Experimental Investigation of Infiltrated Water and Ice on Friction Pendulum Bearings

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Project Sponsor: Alaska DOT&PF

University of Nevada, Reno

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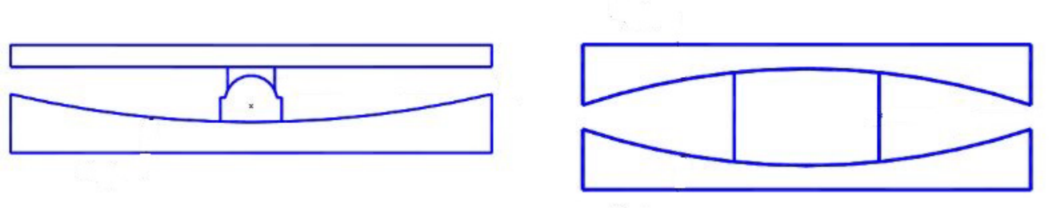
Motivation for the Project

Conceptual cross section of FPS single pendulum and double pendulum bearings



- Water penetration to inside cavity of FPS bearings observed during routine bridge inspections in Alaska.
- Anecdotal evidence supports that the problem extends beyond Alaska (WSDOT, New York TappanZee Bridge).
- Cold temperatures in Alaska's northern regions suggests ice frozen in the bearings much of the year. Will the bearings move properly in an earthquake?
- Little information to evaluate performance of FPS bearings after 15 years of service.

Water contamination in FPS bearing
Source: Alaska DOT&PF

As-received condition of formerly in-service FPS bearing

Test Program

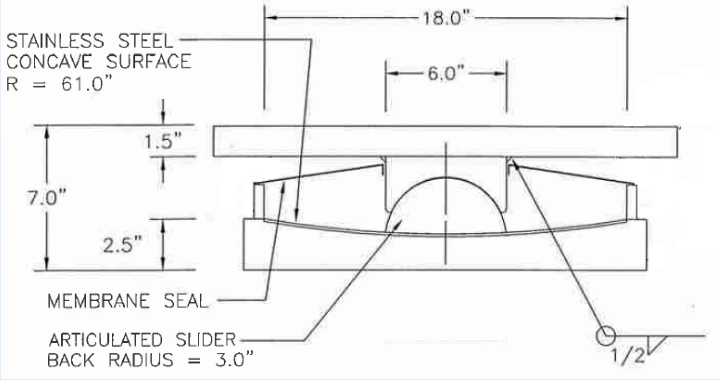
Objectives

1. Characterize response of FPS bearings under water and ice contamination
2. Evaluate “ice breakaway force” or resistance to lateral movement in an ice-filled bearing

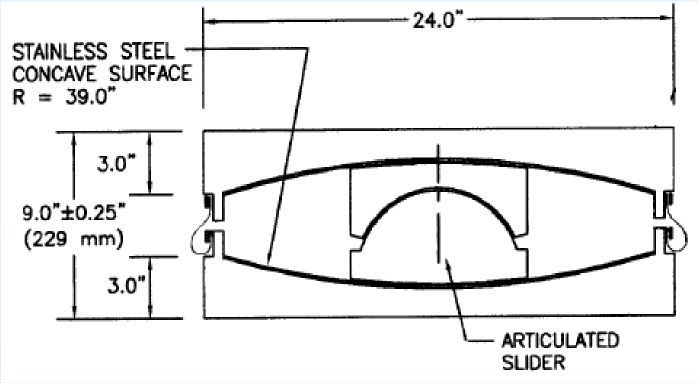
Bearings Selected

- ***Robertson***: New single pendulum bearing (SPB) to represent Robertson River Bridge
- ***Old Susitna 1 and 2***: Formerly in-service double pendulum bearings (DPB) removed from Susitna River Bridge
- ***New Susitna***: New DPB to represent Susitna River Bridge

	Robertson	Susitna
Axial Force (kip)	18	100
Pendulum Length (in)	61	74
T_b (sec)	2.5	2.75
Target Friction μ	0.1	0.075
Design Disp (in.)	4.05	3.72
T_{eff} (sec)	1.58	1.74
Disp Capacity (in.)	6	11



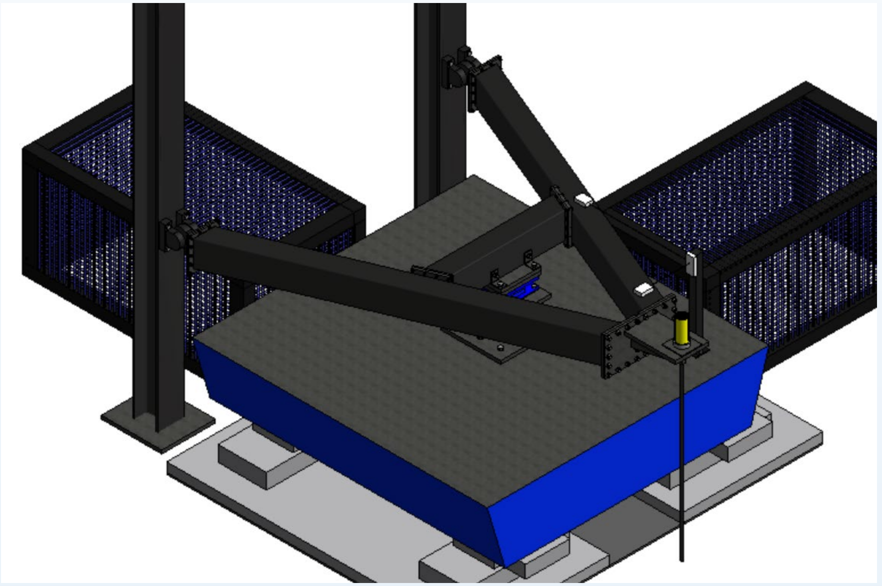
Robertson



Susitna

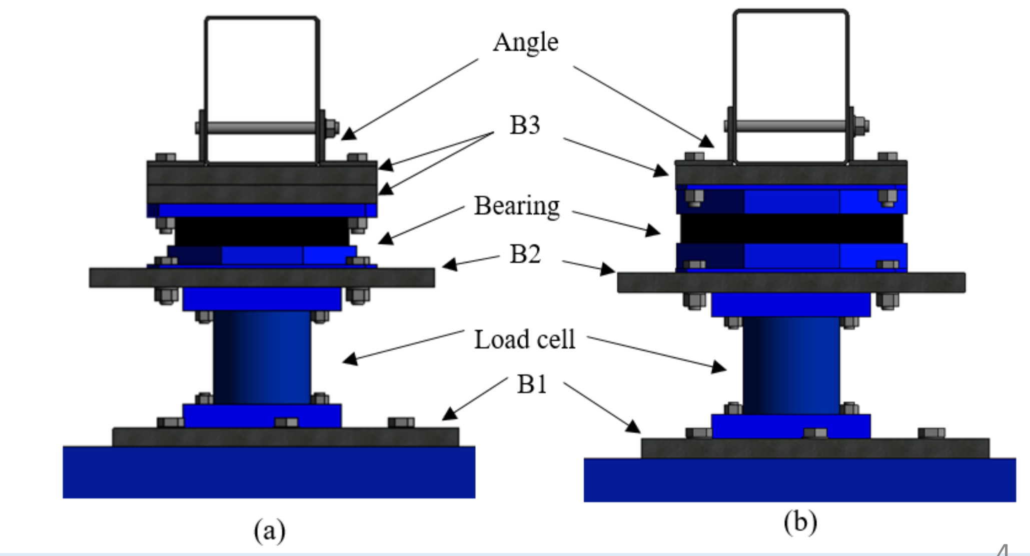
Test Setup

- Bearing was supported by shake table below and stabilizing A-frame above
- Shake table imposed cyclic loading to the bearing
- A-frame was pinned to columns off the table and rotated down over the bearing
- Axial force imposed by hydraulic ram on opposite end
- Bearing forces measured by a load cell and displacements by string pots



3D Rendering of A-frame over Bearing

Photo of Bearing Assembly and Frame

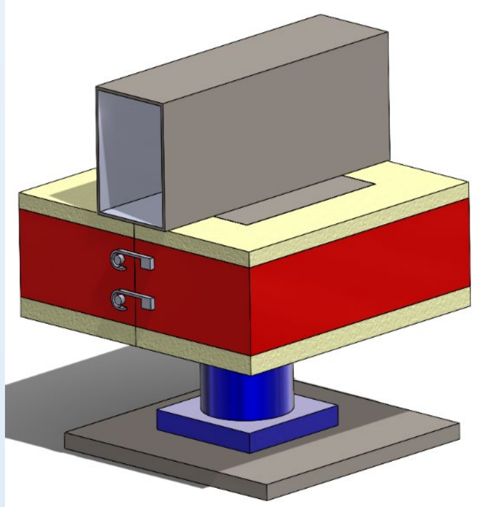


Elevation View of Bearing Connection

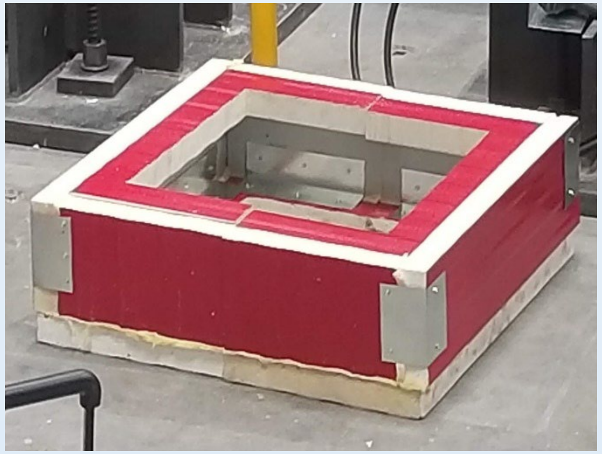
Wetting and Freezing Process

- Most tests were performed with bearing seals off so that bearing cavity could be readily accessed to fill with water through hose in the side.
- For cleanup, A-frame was lifted with bearing attached to open the bearing cavity. The surface was cleaned with a vacuum and ice melted/ surface dried with heat lamps.
- To freeze water inside the bearing, an insulation box was wrapped around the bearing within the setup and filled with dry ice. The box was left in place overnight, and dry ice refilled twice (midnight and early morning) to reach the coldest temperatures.
- Temperature inside the water cavity and on the bearing surface was measured with thermocouples.

Bearing just prior to “frozen test”; insulation box removed



Insulation Box Concept



Insulation Box Photo

Bearing Characterization and Observed Asymmetries

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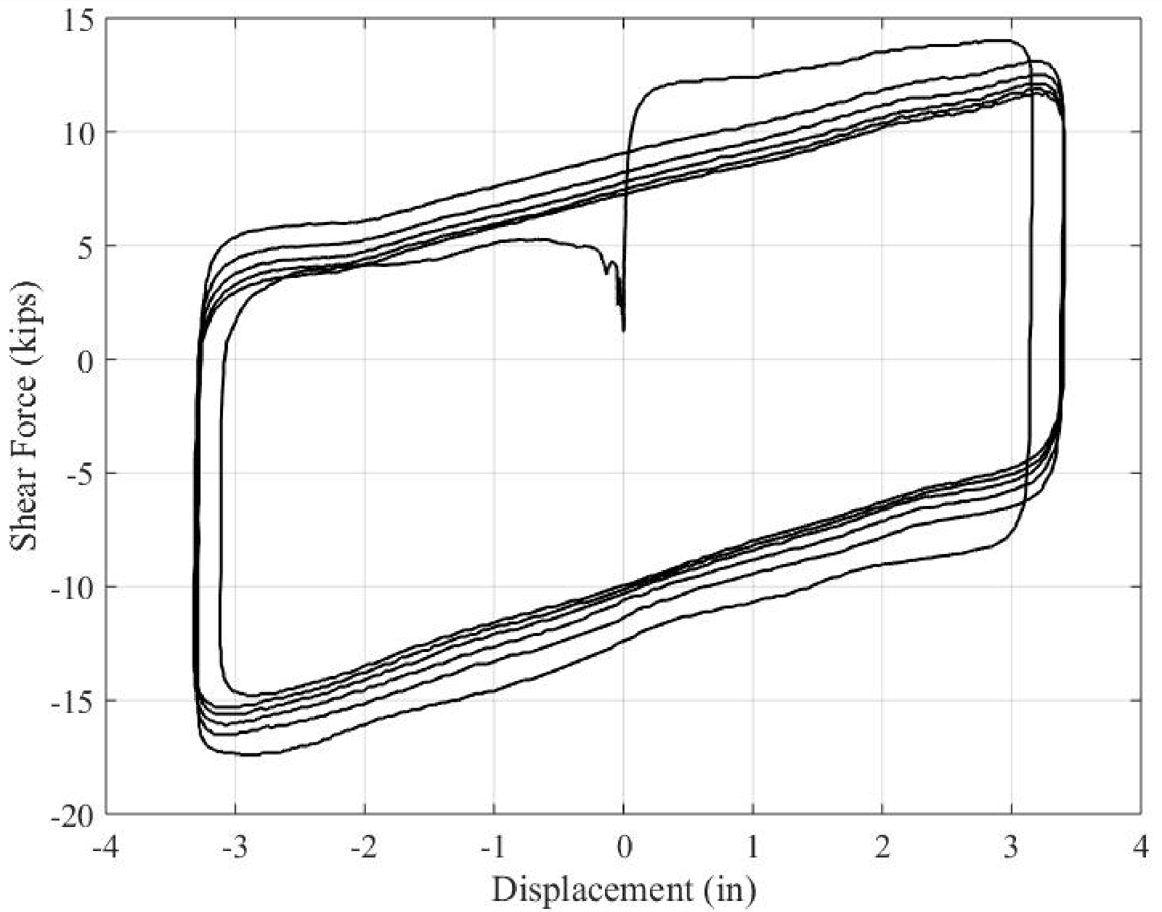
Bearing Characterization Procedure

