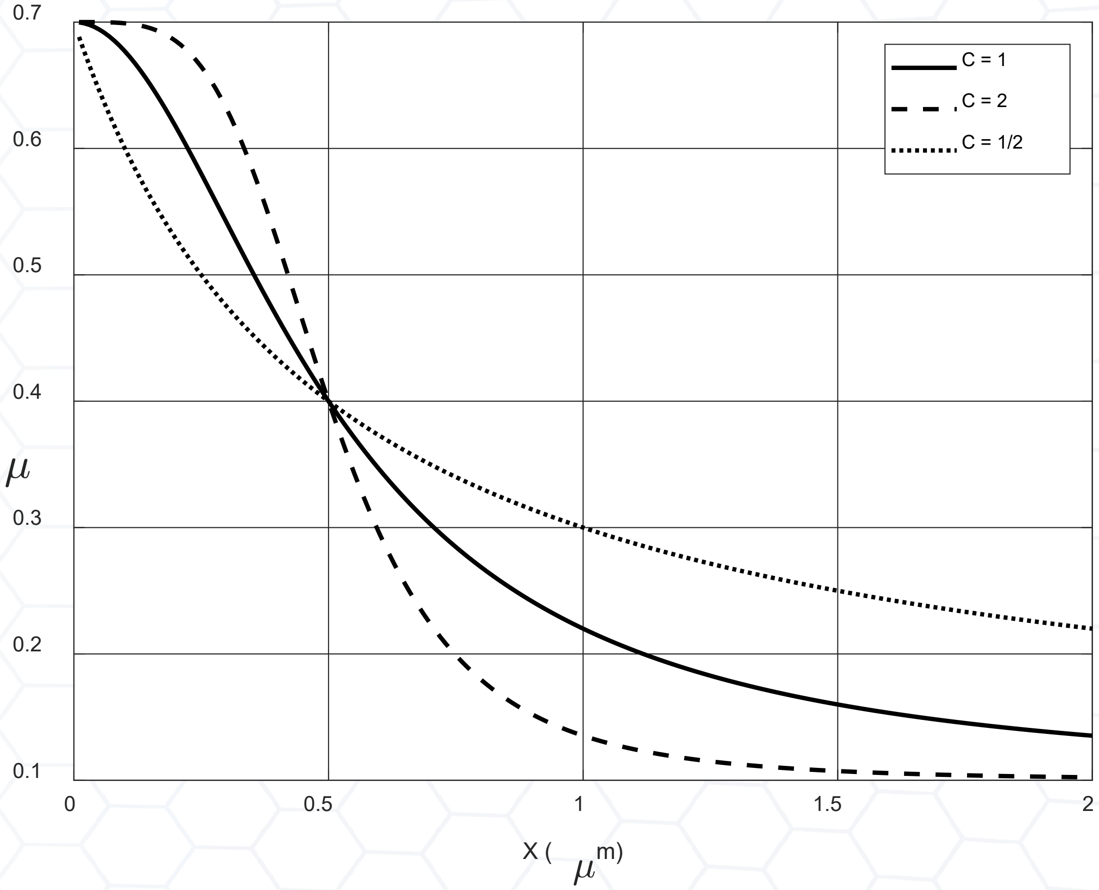
Oxide scale growth and wear rates vs. μ for oxide scale thickness of 100 nm

# Modeling friction coefficient

- Need to transition from $\mu_{oxide} \approx 0.1$ to $\mu_{metal} \approx 1.0$ as oxide thickness approaches zero, want to do this smoothly
- Decided to use:

