NUMERICAL SIMULATION OF MOTION OF HP/LP ASSEMBLY OF FINGER SEALS AND DESIGN CONSIDERATIONS

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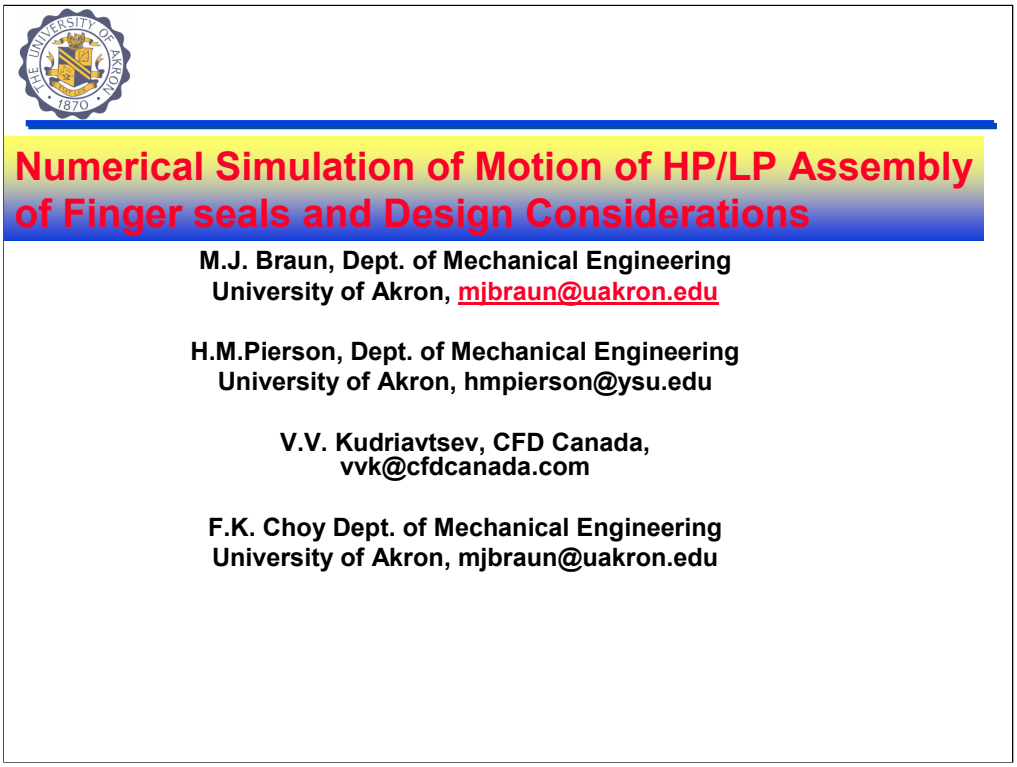


ABSTRACT

- The work concerns the development of the Finger Seal concept and design criteria that ensure finger aerodynamic lifting, while maintaining seal integrity. The FS is a compliant passive-adaptive seal meant to mitigate (and eventually replace) the shortcomings of the entire class of rigid seals used today (labyrinth, honeycomb, mechanical face seals) in the gas turbines and compressors.

GOALS

- **First**,

  we are aiming at developing a fully integrated numerical 3-D model, which couples the hydrodynamic fluid model (Navier-Stokes based) to the solid mechanics code that models the compliance of the fingers.

  The coupled codes that feedback in an iterative mode, allow the full simulation of the passive-adaptive properties of this innovative seal.

- **Secondly**,

  experimentally, we shall test alternative models of finger seals in an effort to better understand their sealing and lifting properties, as well as guide and validate the code numerical development.

GOALS (cont'd)

➢**In Year II**, in collaboration with the Seal Team of the Mechanical Components Branch, we shall extend the University of Akron based experimental/analytical program to the High Temperature Test Rig at NASA Glenn Research Center. This will allow moving our technology readiness level from a room temperature laboratory environment (TRL-4) to the high temperature, engine relevant environment (TRL-5).

NUMERICAL SIMULATION COMPONENT MODULES

⇨**Mechanical model of the single finger and assembly of fingers.** This model entail the use in dual mode both of ALGOR and FEMSTRESS to simulate the motion and deformation of single fingers as well as an assembly of HP/LP fingers as they are subject to engine environment pressures (high and low side), hydrodynamic pressures at the finger foot/shaft interface, and Coulomb friction between the two rows of fingers.

⇨**)Hydrodynamic fluid model.** This model uses CFD-ACE+ to simulate the hydrodynamic lifting effects on the finger seal, as well as the primary and secondary leakages as they occur between the fingers and at the shaft/finger foot interface.

NUMERICAL SIMULATION COMPONENT MODULES

⇨)**Solid/fluid Interaction with the Dynamics module**. Through the implementation of a) and b) we shall obtain a fully interactive model that will model the interaction between finger mechanics and the 3-D fluid hydrodynamic behavior In this context we shall generate a complete pressure map of the hydrodynamic pressures ensuing under the finger pad footprint. All external body forces acting on the finger will be accounted for, in this model.

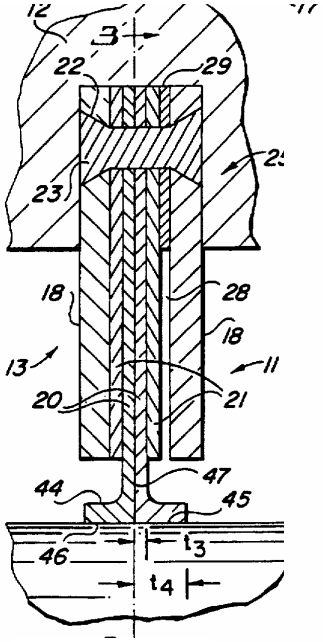
⇨)**Simplified spreadsheet design.** With a), b) and c) implemented we project the possibility that a detailed parametric run will allow creation of a database that can be used for the creation of a simplified calculation methodology that will use a spreadsheet format, without any further need of 3-D calculations.