Sample Force vs. Displacement, or Hysteresis Loop to Constant Sine Wave Motion

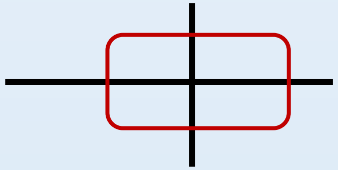


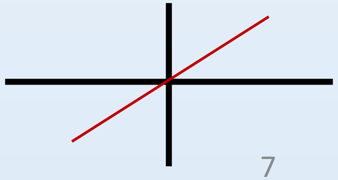
- Bearings were subjected to constant sine wave, increasing amplitude ramping sine wave, and earthquake displacement histories.
- Yield force of hysteresis loop reflects the friction coefficient and the slope represents the stiffness, or pendulum length.
- Bearing hysteresis behavior.

$$F = \mu W sgn(\dot{u}) + \frac{W}{L}u$$

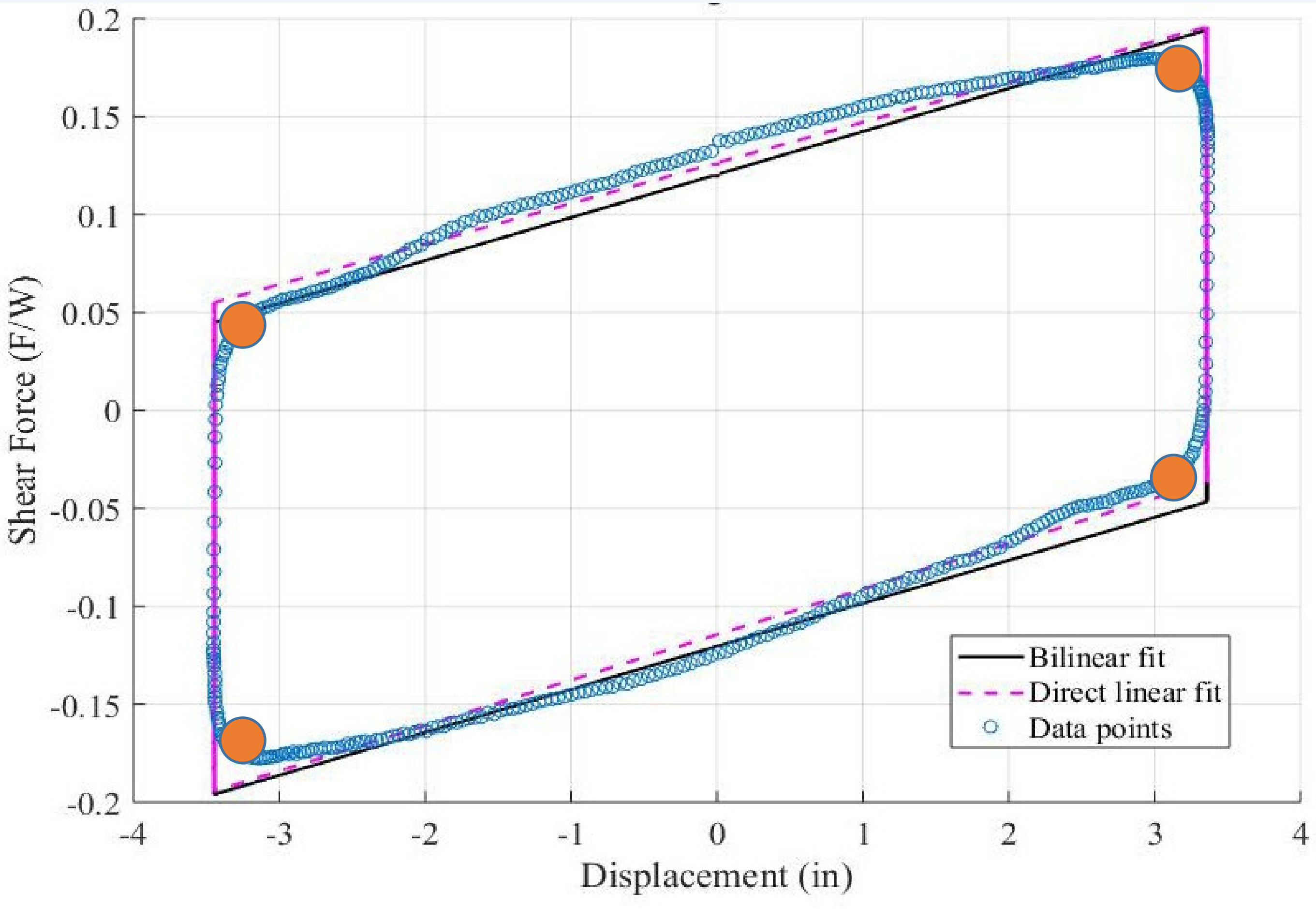
Rectangular hysteresis

Linear





Bearing Characterization Procedure

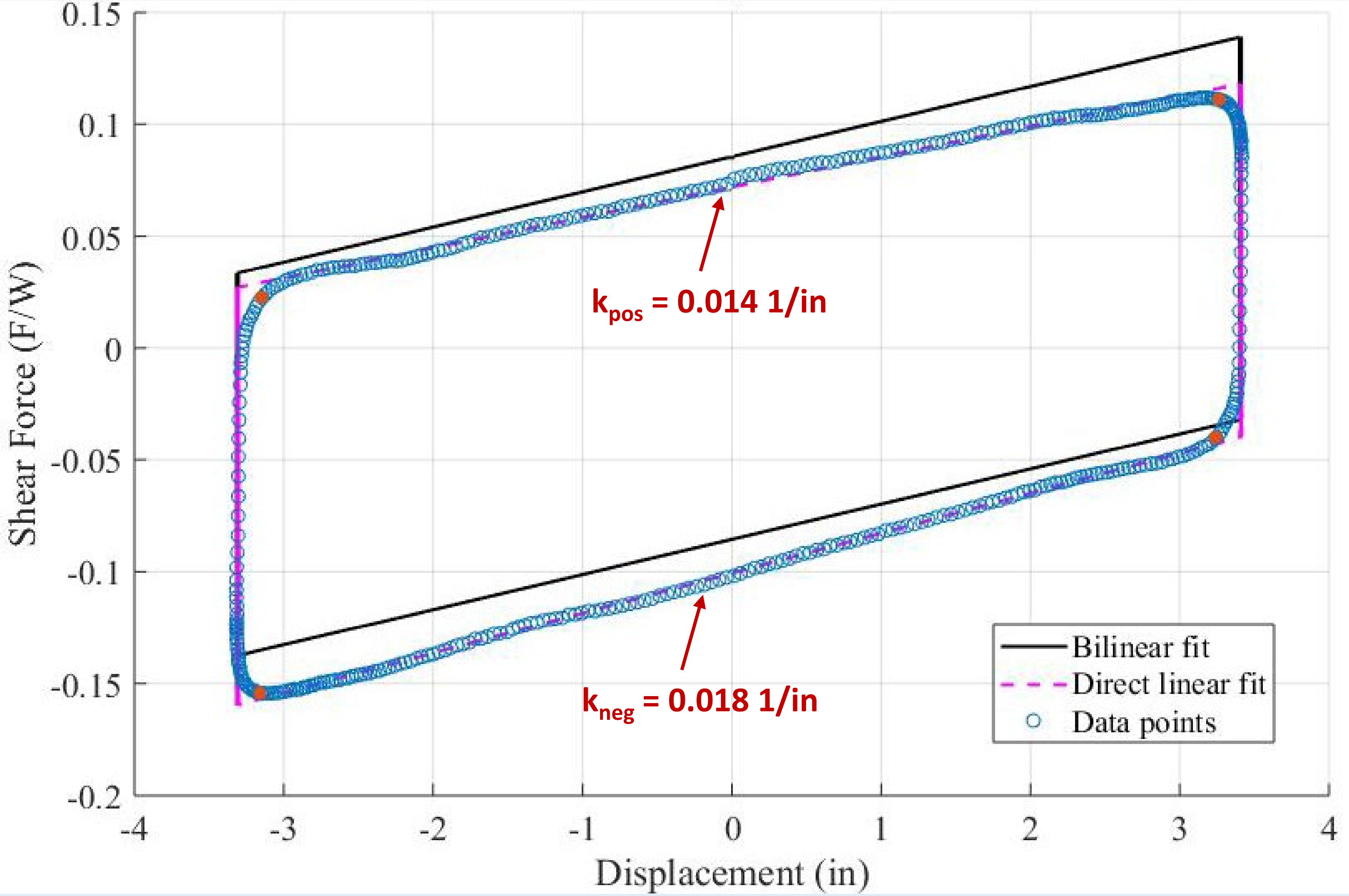


- First and last cycle were discarded; other cycles were isolated and processed independently.
- Shear force (normalized by axial force) was plotted against displacement.
- Top and bottom surface stiffness were fit independently using linear regression (Direct linear fit).
- Top and bottom surface stiffnesses were averaged.
- Energy dissipation (EDPC) was computed by numerically integrating to get the area of the hysteresis loop, and friction coefficient μ computed by:

$$\mu = \frac{EDPC}{2(d_{max} - d_{min})}$$

- Final idealized loop (Bilinear fit) was plotted

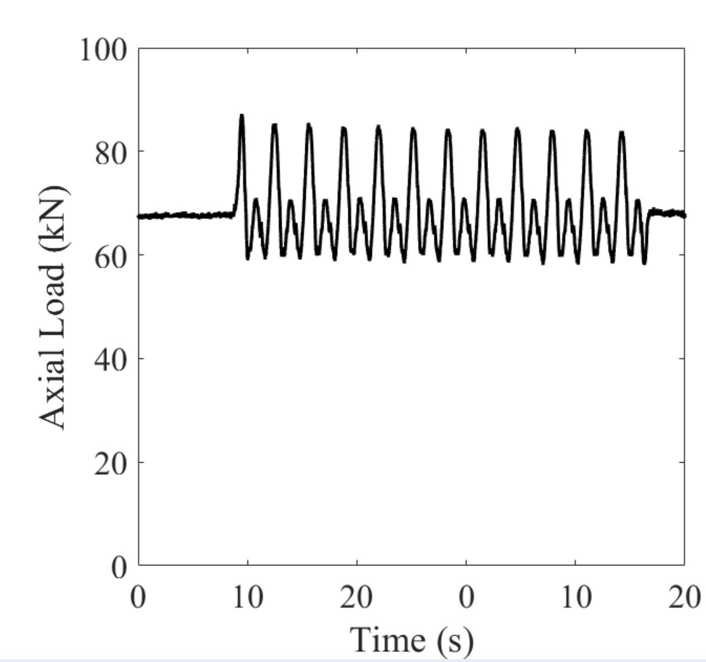
Stiffness asymmetries were observed in many hysteresis loops



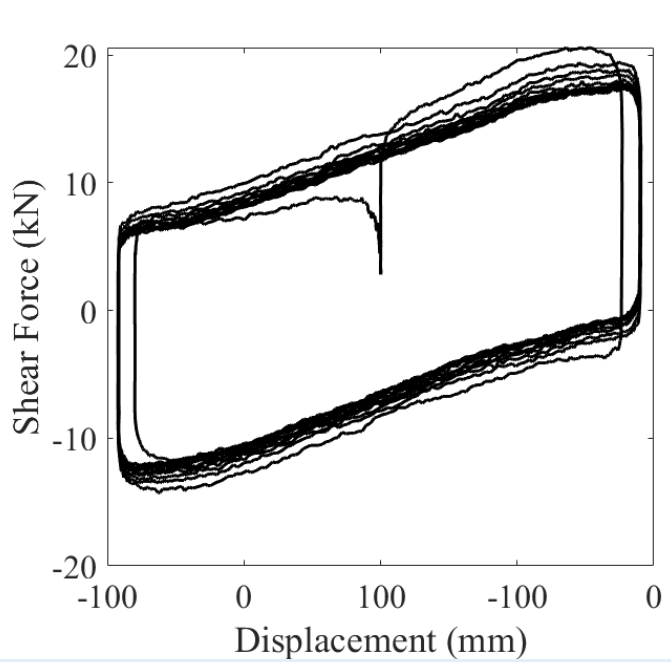
Axial Force Variation

- We were not able to hold the axial force constant. A slight variation in axial force was induced by movement of the bearing and frame rotation imposing displacement on the ram.