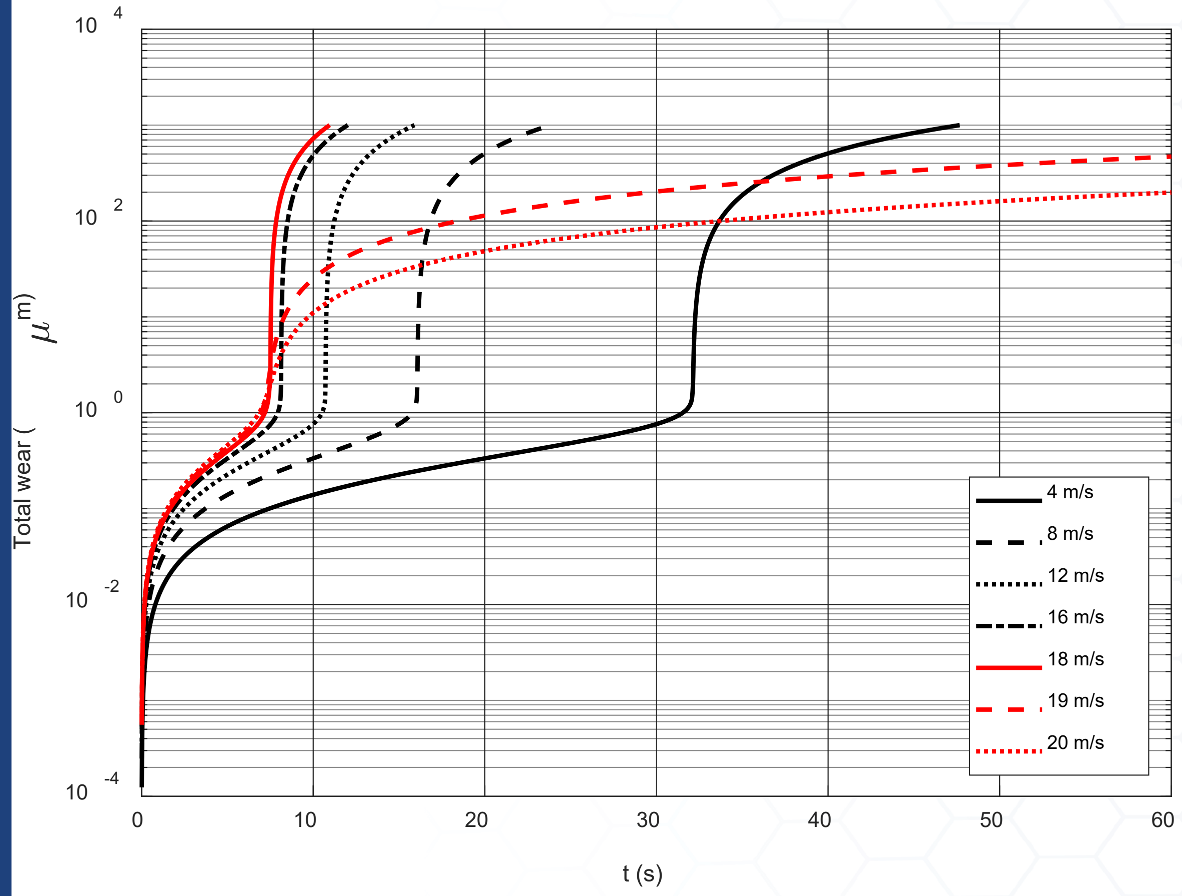
$$\mu(X) = \mu_{metal} + \frac{1}{2}(\mu_{oxide} - \mu_{metal})\left(\tanh\left(C \ln\left(\frac{X}{X_0}\right)\right) + 1\right)$$

- C and $X_0$ are model parameters
  - $X_0$ specifies the midpoint of the transition
  - C determines how fast the transition happens
- Seems more reasonable that doubling/halving the thickness would do about the same thing than adding or subtracting the same amount to the thickness as it gets closer to zero