EXPERIMENTAL PROGRAM

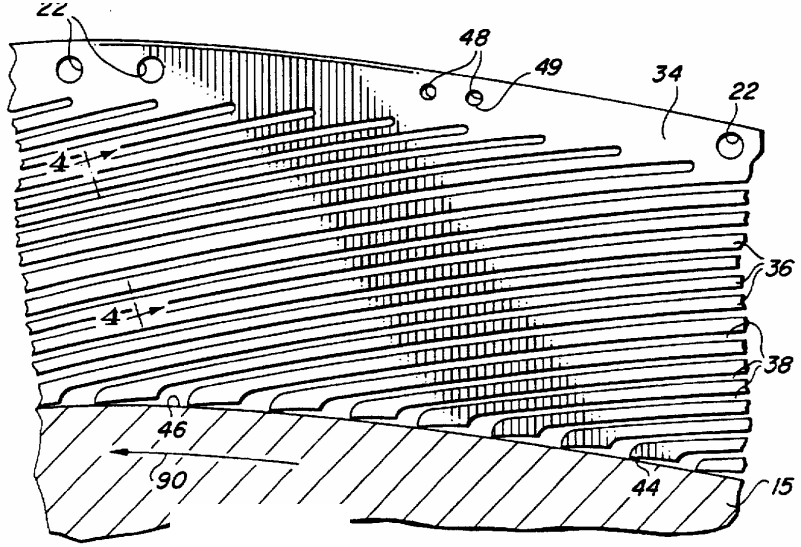
⇨The Tribology Laboratory at the University of Akron possesses a high-speed rig that can be run up to 15,000 rpm. The rig contains all necessary controls and data acquisition system for measuring pressures, temperatures, rotor orbits. The spindle is mounted in cantilever and allows installation of a slip ring at its axial end.

- ➔full pressure and temperature maps
- ➔identification of lift-off and torque characteristics
- ➔high speed visualization of the finger motion and subsequent leakage patterns
- ➔identification of the physics of finger lift-off
- ➔flow visualization of flow patterns before and after finger seal pad lift-off
- ➔effects on sealing efficiency and seal hydrodynamics when
- ➔spiral grooves are etched in the shaft
- ➔grooves are etched on the seal footpads.
- ➔effect of eccentric rotor on seal performance

GEOMETRY OF THE FINGER



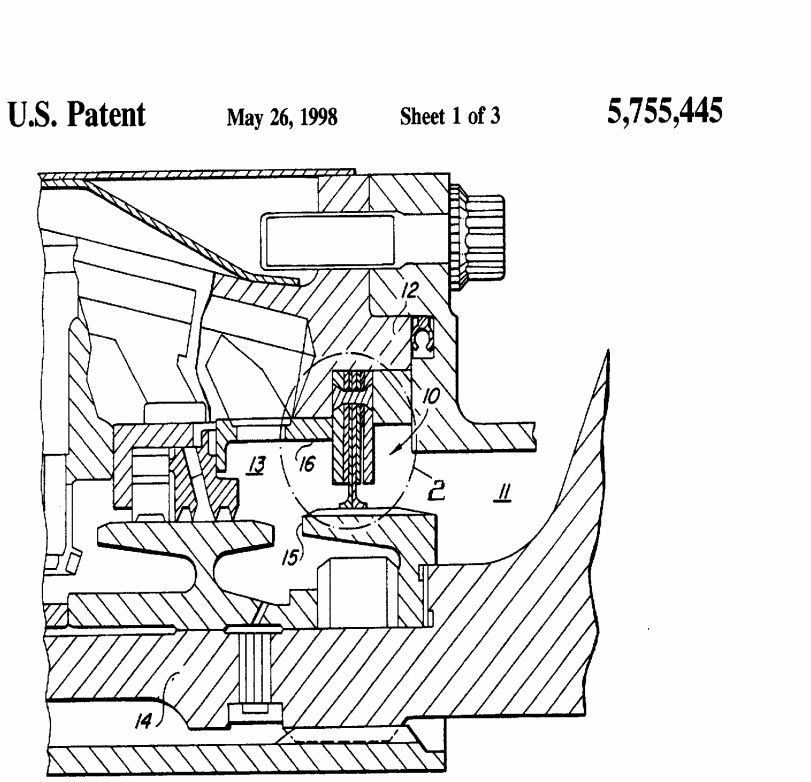
4a)



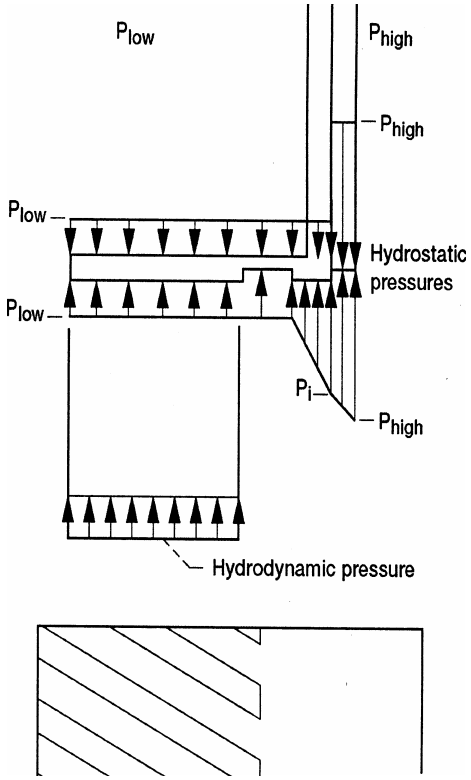
4b)

Seal Two row Configuration with Wide Finger Pads. Cross Section and Side View of the Seal (U.S. Patent No. 5,755,445)