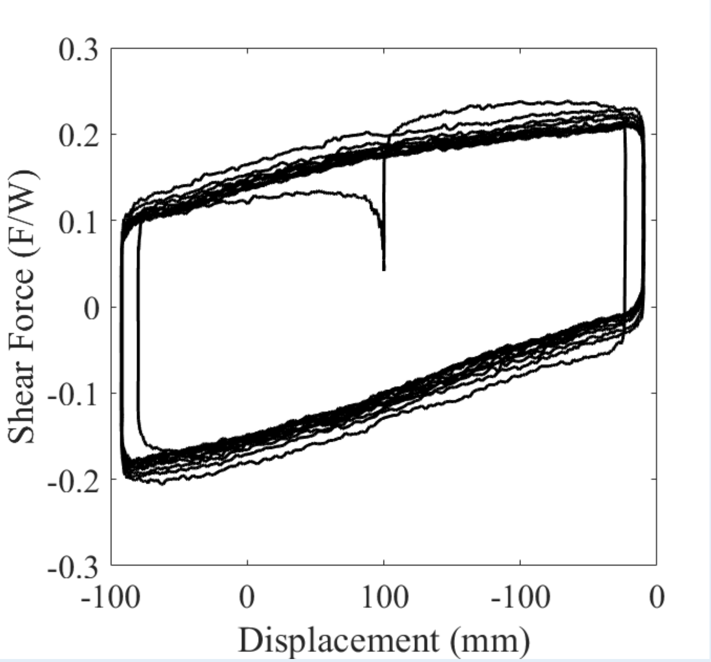
Axial Force History



Absolute Hysteresis Loop



Normalized Hysteresis Loop

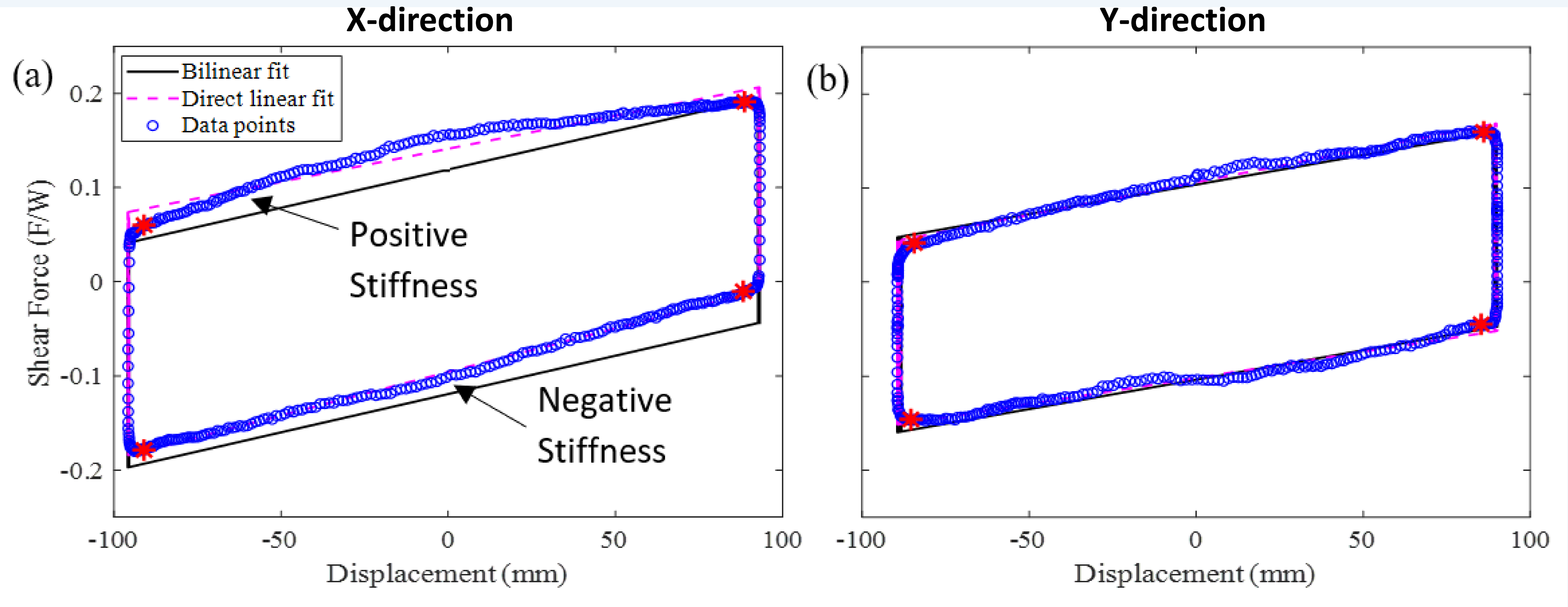


- We were not able to hold the axial force constant. A slight variation in axial force was induced by movement of the bearing and frame rotation imposing displacement on the ram.

- Normalized hysteresis loop is more asymmetric. Friction coefficient can increase with decreasing axial force, but appears that some other phenomenon is at play.

Stiffness Asymmetry

- Stiffness asymmetries were worse in the X-direction (parallel to frame) than the Y-direction



Example from Robertson

Modifications to Test Set Up

- We suspected the asymmetry was due to rotation and overall play in the system.
  - Lots of “play” in the system, e.g. at column to frame connection
  - We begin to notice slip in the bolts at the angled connection
- Changes were made for a second phase of tests.

Phase 1

Phase 2

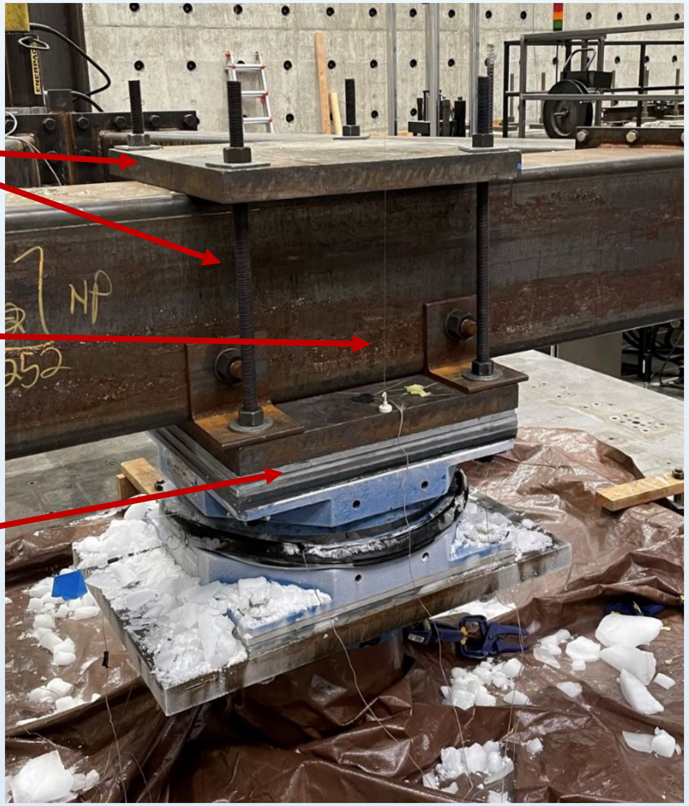


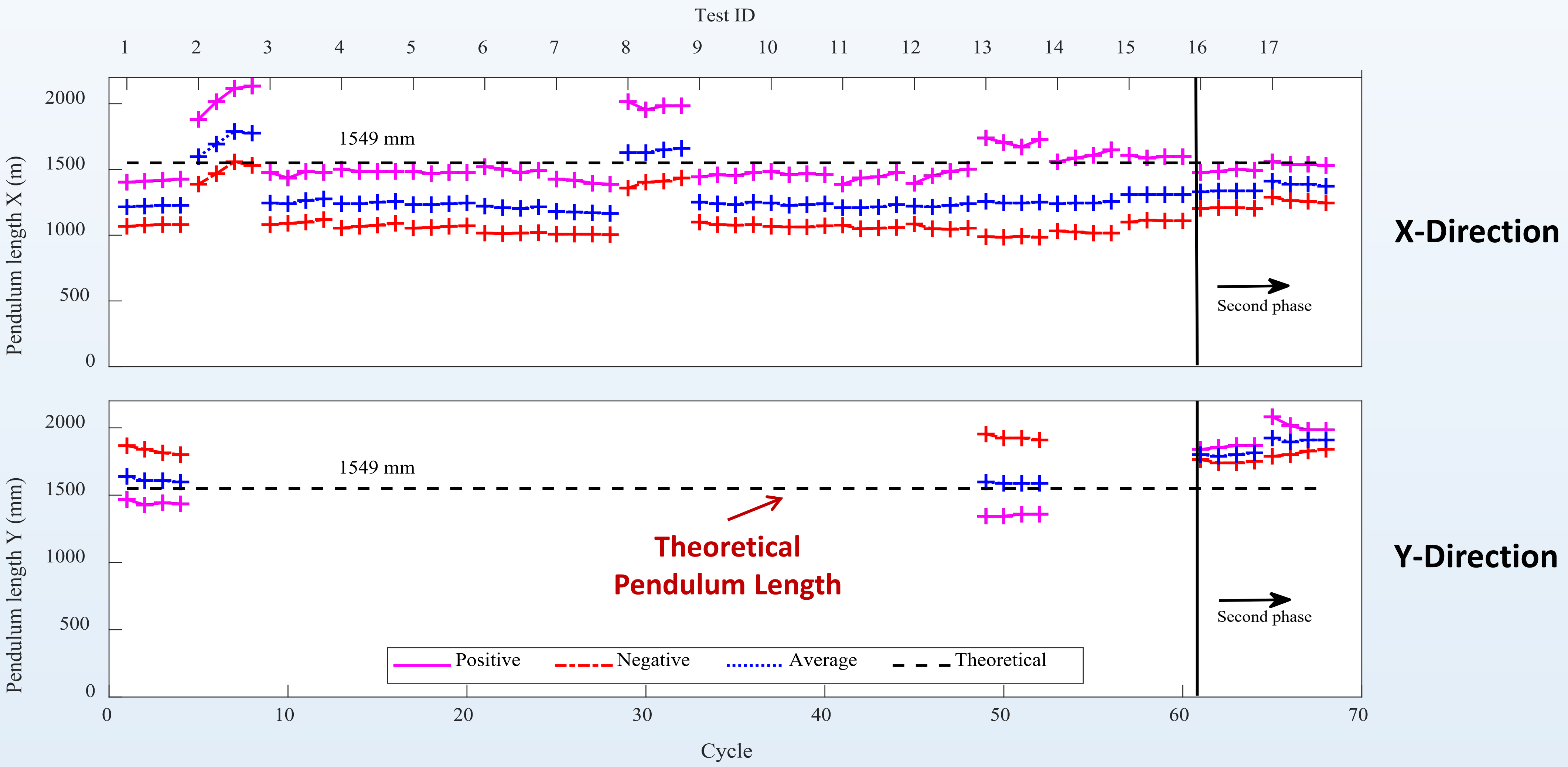
Plate and rods arrangement to prevent slip

Vertical string pots (both ends) to measure rotation

Extra insulation plates to level the frame

More y-direction tests

Observed Pendulum Length - Phase 1 vs. 2



- No significant improvement in Phase 2 compared to Phase 1.

Susitna DPB Sliding Regimes

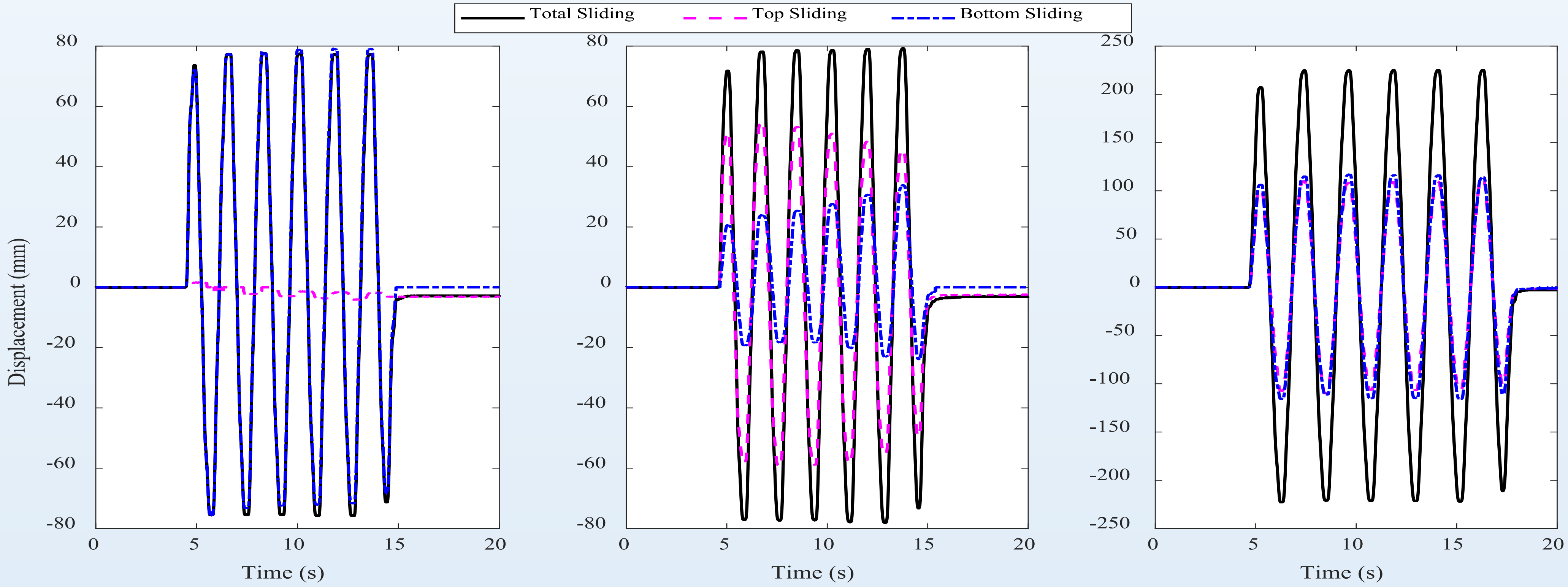
Susitna DPB Sliding Regimes

- Displacement of each sliding surface was observed in many tests. Multiple sliding regimes were observed.

Sliding limited to one surface (wet test at 244 kN axial load)

Sliding dominant on one surface but still sliding on the other surface (dry test at 244 kN axial load)

Even sliding on both surfaces (dry test at 244 kN axial load)



Susitna DPB "Single Surface Sliding"

- A number of conditions might limit sliding on one surface (or suppress it altogether).
  - Old Susitna 1 "as-received" with dirt caked onto the bottom surface.
  - In frozen condition with ice frozen into the lower bearing cavity.
  - In wet tests, possibly due to an increased static friction or decreased dynamic friction on the wet surface.
  - In dry tests, possibly due to corrosion of one surface relative to the other.
  - *In any test, possibly due to an initial rotation.*
- If a Double Pendulum Bearing slides only on one surface, the pendulum length is approximately half or the post-yield stiffness doubles. It responds like a Single Pendulum Bearing.