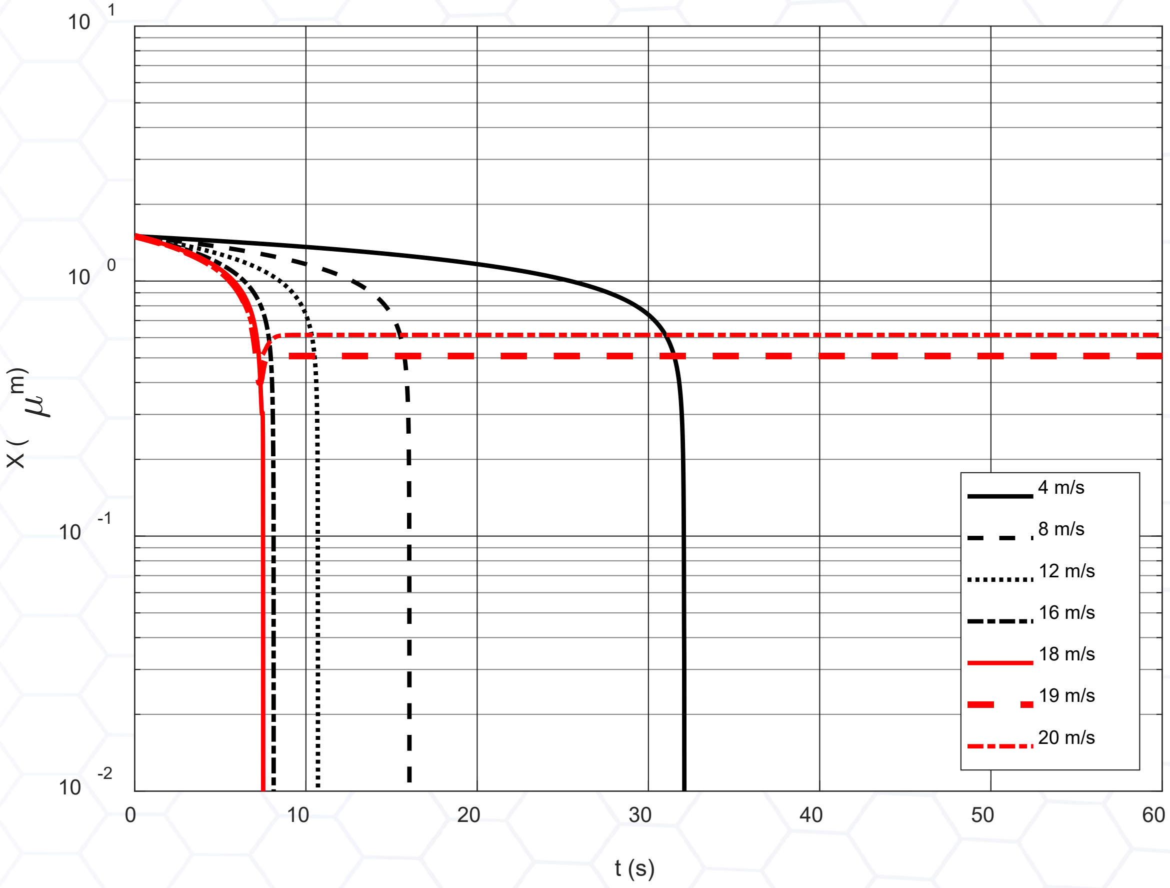


Friction coefficient vs. oxide scale thickness for $X_0$ = 0.5 µm
$\mu_{metal}$ = 0.7, $\mu_{oxide}$ = 0.1