Typical application and Free Body Diagram

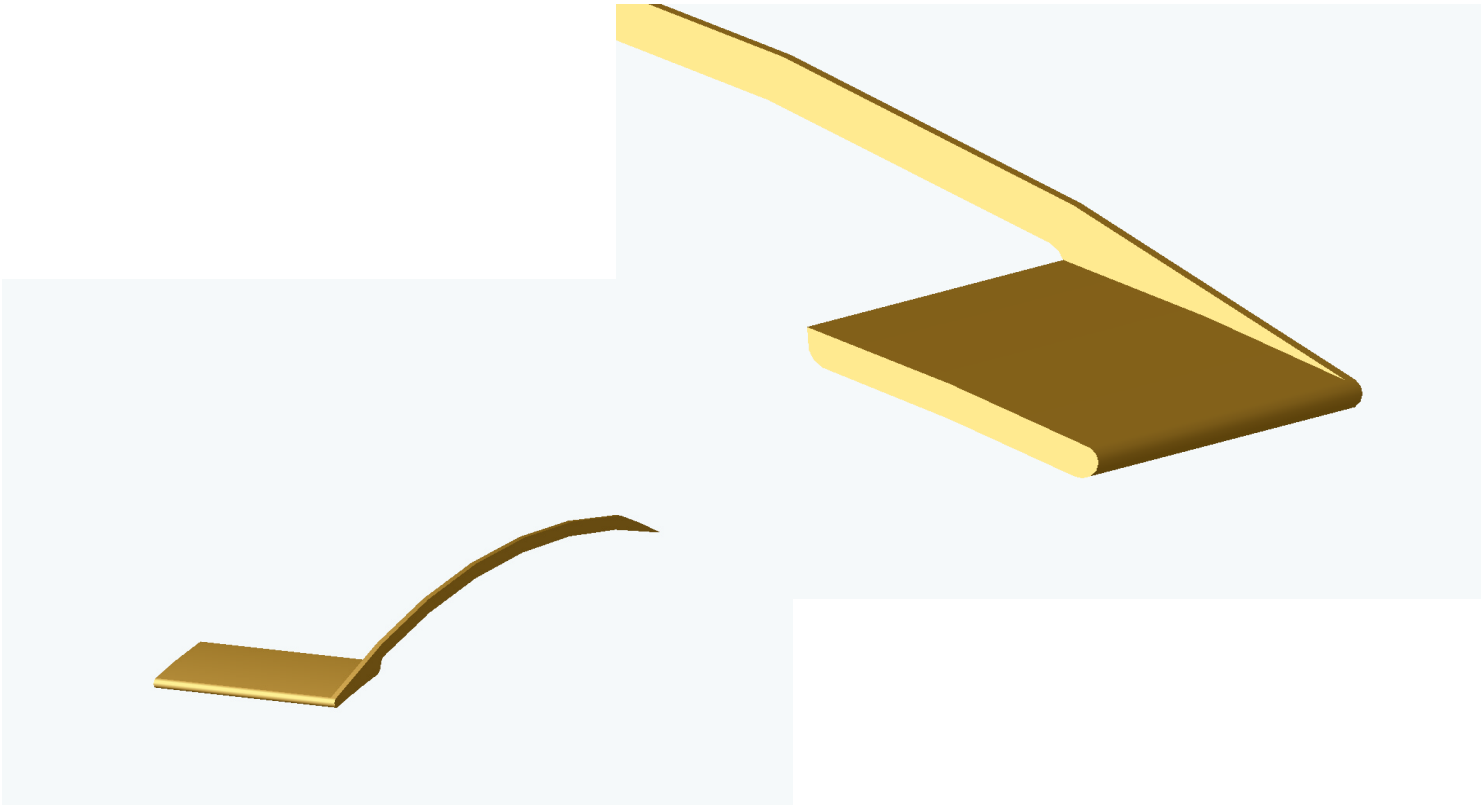


Typical Application of the Finger Seal presented on previous slide (U.S. Patent No. 5,755,445)

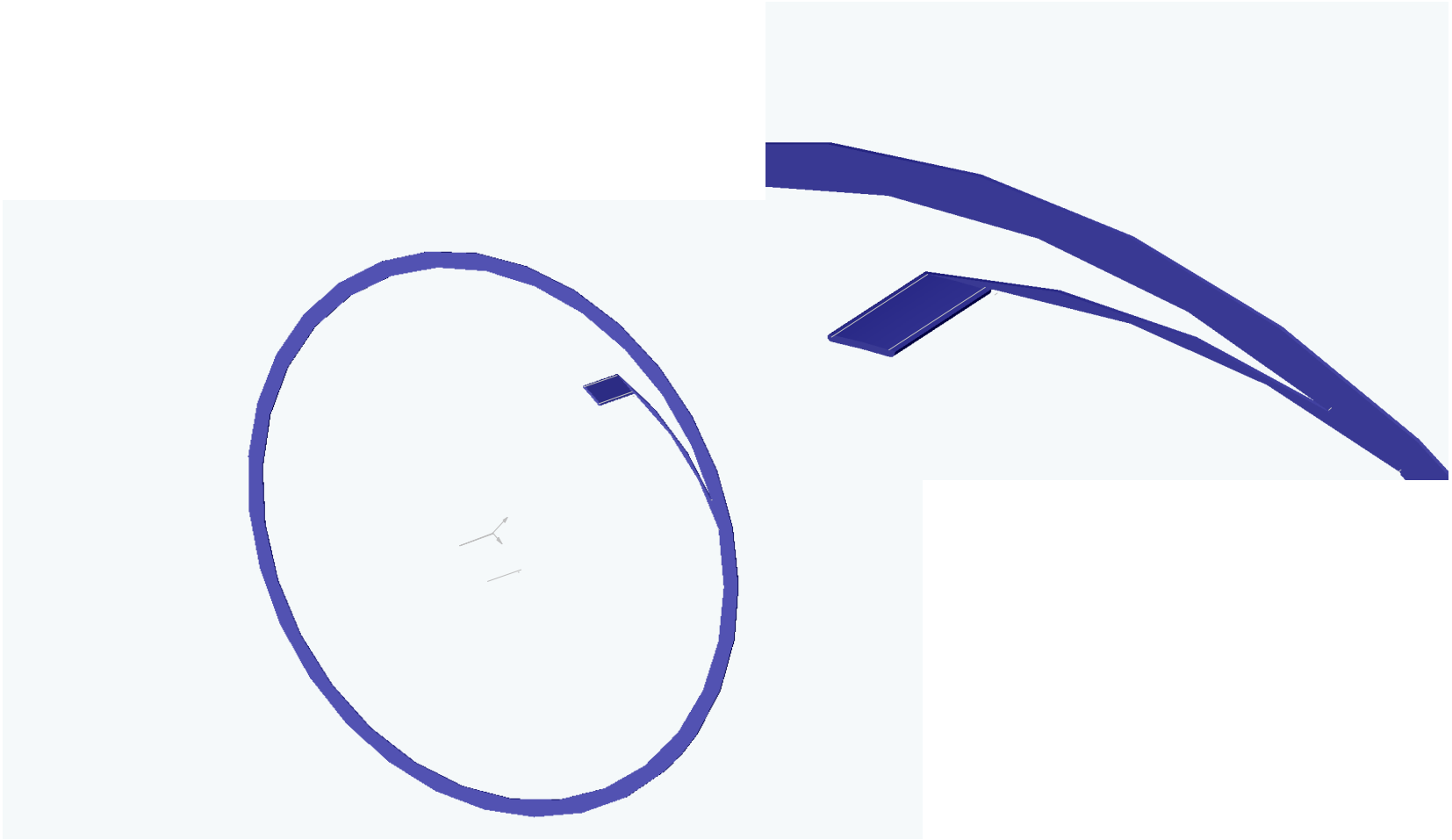


Single Finger as a Free Body Diagram and Geometrical Changes Proposed For Better Wear Behavior