Old Susitna 1 as received

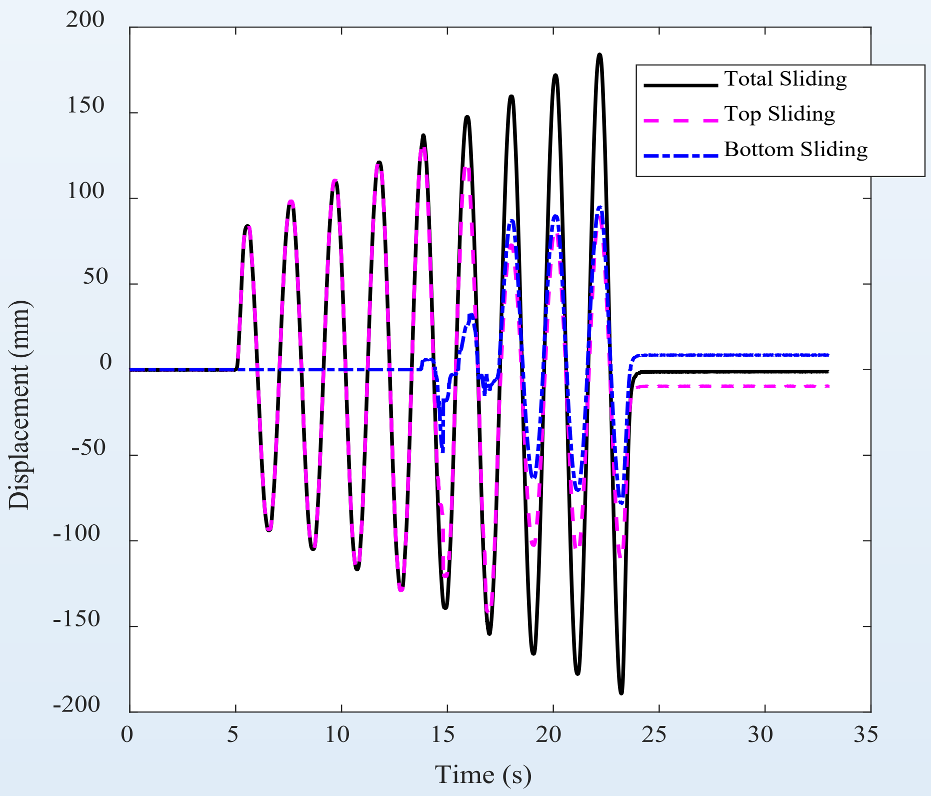
Old Susitna 1 with ice frozen in cavity

Old Susitna 2 top rusted/pitted surface

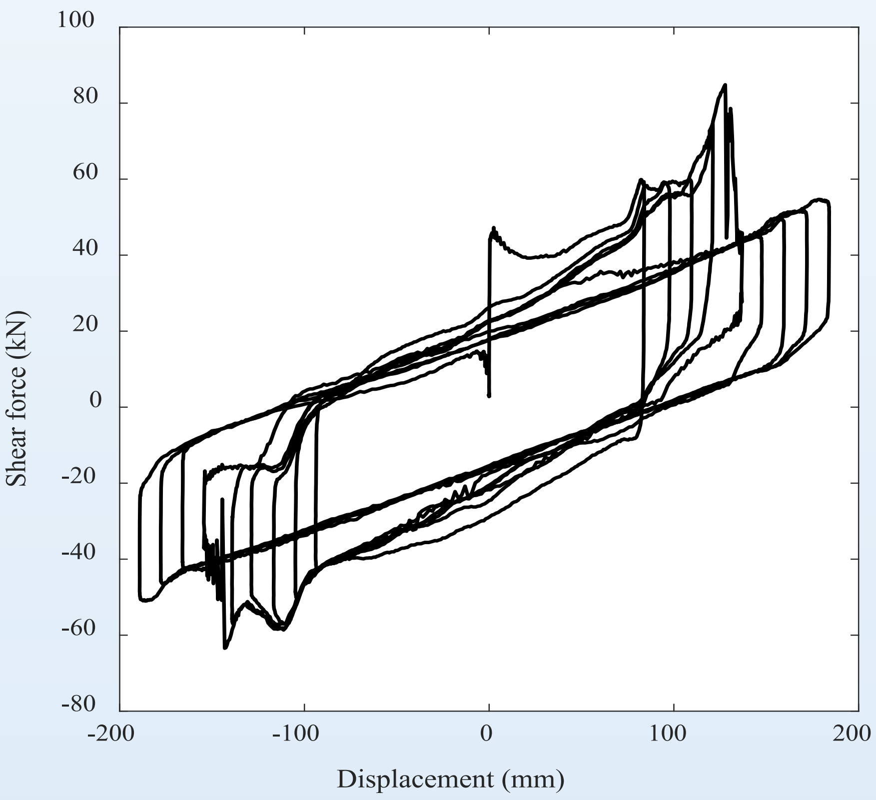
Example from a Frozen Bearing Test

- Initially, the ice in the bottom cavity constrained movement to the top surface. The effective stiffness of the hysteresis loop is approximately double.
- Both surfaces start sliding once the top surface displacement limit is reached. A clear reduction in effective stiffness of the hysteresis loop is observed.

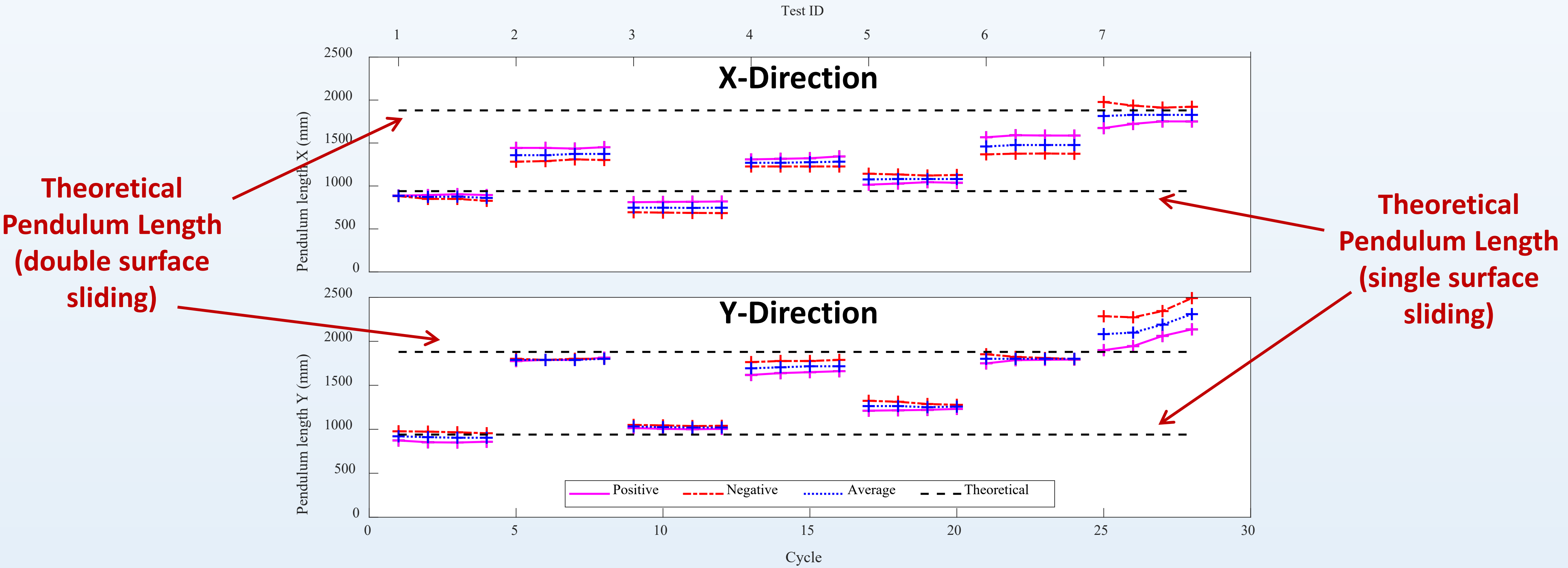
Surface Displacement Histories



Total Bearing Hysteresis Loop



Observed Pendulum Length in DPB Dry Tests on Old Susitna 2



- Observed pendulum length aligns roughly with observations of single surface vs double surface sliding.
- Theoretical double surface sliding pendulum length should apply even in uneven sliding as long as both surfaces are engaged.

Rotation Theory

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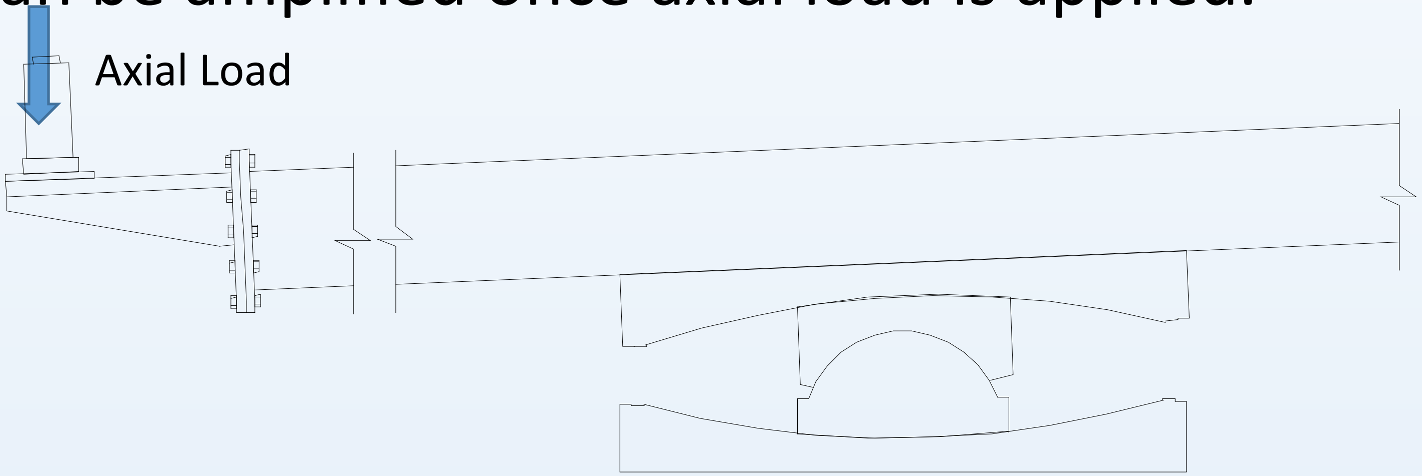
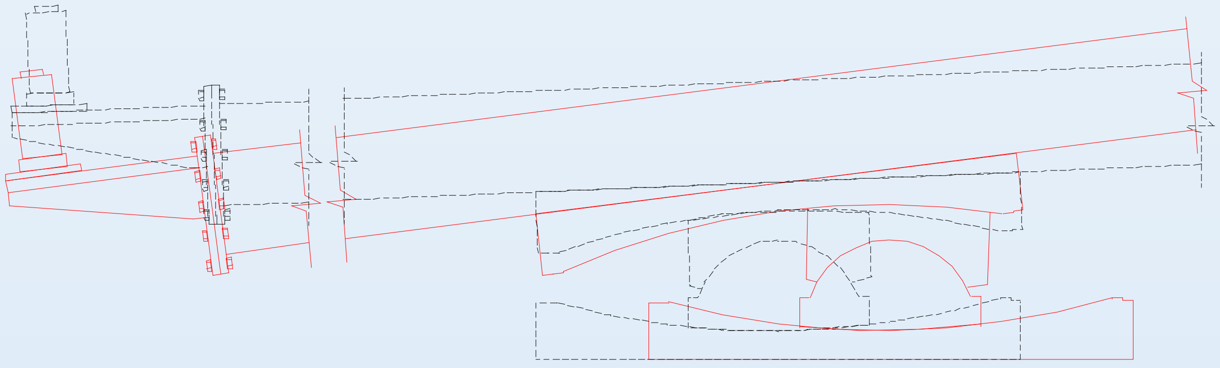
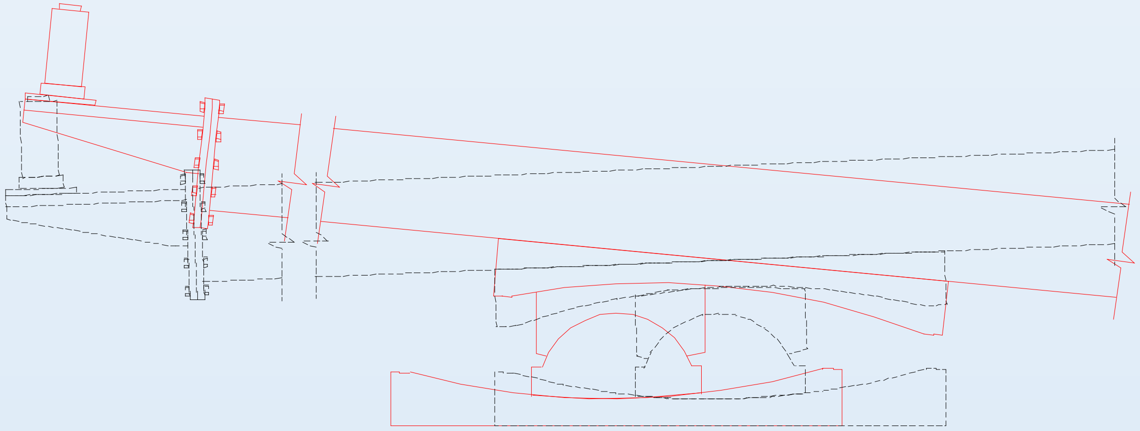
Rotation of the Bearing Top Plate

- Bearing is slightly rotated before the test starts due to the lever arm of the frame, and can be amplified once axial load is applied.



- Movement of the bearing can cause dynamic rotation during the test.





Effects of Rotation on Single Pendulum Bearings

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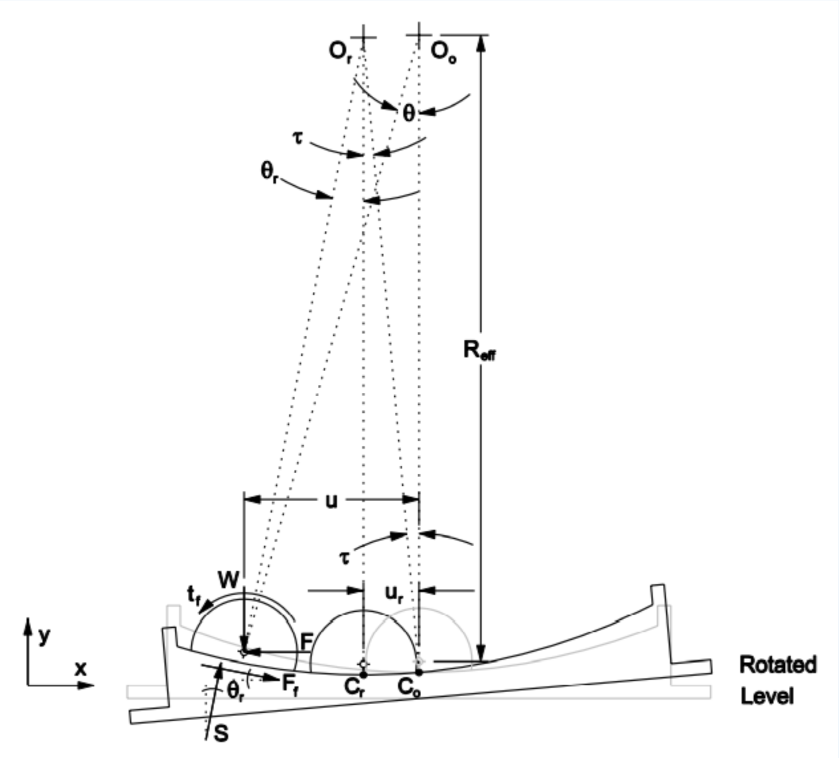
Effect of rotation was considered by Mosqueda et al. (2004) and Fenz and Constantinou (2008)

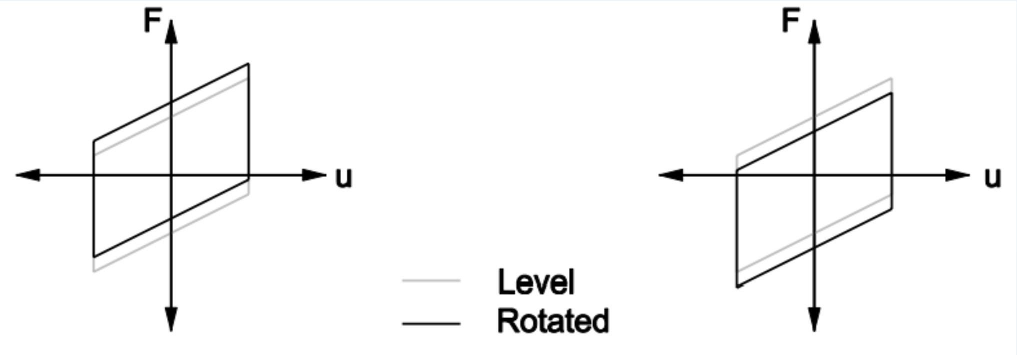


Fenz and Constantinou (2008)

- Considering the geometry of the bearing, force-deformation is modified as follows.

$$F = (\mu \pm \tau)W + \frac{W}{L}u$$

- Rotation causes a slight increase or decrease in the effective friction coefficient, depending on the direction of movement.
- Constant rotation would induce an upward or downward shift in the hysteresis loop.