# Transient simulation results

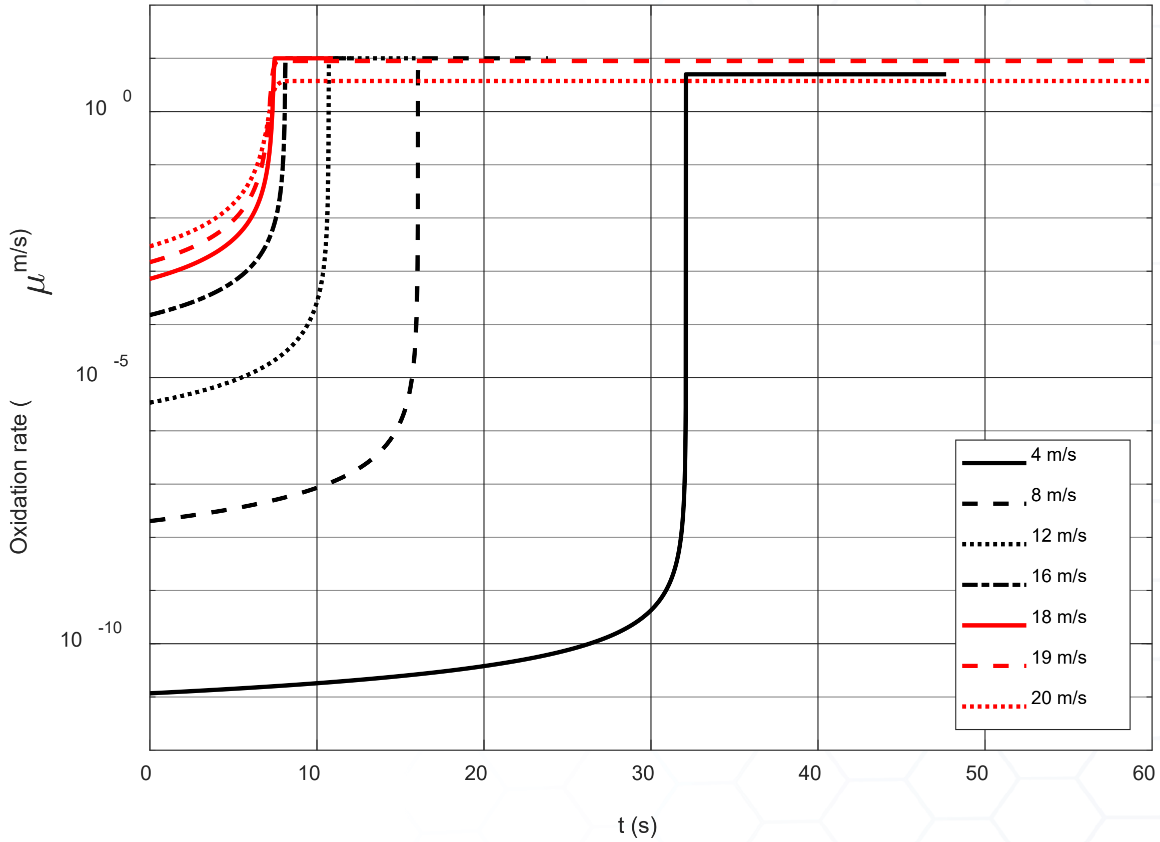


Total wear vs. time

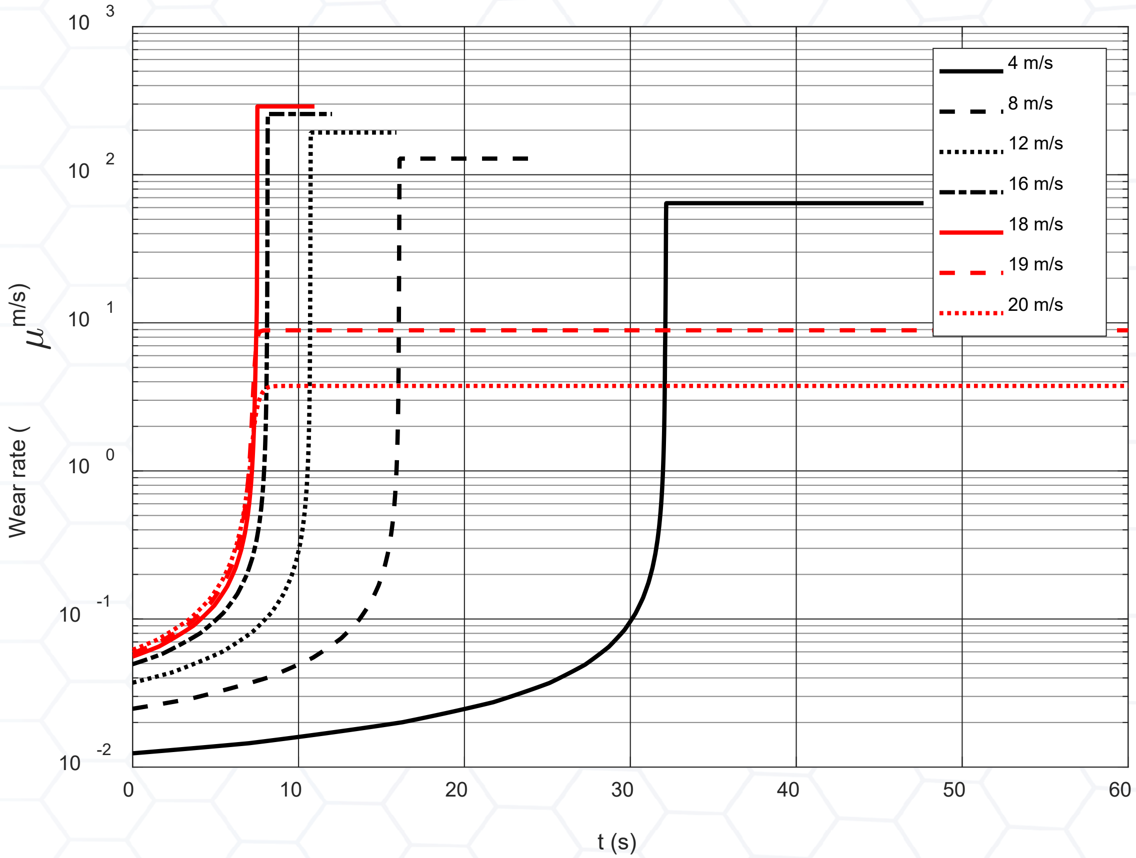


Oxide scale thickness vs. time

# Transient simulation results

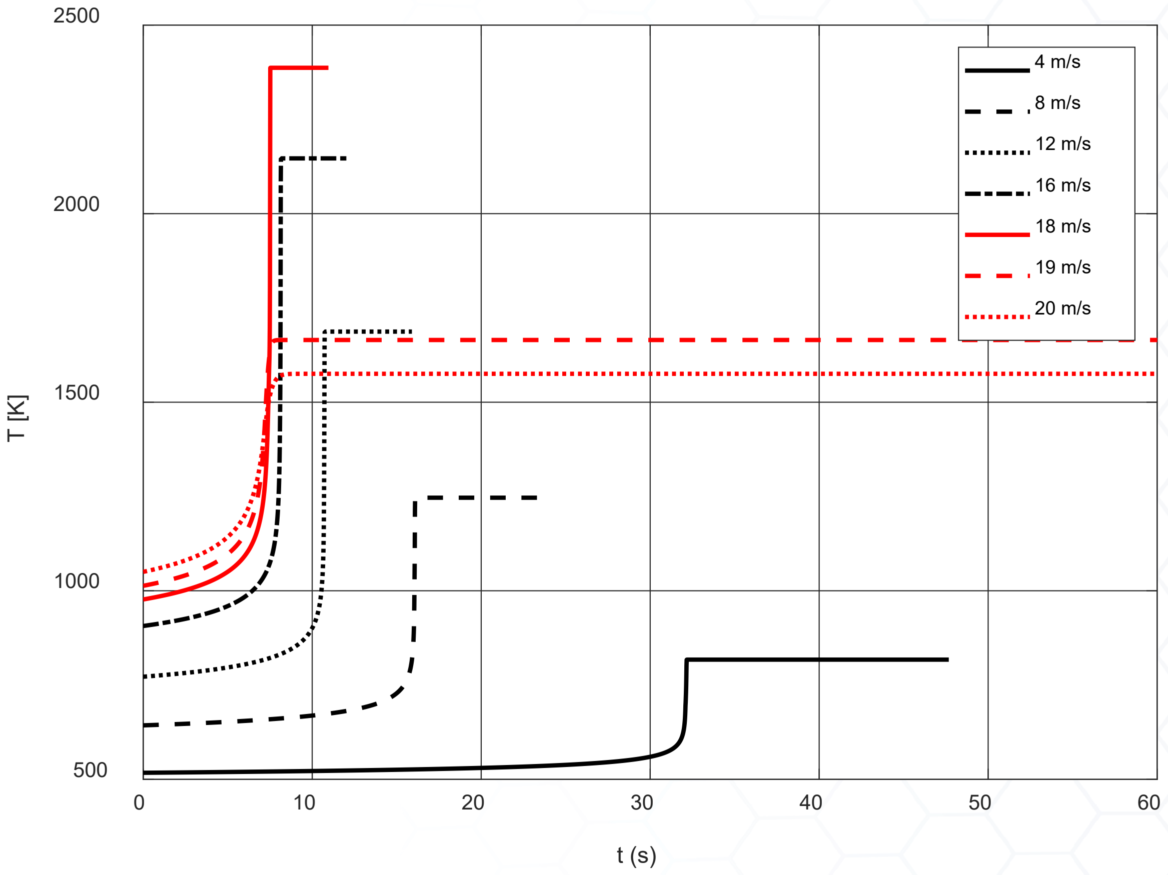


Oxide scale growth rate vs. time

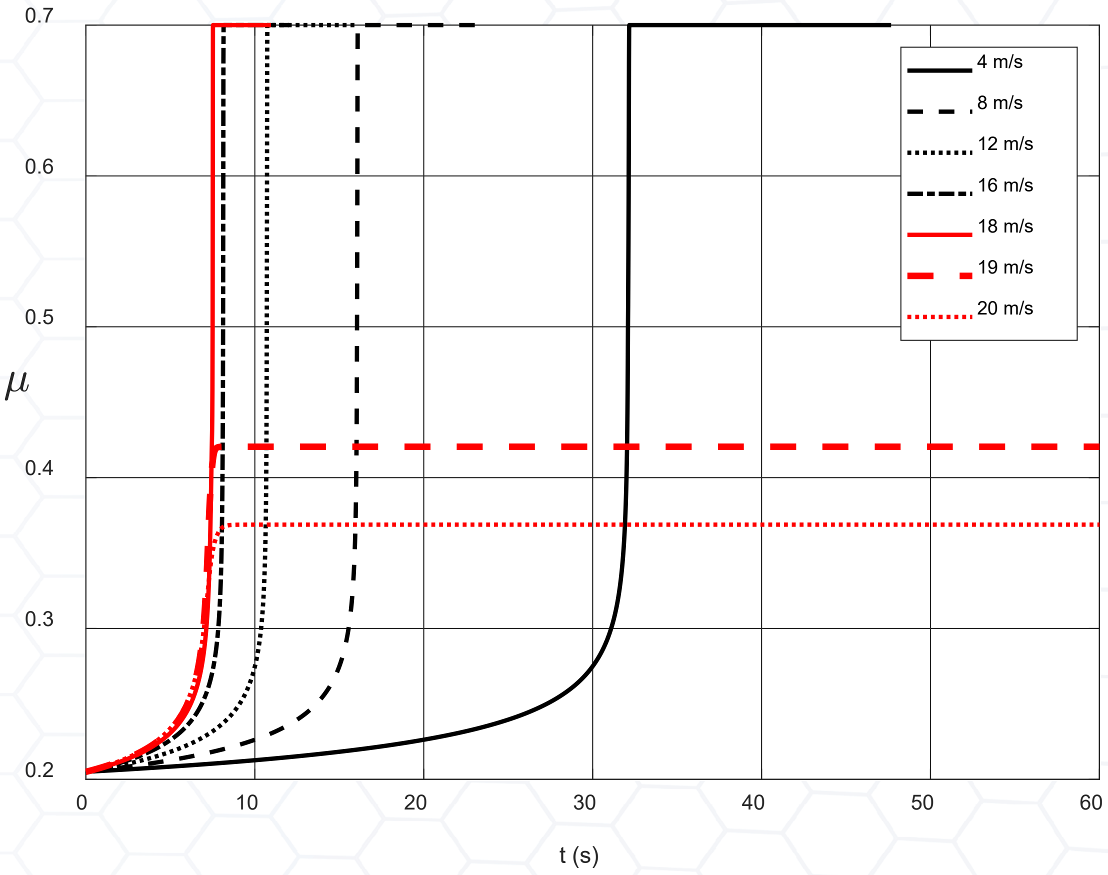


Wear rate vs. time

# Transient simulation results



Contact temperature (bulk + flash) vs. time



Friction coefficient vs. time

# Observations

- Model can accommodate wear-oxidation equilibrium now
- Current wear model:
  - Under-predicts wear when friction is low
  - Under-predicts wear when oxide film is thick
- Need to accommodate different material properties (strength, endurance limit, etc.) of oxide vs. metal
- Wagner's parabolic growth model is not applicable for very thin scale; predicts unbounded oxidation rate as X→0
- Need to accommodate the transition to linear growth as other factors limit the scale growth rate as X→0
  - Oxygen ion/electron transport
  - Partial pressure of oxygen

# Next steps

- Transition from parabolic to linear growth as scale thickness →0
- Incorporate oxide properties into wear model
- Add rolling contact fatigue/pitting-type wear for low friction conditions

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