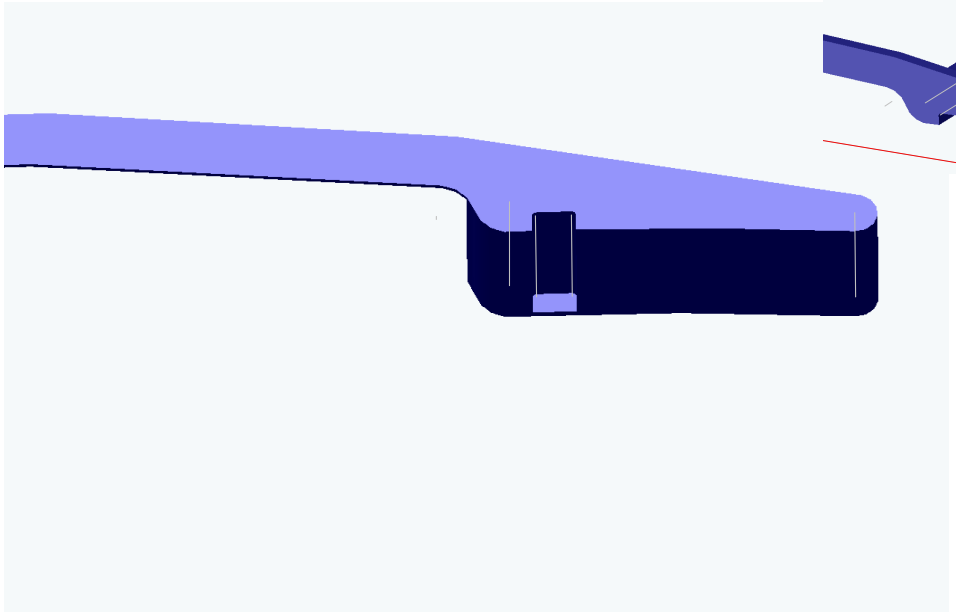
Various finger configurations that are being considered

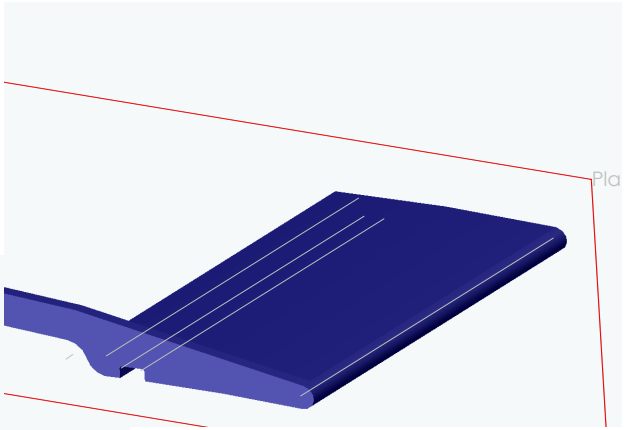


One finger in the rim geometry with basic pad

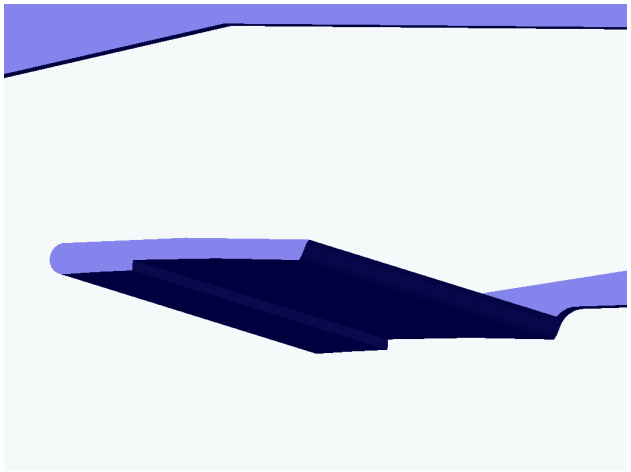


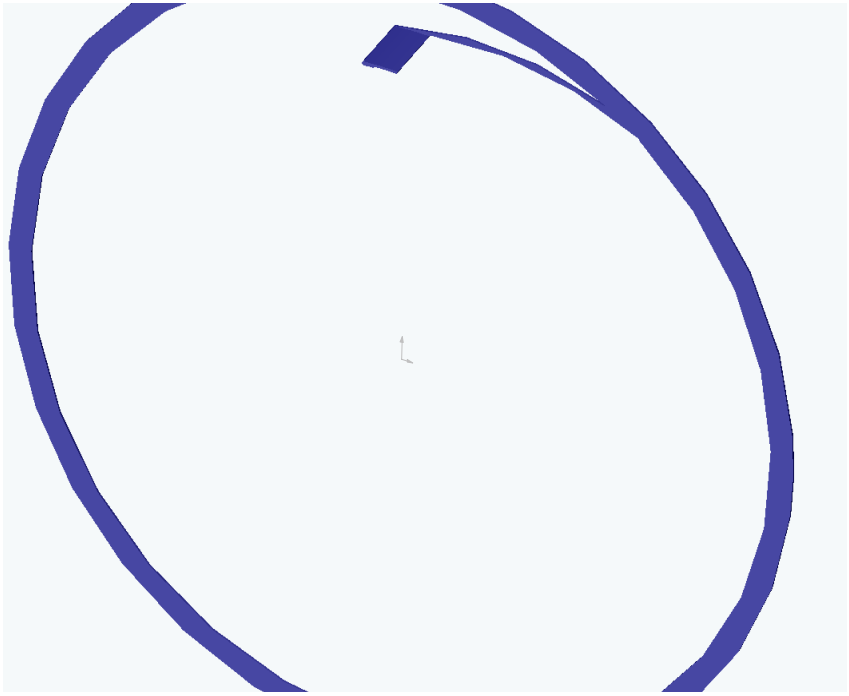
Various finger configurations that are being considered