Effects of Rotation on Single Pendulum Bearings

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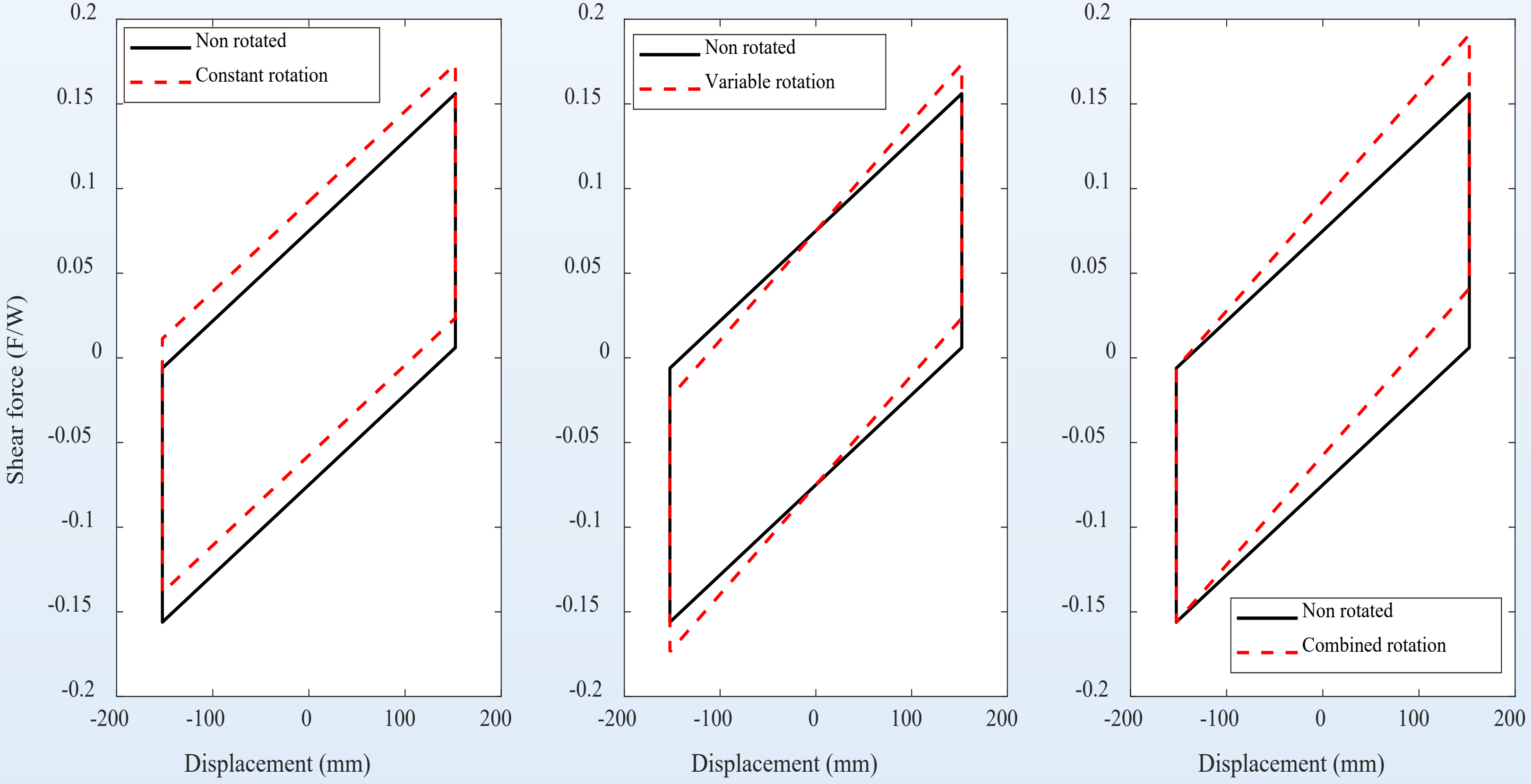
$$F = (\mu \pm \tau)W + \frac{W}{L}u$$

- Linearly varying rotation could cause a dynamic variation to the friction coefficient that could manifest as a change in effective stiffness (pendulum length).

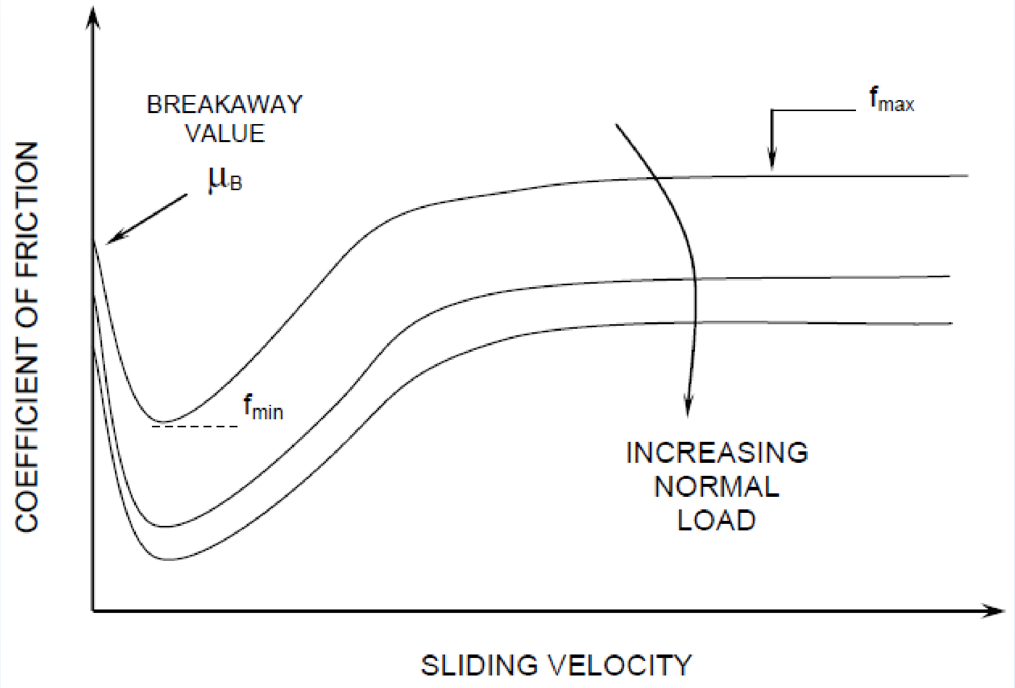


- The combination of constant rotation and linearly varying rotation could manifest as hysteresis loops that are:
  - Off center
  - Apparent effective pendulum length that varies from theoretical
  - Asymmetric loops with different positive and negative stiffnesses.

Effects of Rotation on Single Pendulum Bearings

$$F = (\mu \pm \tau)W + \frac{W}{L}u$$

- Linearly varying rotation could cause a dynamic variation to the friction coefficient that could manifest as a change in effective stiffness (pendulum length).



Source: Constantinou et. al (2008)

- However, friction coefficient is also affected by variations in velocity and axial force; in summary complex phenomena make it difficult to tease out the effects.

Series Spring Model for Double Pendulum Bearings

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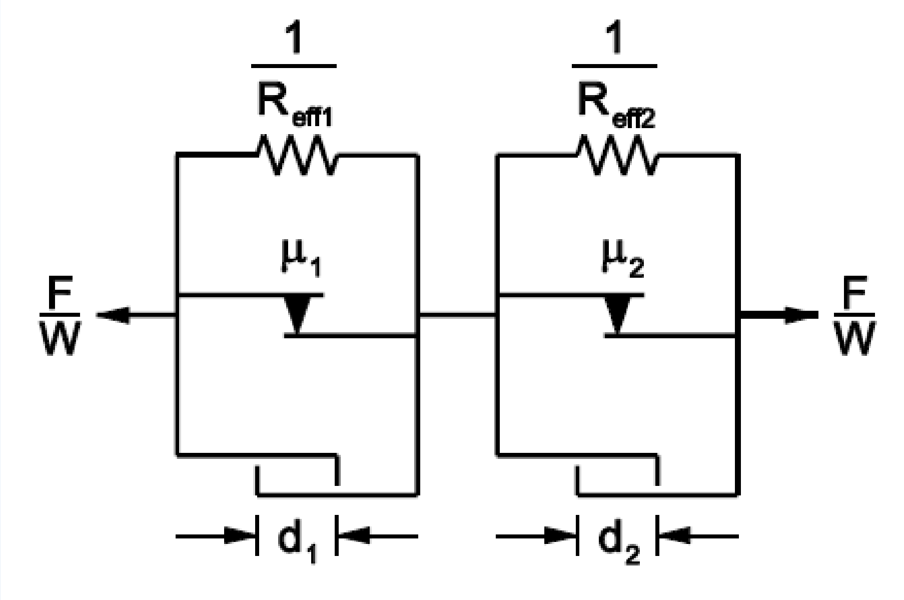
Fenz and Constantinou (2008) presented rotation in the context of the series spring model for double pendulum bearings.

- For each sliding assembly we have the following equation: $$F = \mu_i W + \frac{W}{L_i} u_i$$
- Each sliding assembly is a parallel arrangement of a rigid-plastic friction device, a linear spring for the curvature, and a gap element accounting for the finite displacement limit.
- The force passing through each sliding assembly must be the same.



Source: Fenz and Constantinou (2008)

Proof of Sliding vs Pendulum Length

- Equation for a single surface

$$F_i = \mu_i W + \frac{W}{L_i} u_i$$

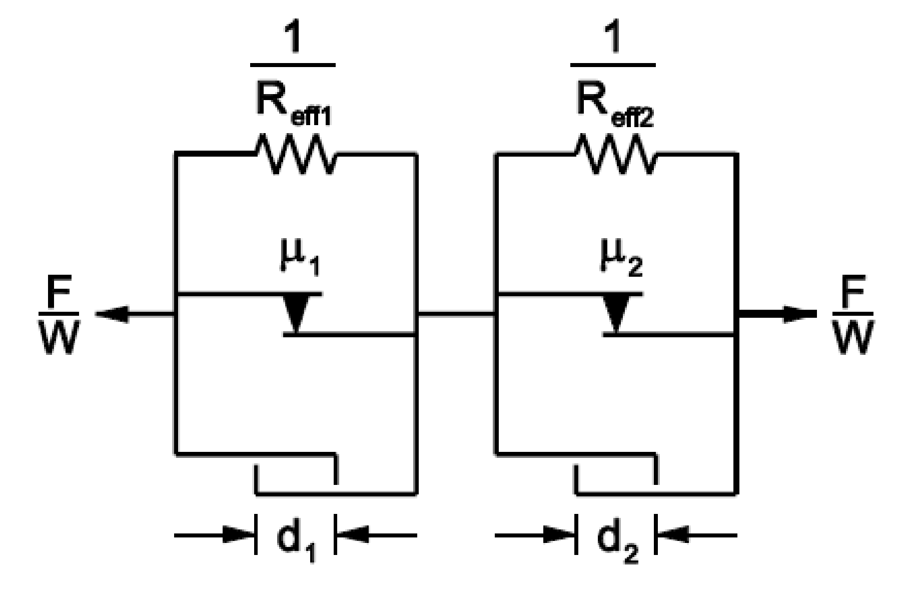
- Equation for general sliding is derived by setting $F_1 = F_2$, and $u = u_1 + u_2$

$$F = \frac{[\mu_1 L_1 + \mu_2 L_2]W}{L_1 + L_2} + \frac{W}{L_1 + L_2} u$$

- When bearings have the same pendulum lengths on both sliding surfaces, $L_{eff} = L_1 + L_2$

$$F = \frac{[\mu_1 + \mu_2]W}{2} + \frac{W}{L_{eff}} u$$

- Pendulum length is twice that of a single surface.



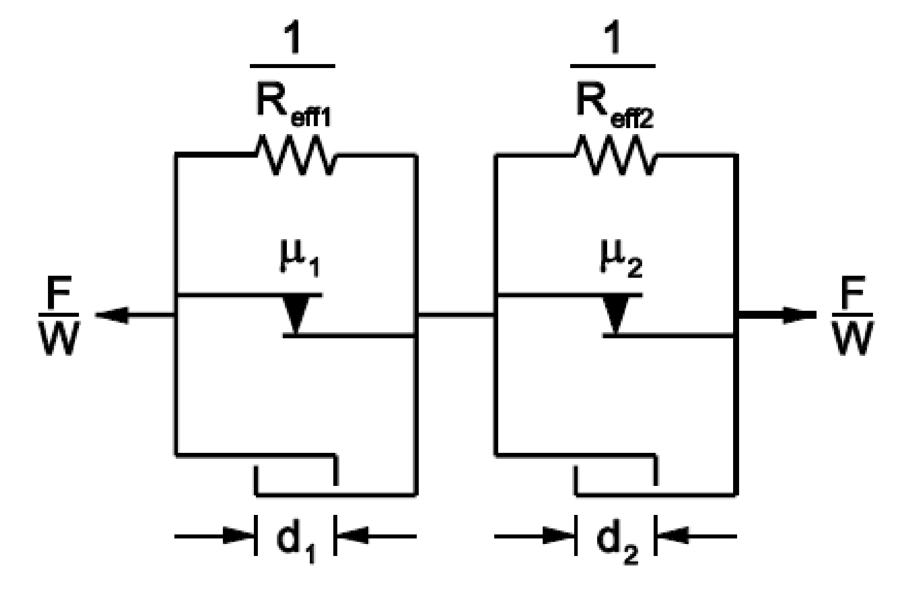
- But if only one surface slides, pendulum length is $L_{eff}/2$

Effect of Rotation on Double Pendulum Bearing

What if there is rotation of the top surface or the bottom surface?

- Each sliding assembly will have the following equation: $$F = (\mu_i \pm \tau_i)W + \frac{W}{L_i}u_i$$

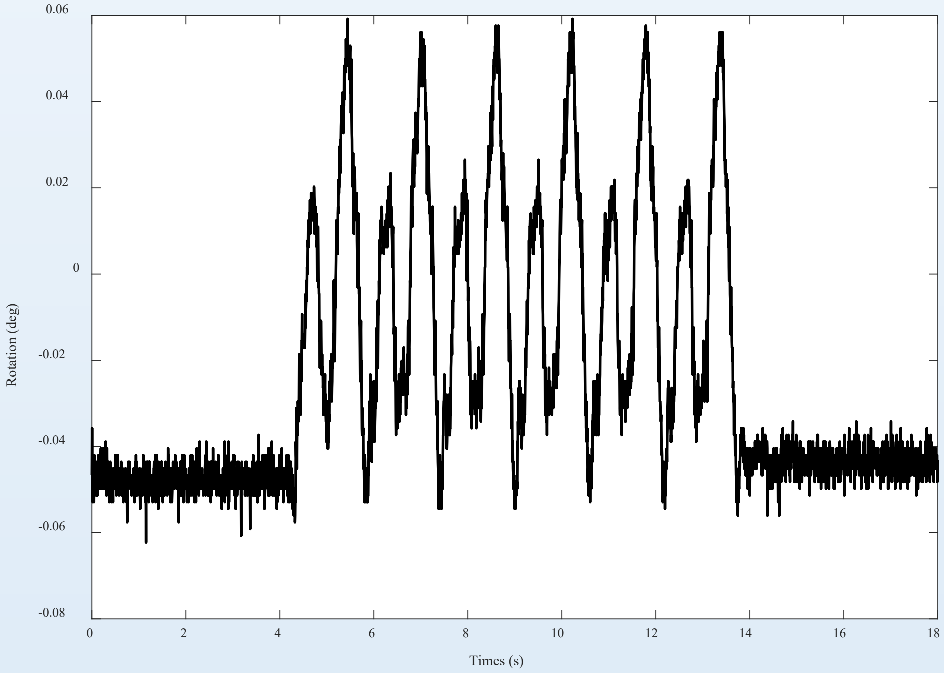
- To keep the force in each assembly equal, a rotation on one or both plates could cause unequal sliding as displacements u_i adjust to compensate.
- If the rotations and/or friction is really unbalanced, sliding might be limited to one surface.

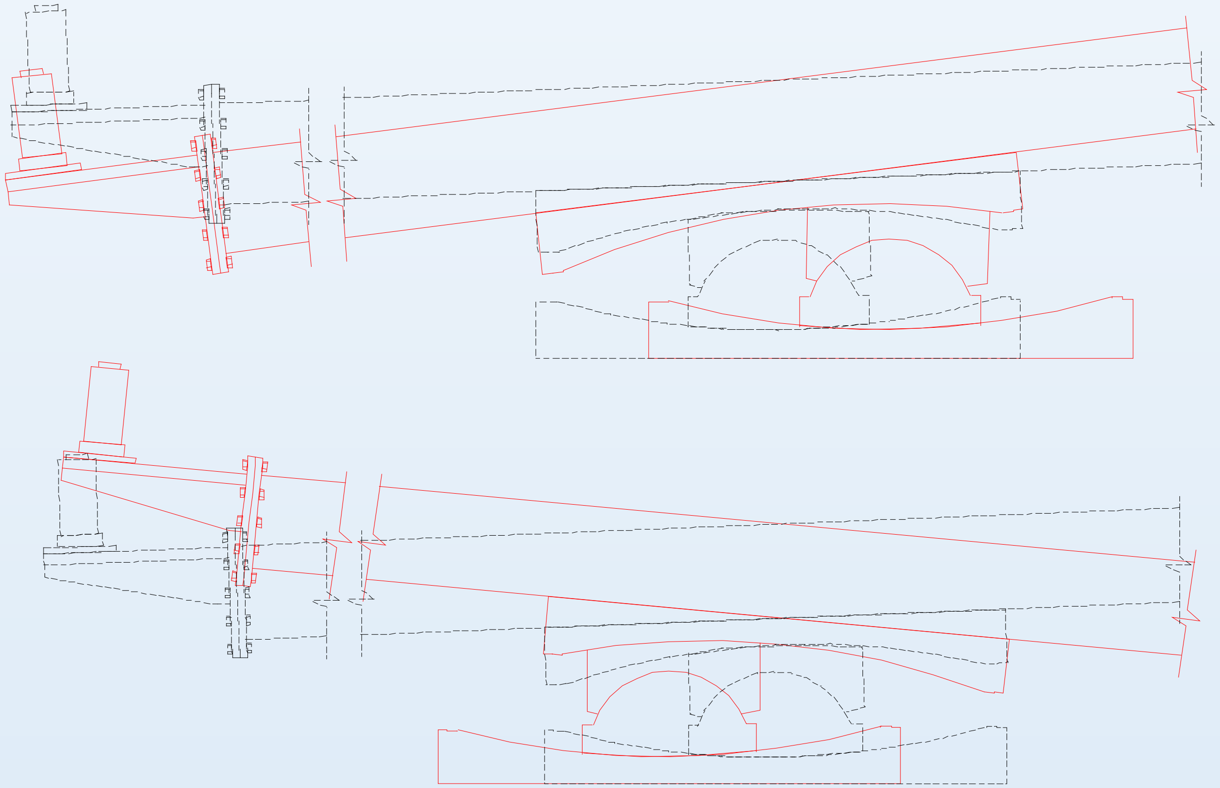


Source: Fenz and Constantinou (2008)

Rotation of the Bearing Top Plate

- Rotations were measured in Phase 2 tests, by vertical string pots on each side of the frame.
- Initial rotation could not be measured (string pots are not perfectly level and even).
- Observed rotations were generally small, and did not match the expected pattern closely (possibly due to interaction with bearing vertical movement).





Alternate Characterization

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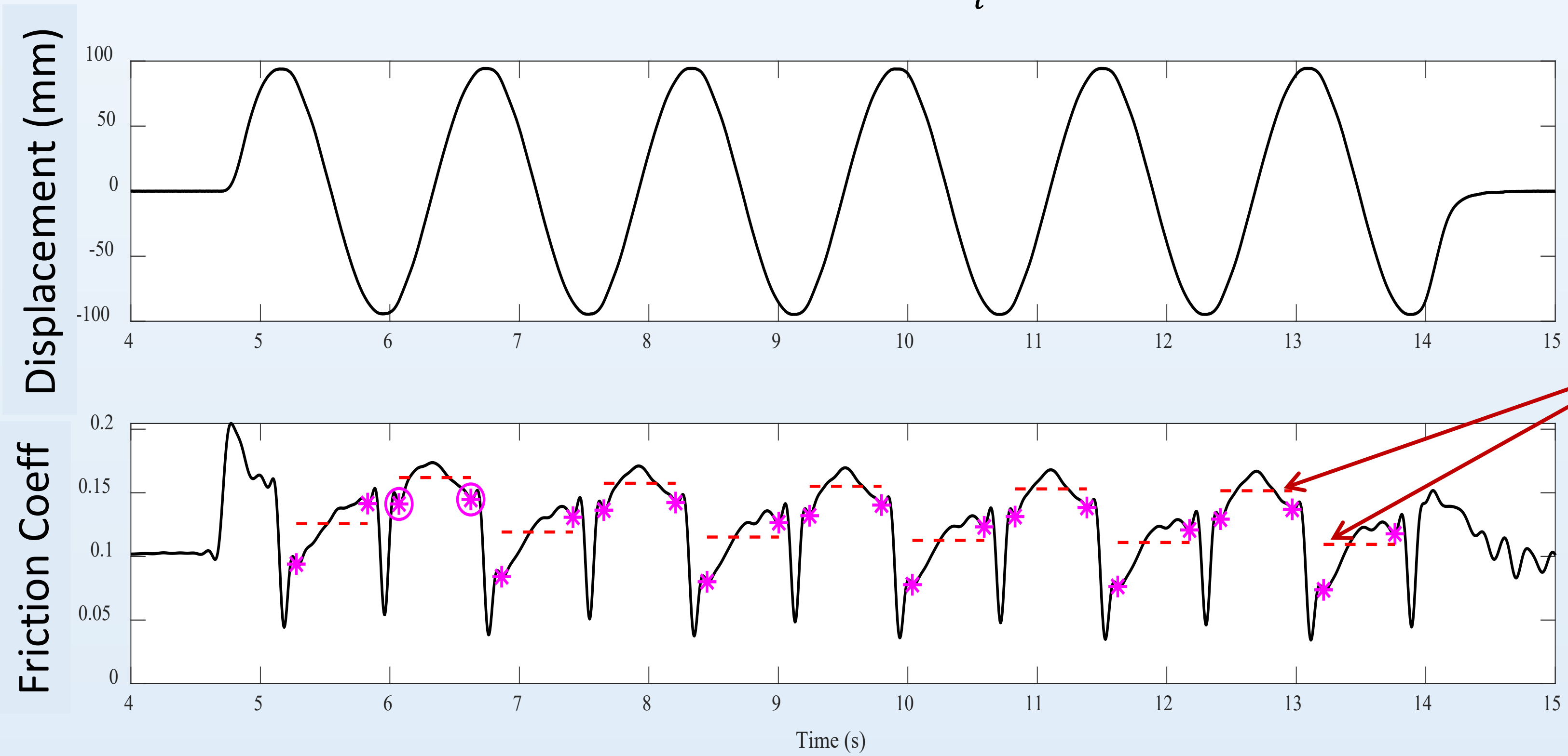
Investigation of Initial Rotation

- Alternate Characterization: Assume pendulum length is a known parameter, and the instantaneous friction is computed from the experimental data, which may include an initial rotation component