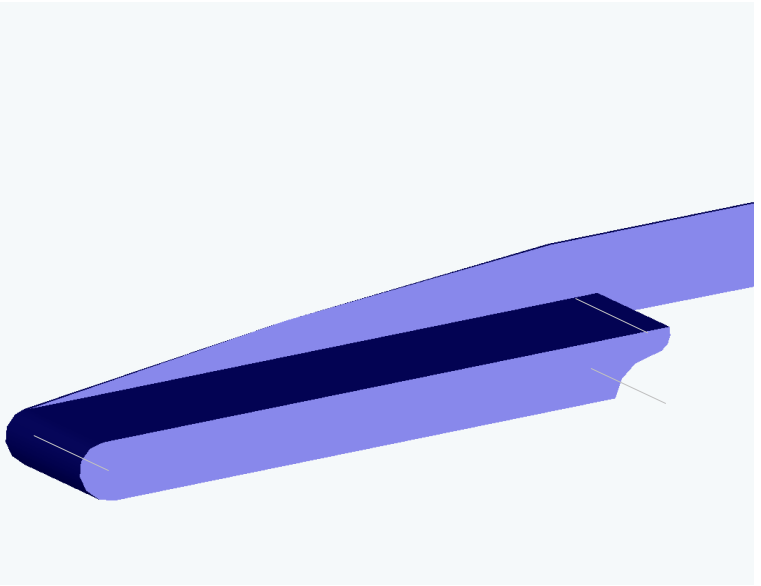


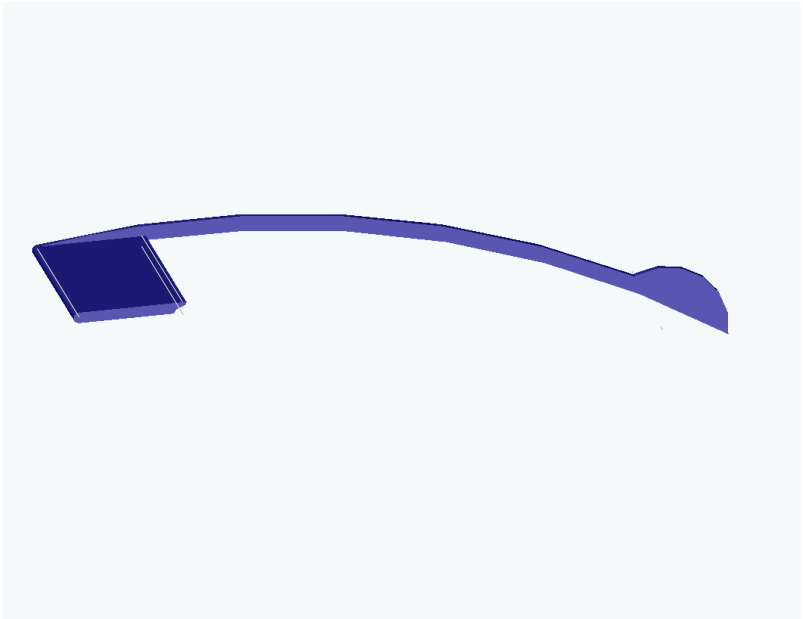
One finger in the rim geometry with Rayleigh pad