$$(\mu \pm \tau)_i = \frac{F_i}{W_i} - \frac{1}{L} u_i$$



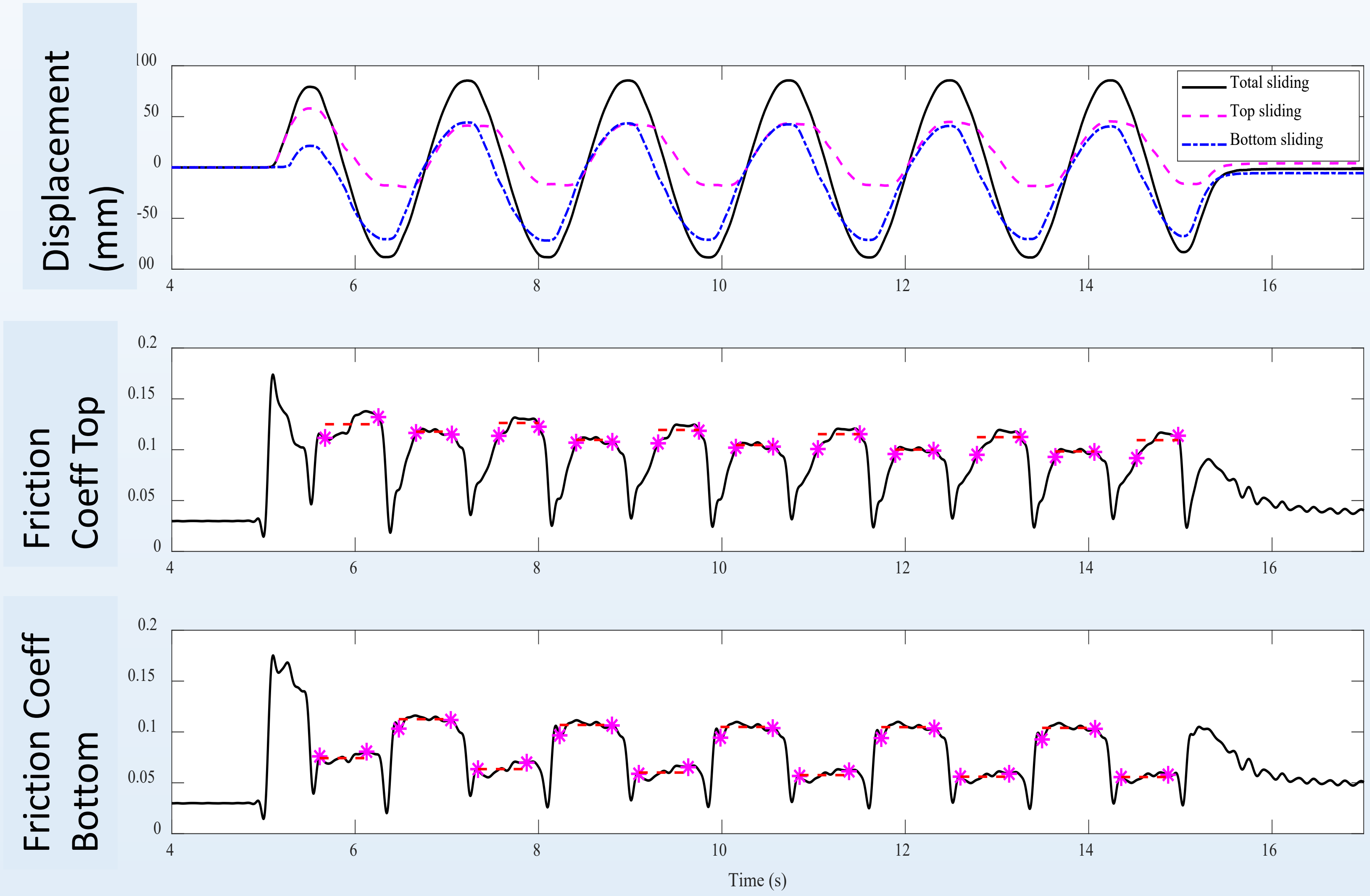
Sample for Robertson bearing

Red dashed lines = average friction through the half cycle

Constant τ interpreted as difference between red lines divided by 2

Friction coefficient did not vary linearly through the half cycle as we expected

Sample for a double pendulum

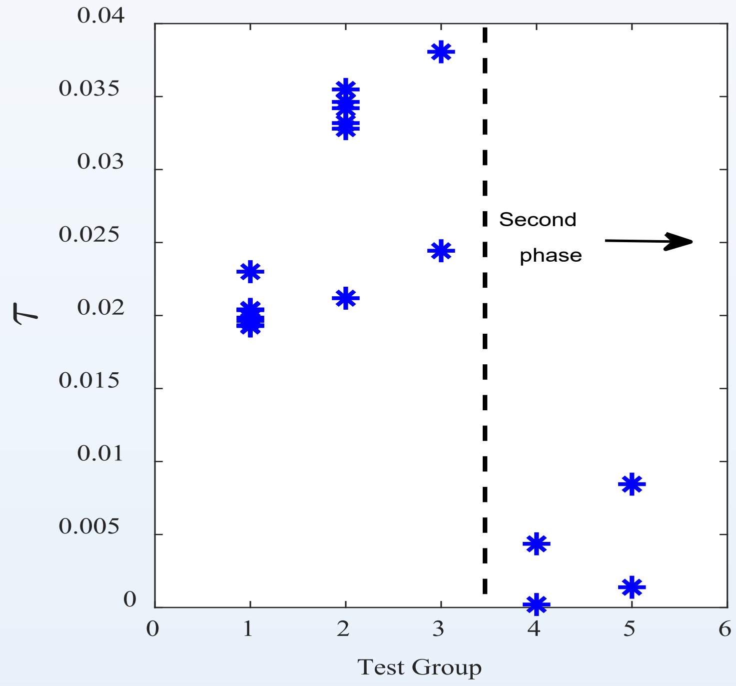


A case of uneven sliding, with much more sliding on bottom surface (but directionally dependent)

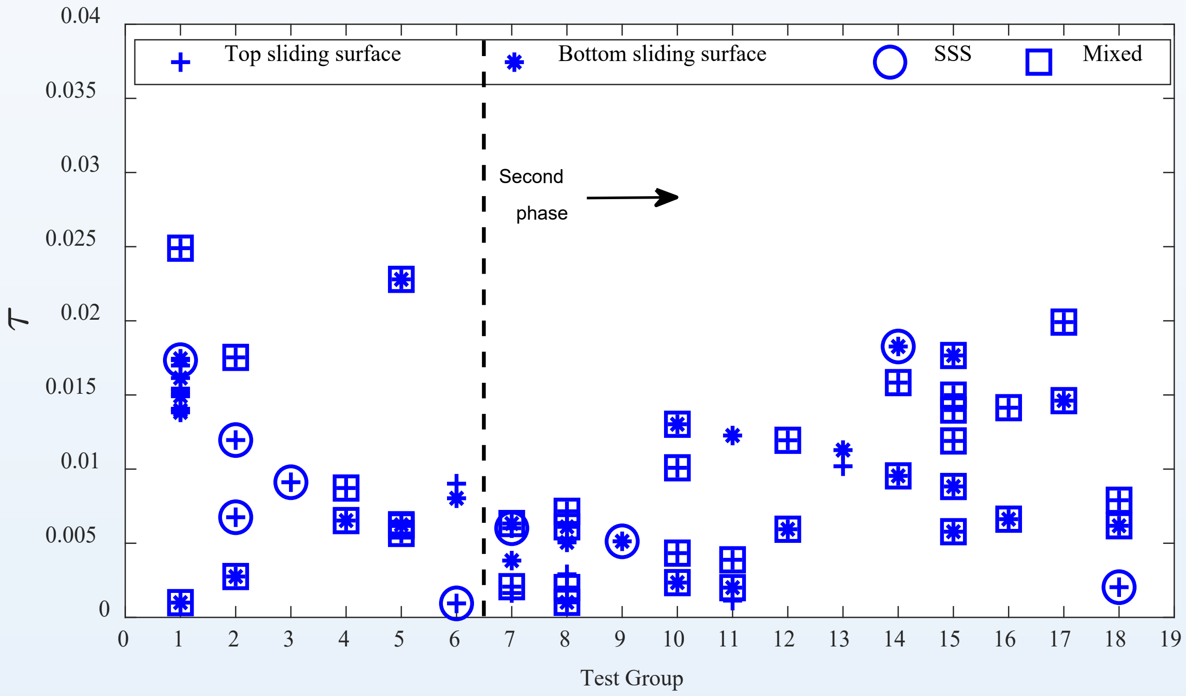
The effective τ was much larger for bottom surface, but we might expect the opposite.

Sample for Susitna bearing

Systematic evaluation of initial rotation τ



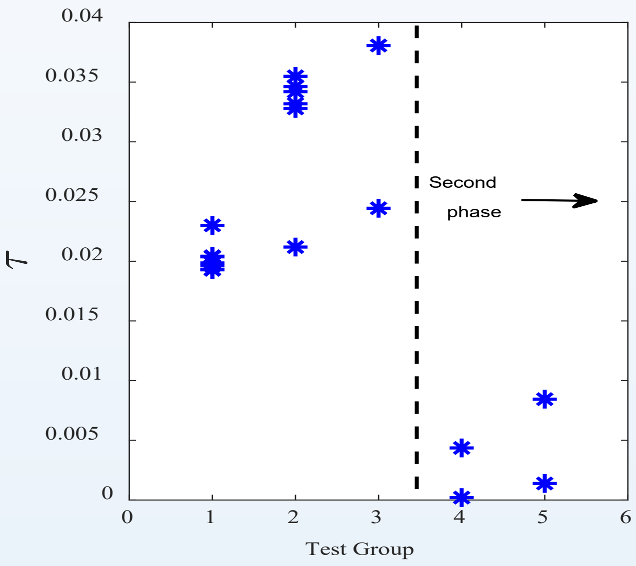
Single pendulum bearings



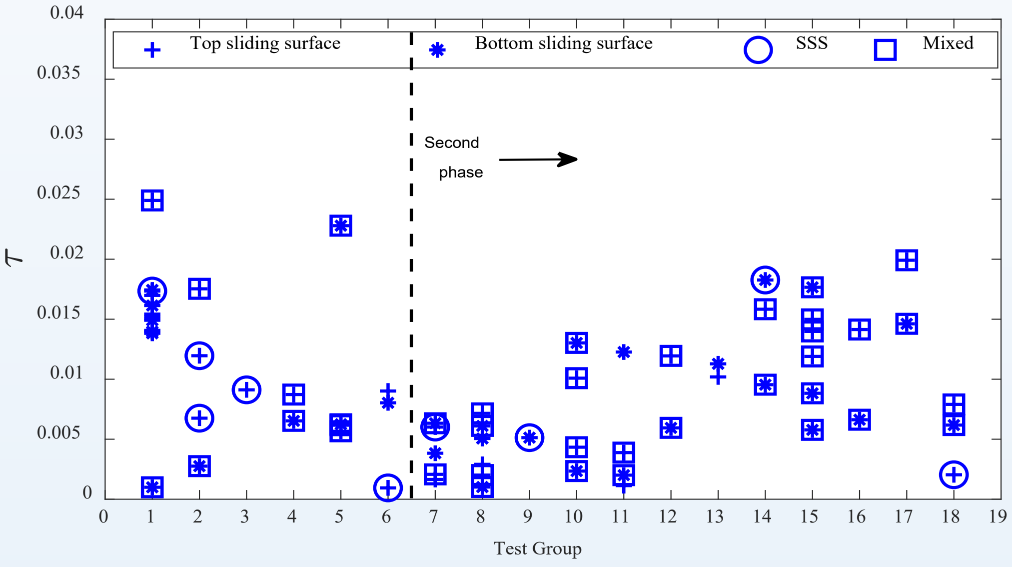
Double pendulum bearings

- Tests grouped by categories where initial rotation should be the same (i.e. same bearing, day and axial load)
- Results shown for each surface in double pendulum bearings, and episodes of mixed or single surface sliding are indicated.

Is the data consistent with initial rotation?



Single pendulum bearings



Double pendulum bearings

Expected

- Rotations would be similar for the same test group – Inconsistent
- Rotations would reduce in Phase 2 – Yes for single pendulum
- Rotations would be larger in top surface of DPB and close to zero in bottom surface – Inconsistent
- Rotations would be the same and close to zero in double surface sliding, and lower than for SSS or mixed - Inconsistent

Not conclusive data that apparent shift in hysteresis loop is due to initial rotation

Investigation of Dynamically Varying Rotation

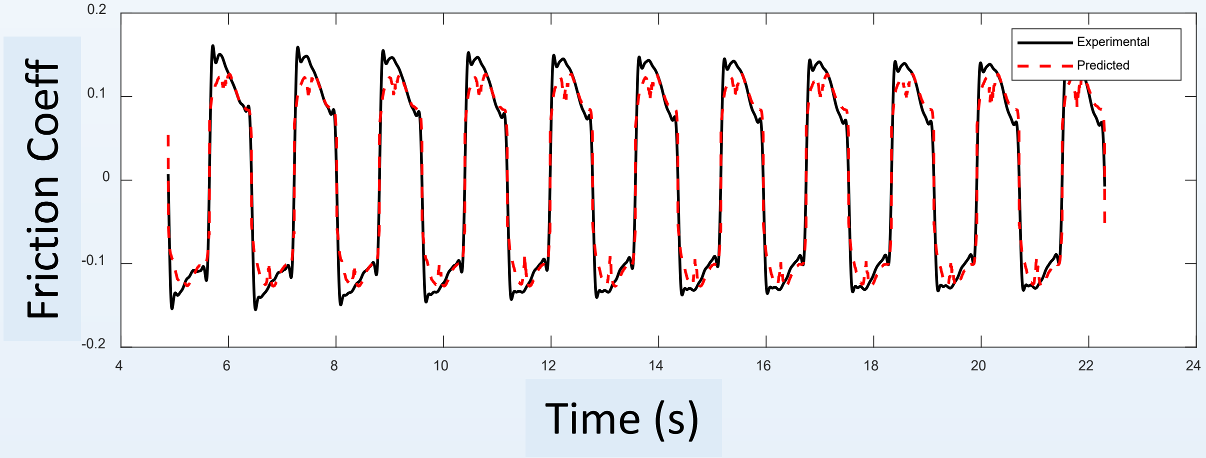
- Next Steps:
  - Fit a model that accounts for observed variation due to axial load and velocity.
  - Plot the friction μ_{pred} (predicted from the model, varies with axial force and velocity).

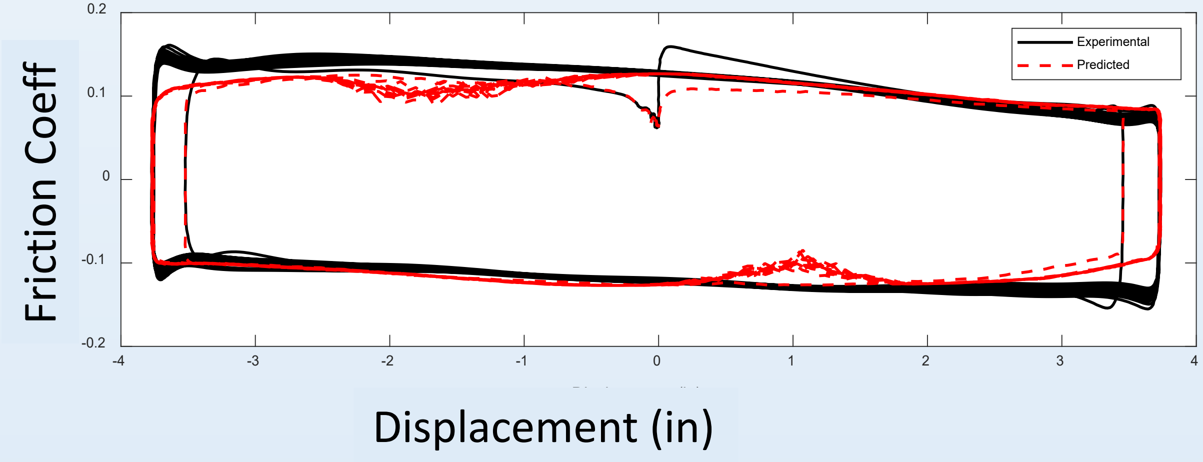
$$\mu_{pred} = \mu_{max}(W) - (\mu_{max}(W) - \mu_{min}(W))e^{-a\dot{u}}$$

  - Plot the observed μ_i directly from experimental data, where the initial rotation is subtracted.

$$\mu_i = \frac{F_i}{W_i} - \frac{1}{L}u_i \pm \tau_{init}$$

  - Observed may include rotation component; compare the difference of $\mu_{pred} - \mu_i$ to the measured rotation data.





Observed Friction Coefficient

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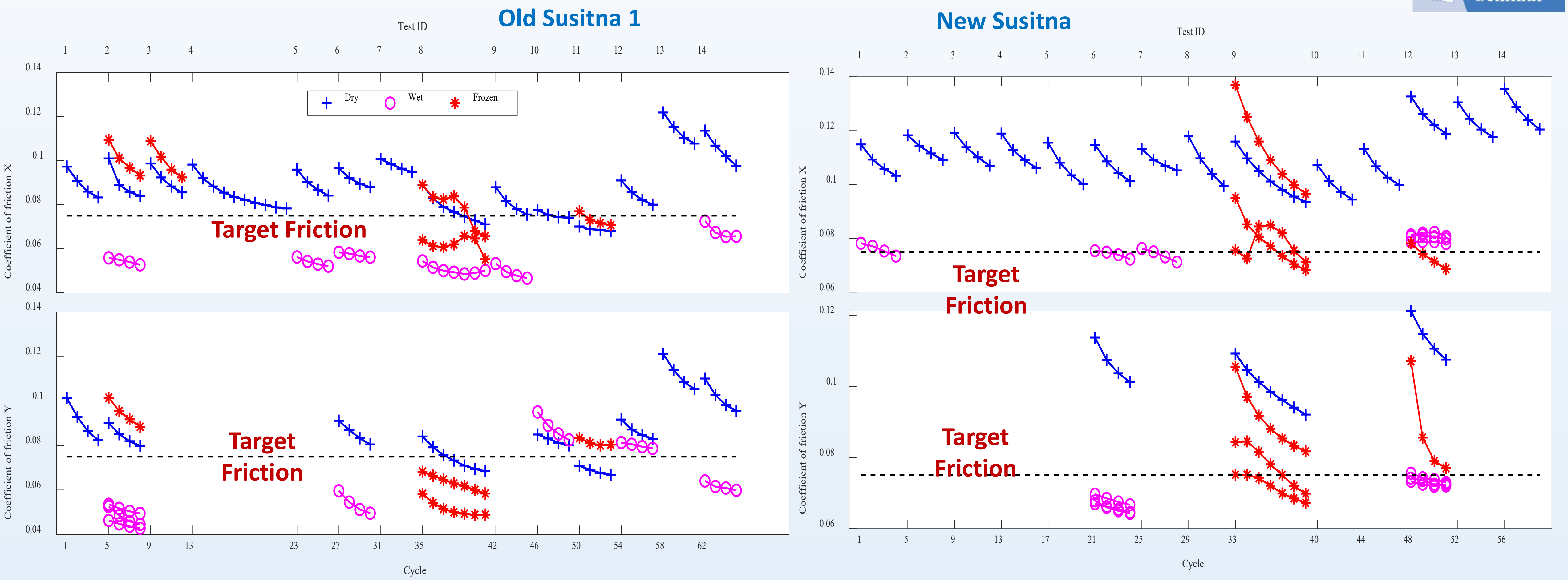
Friction Coefficient: Old Susitna 1 and New Susitna



- Dry friction is consistently larger than target friction. Target friction should be interpreted as a minimum.
- Friction coefficient decreases over the cycles in a given test. No real control for continual heating in repeated tests over the course of a day.
- Observed friction coefficient is larger in New Susitna than Old Susitna
- Wet friction is consistently lower than dry friction. No obvious trend for frozen friction.

Friction Coefficient: Robertson and Old Susitna 2

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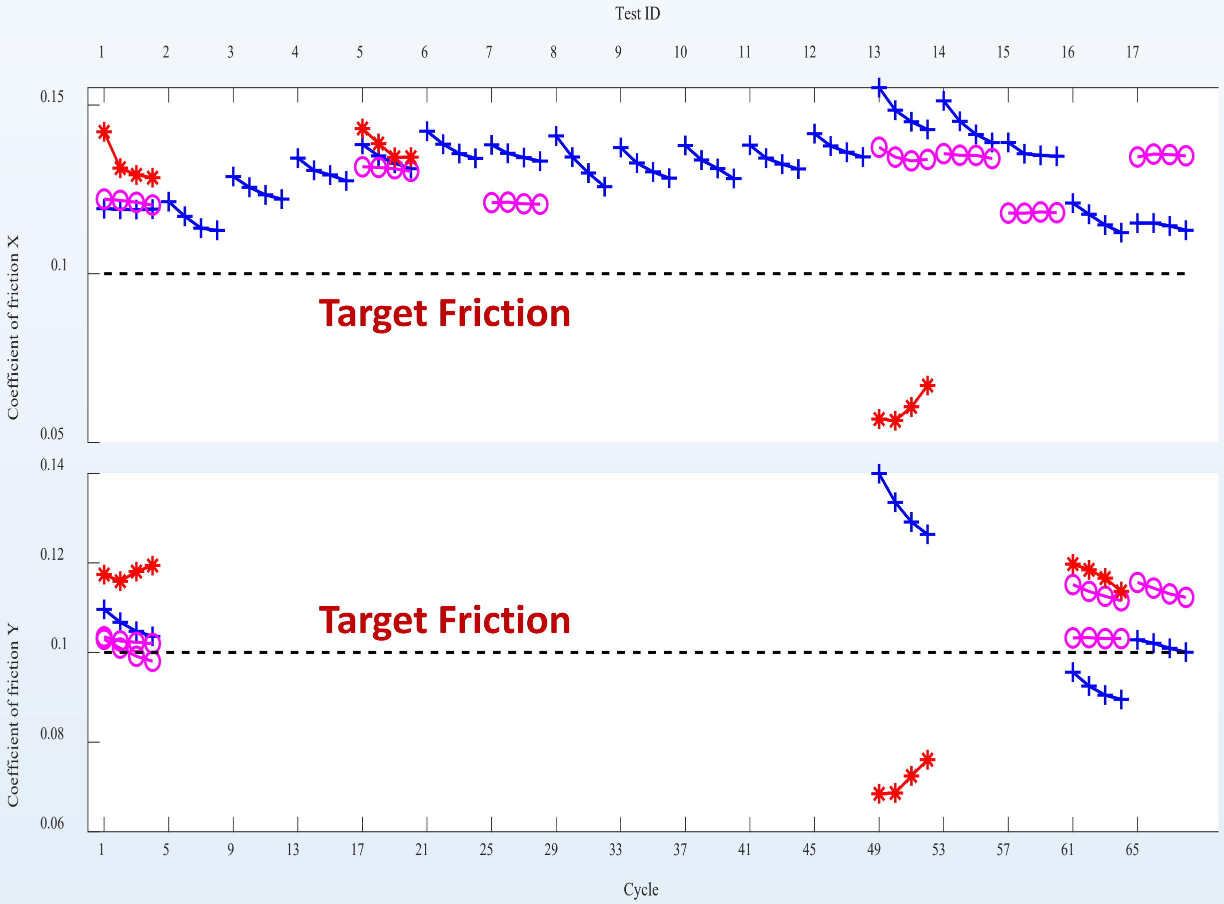
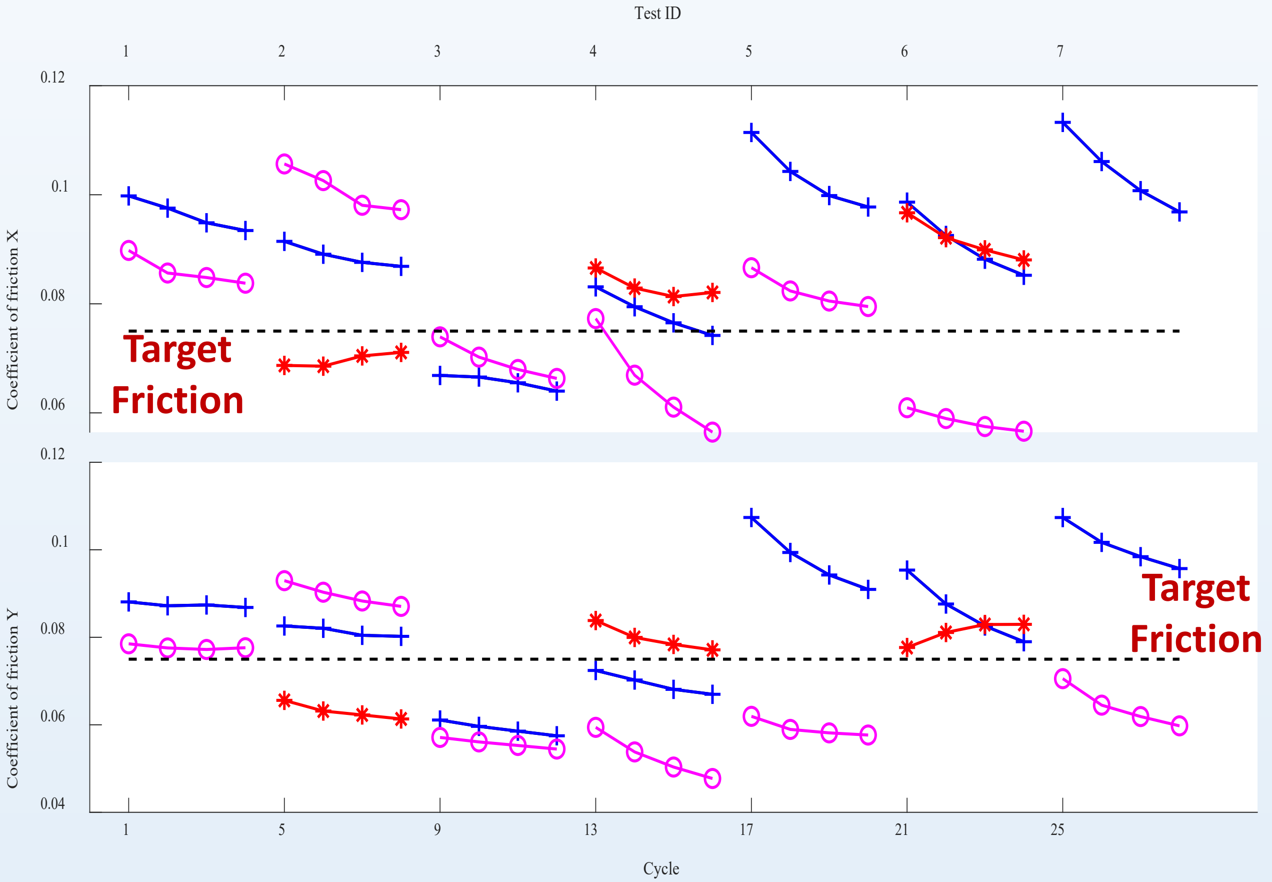
Robertson



Old Susitna 2



- Robertson does not exhibit the same trend for reduced friction in wet sliding vs. dry sliding. It is not known why and if it is related to SPB vs. DPB. More data is desired to verify the trend.
- The method of characterization is as described earlier, and produces an average friction for each cycle that does not control for effects of axial load, velocity or rotation.

Conclusions

Conclusions

- Conditions from the test setup led to unsymmetric hysteresis loops and bearing response that deviates from theoretical.
  - “Field imperfections” could induce some asymmetry and variation in bearing response, but probably not to the extent observed in this experiment.
- For the Susitna DPBs, many instances of mixed sliding and single surface sliding were observed throughout the test program.
  - In the field, similar behavior could be induced by a variety of imperfections (e.g. surface contamination, water or ice contamination, rotation.) Anything leading to unequal friction on the two surfaces.
  - Single surface sliding doubles the post-yield stiffness of the bearings.
- The wet condition induced lower dynamic friction consistently for 3 of the 4 bearings.

Conclusions (Continued)

- A theory of the mechanics of bearing rotation was presented, and bearing top plate rotation could have caused some of the unusual behavior observed in the test program, such as:
  - Asymmetric and shifted hysteresis loops
  - Observed pendulum stiffness differing from theoretical
  - Mixed sliding and single surface sliding of double pendulum bearings.
- However, the phenomenon of friction proves to be very complex, and we were unable to conclusively show that the observed behaviors of the bearing were due to rotation.

Significance of this Work

University of Nevada, Reno

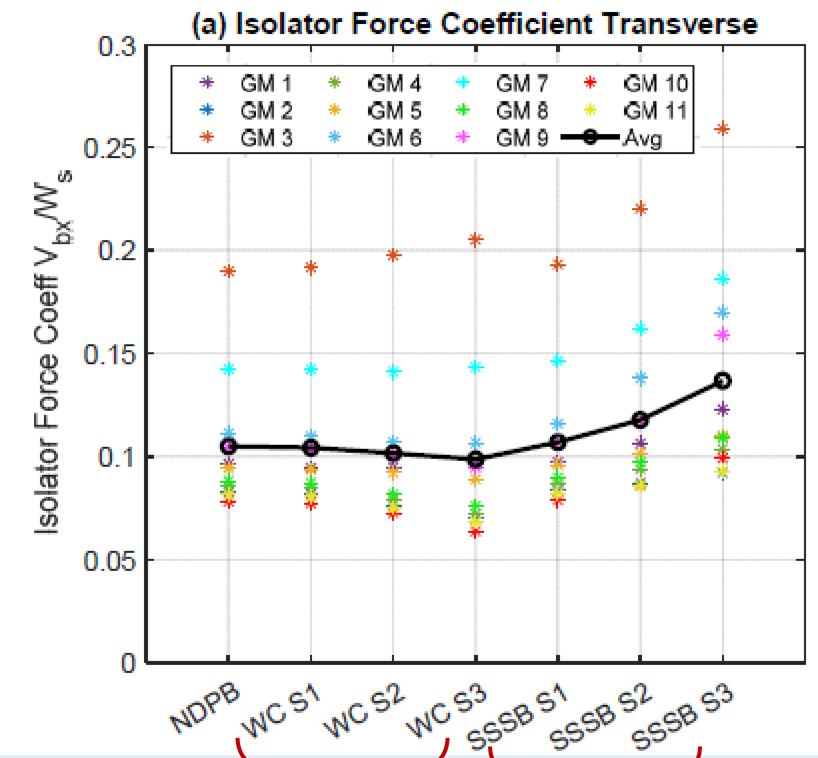
Implications for Modeling and Design

- In the field, isolation bearings may not behave so nicely as they do in the lab. Some examples that could affect response.
  - Reduced wet friction coefficient
  - Increased post-yield stiffness due to single surface sliding
  - Increased resistance due to ice resistance (ice breakaway strength)
- These effects can (for the most part) be accounted for in the design process through property modification factors and bounding analysis.
- But how significant are these effects on the response, really?

Results of a Parallel Modeling Effort

Western Bridge Engineers' Seminar

- We created a few different bridge models, and contamination scenarios to represent each of the effects.
- We evaluated potential changes to isolator displacement, isolator force coefficient, and pier-column force coefficient.



Water contaminated

Single surface sliding

- Sample results: isolator force coefficient increased sometimes substantially under single surface sliding.
- Does this translate to increased force in the bridge pier? It depends.
  - If there is no (or little) substructure mass, the pier and isolator forces are essentially identical.
  - If the bridge pier substructure is stiff and massive, its force demand seems to be fairly insensitive to what happens in the isolator.
  - Overall Conclusion: There seems to be some compulsion to better understand how field conditions might affect the response of the bearings.

Acknowledgements

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