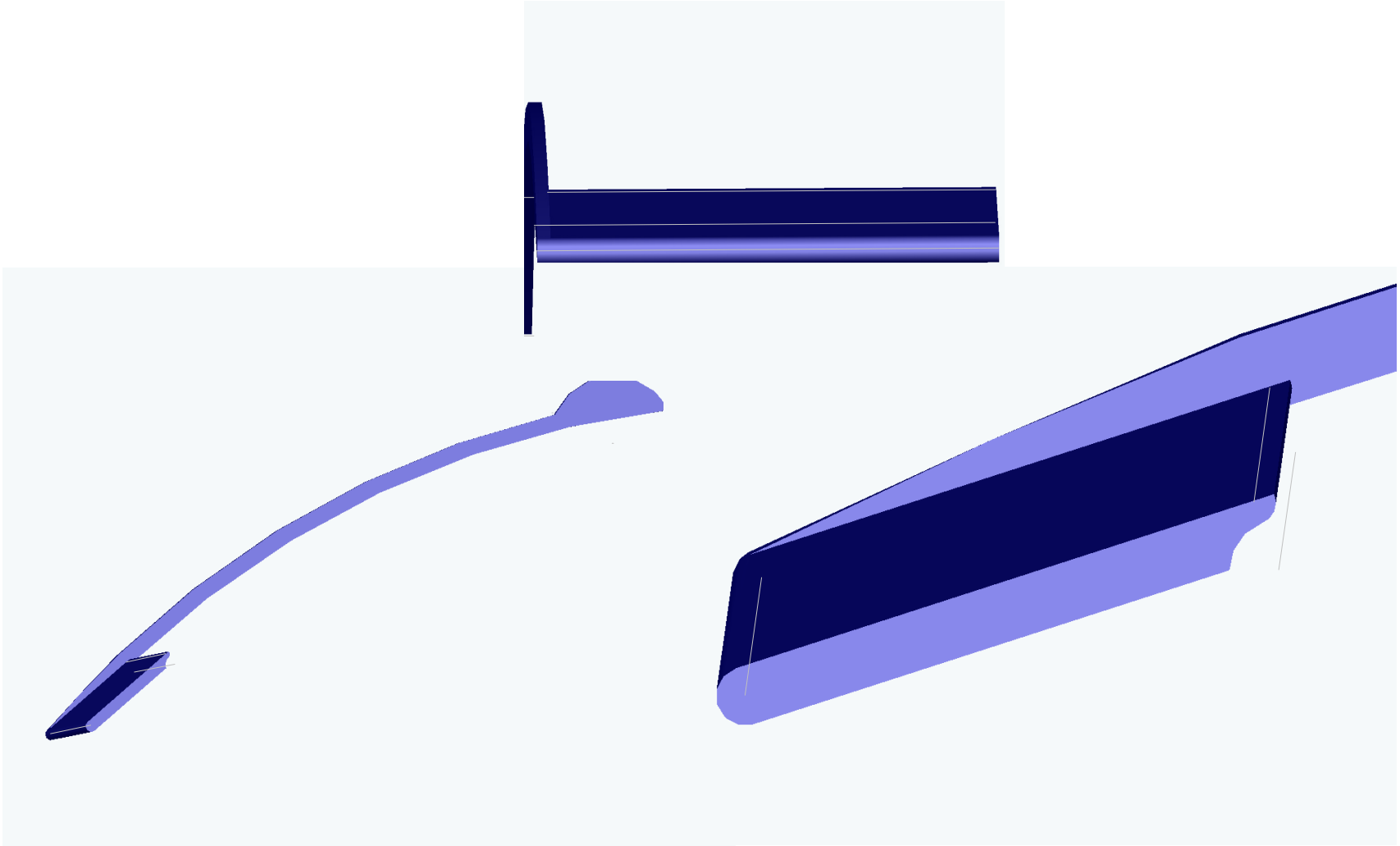


One finger with wedge pad



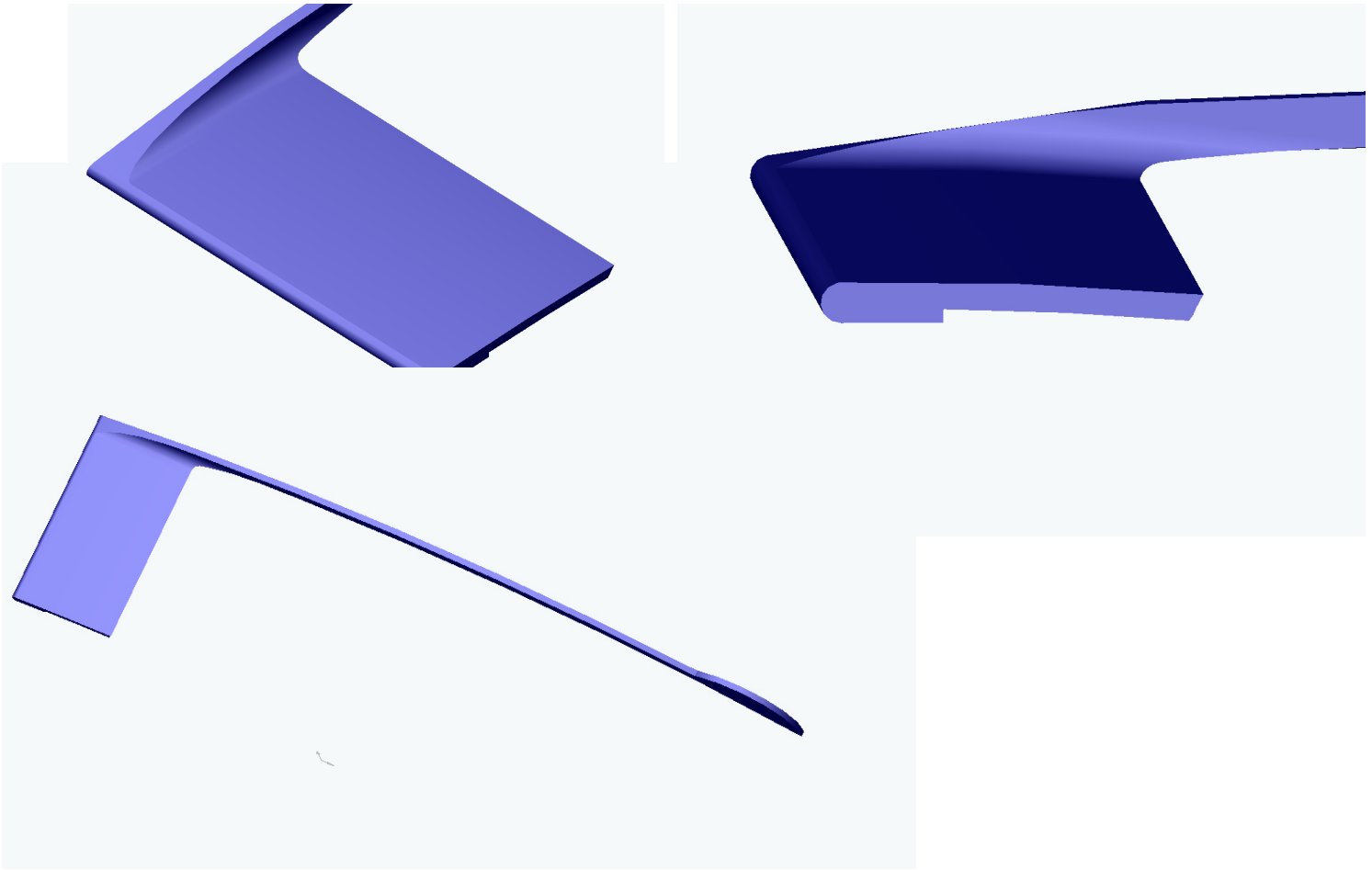


One finger with double wedge pad axial 0.5

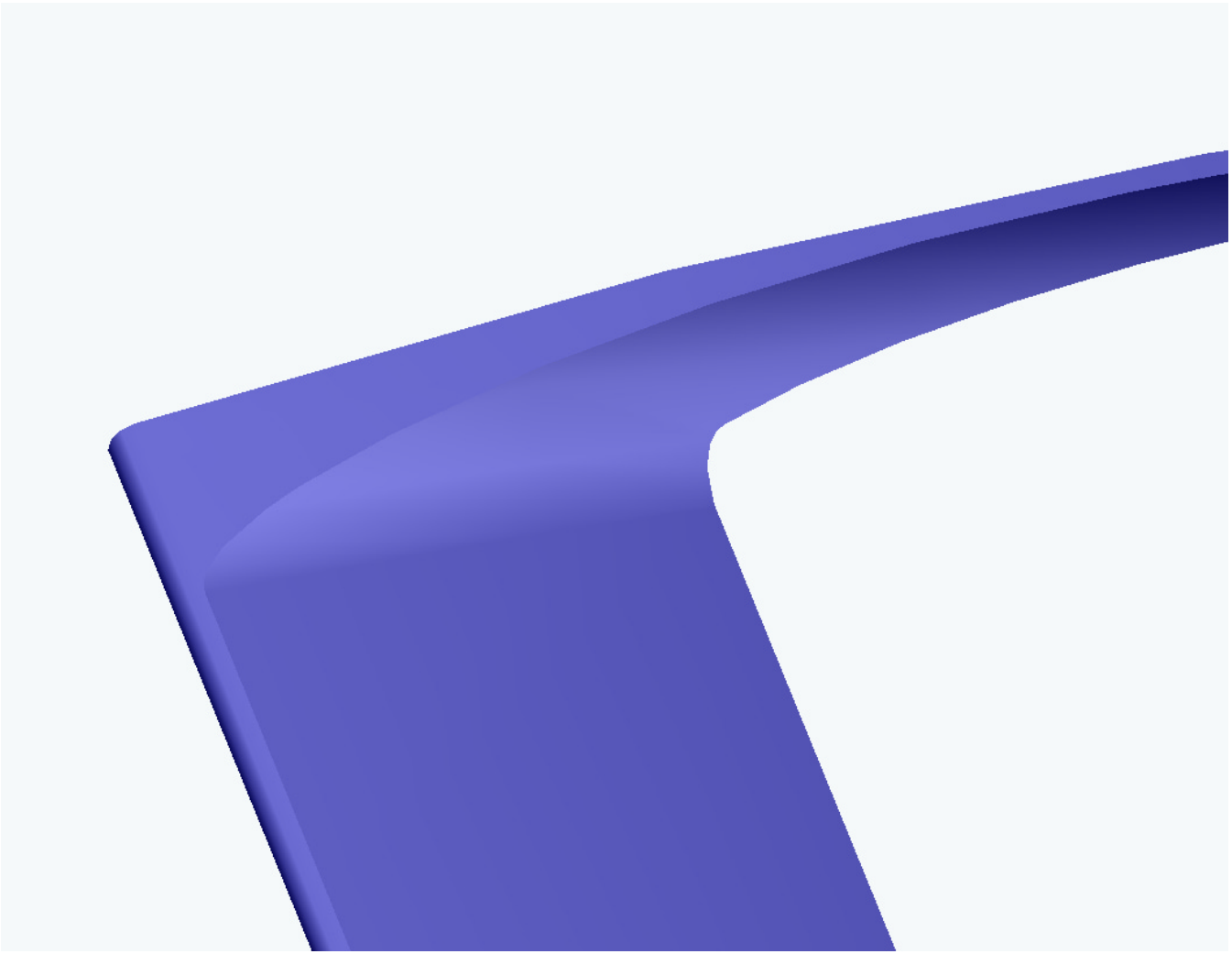


One finger in the rim geometry no pad

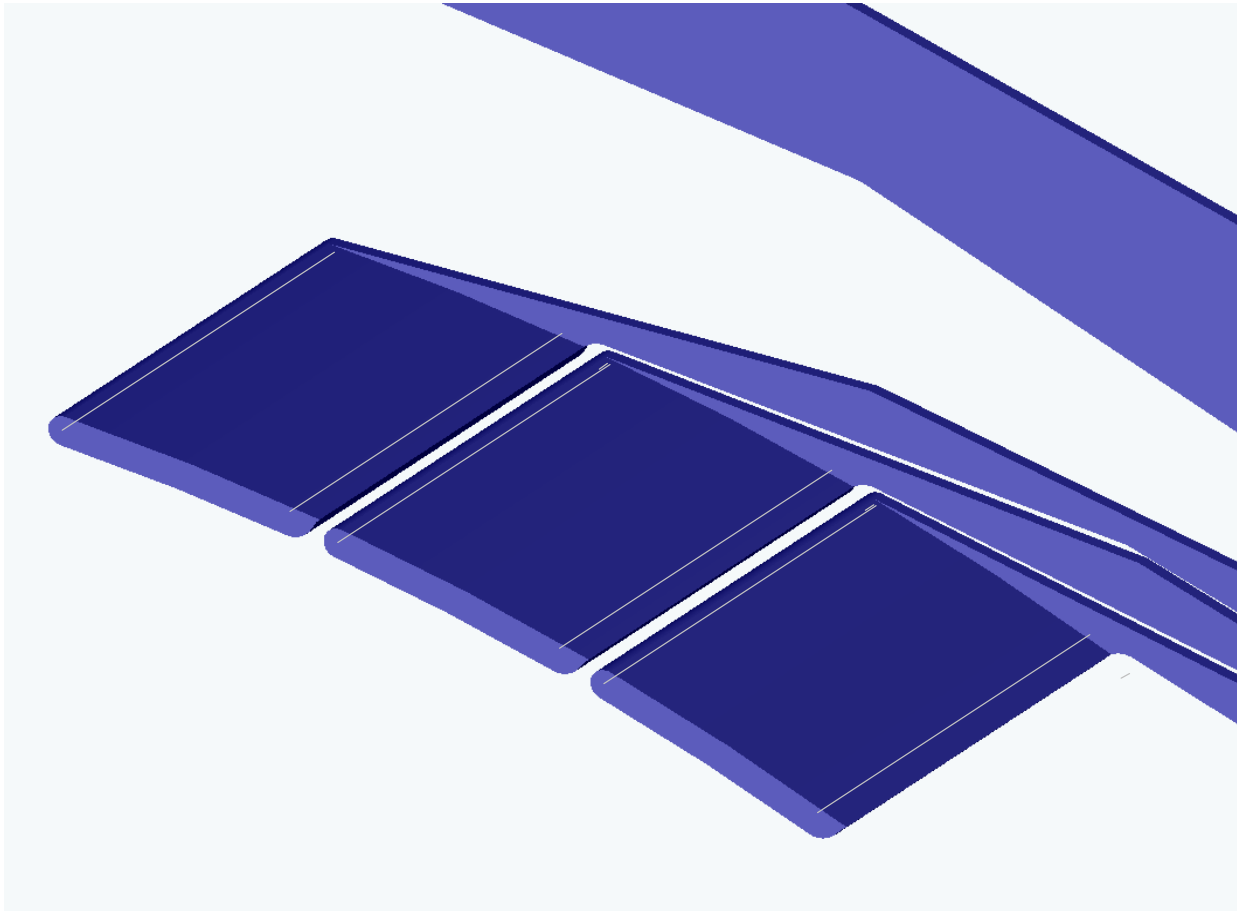
One finger with Rayleigh pad enforced pad R040



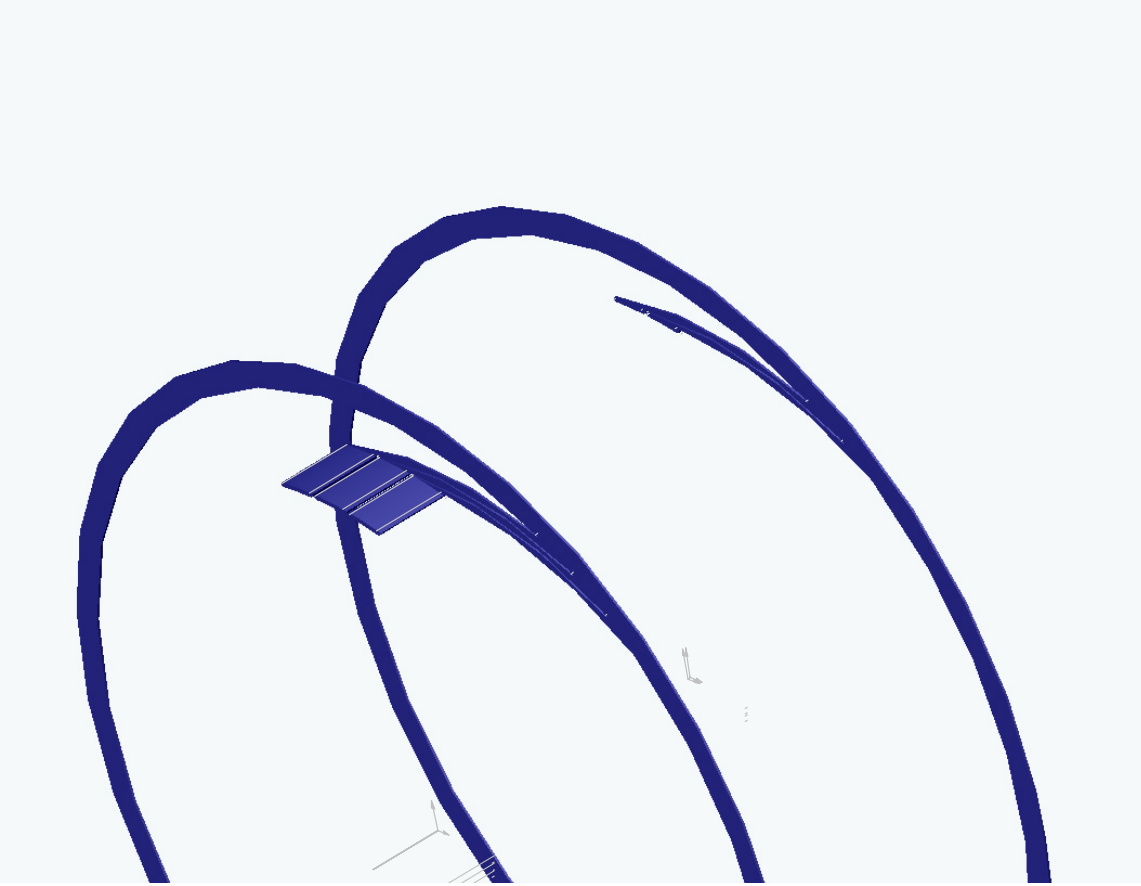
One finger with Rayleigh pad enforced
R 0.100



Three fingers in the rim geometry with basic pad

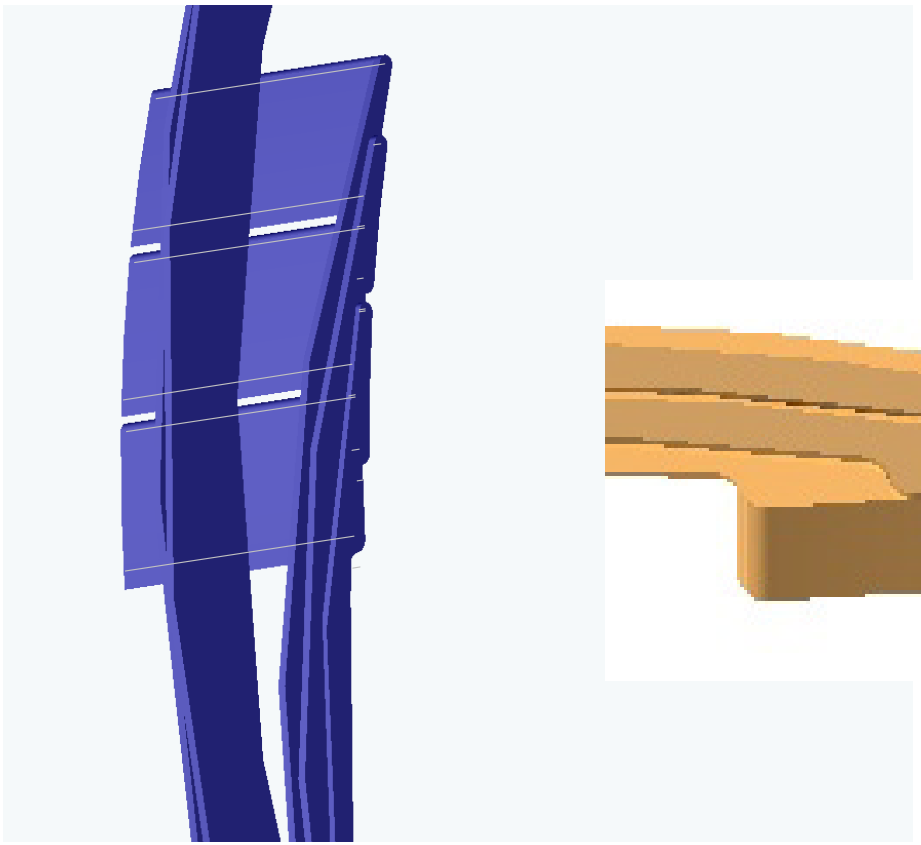


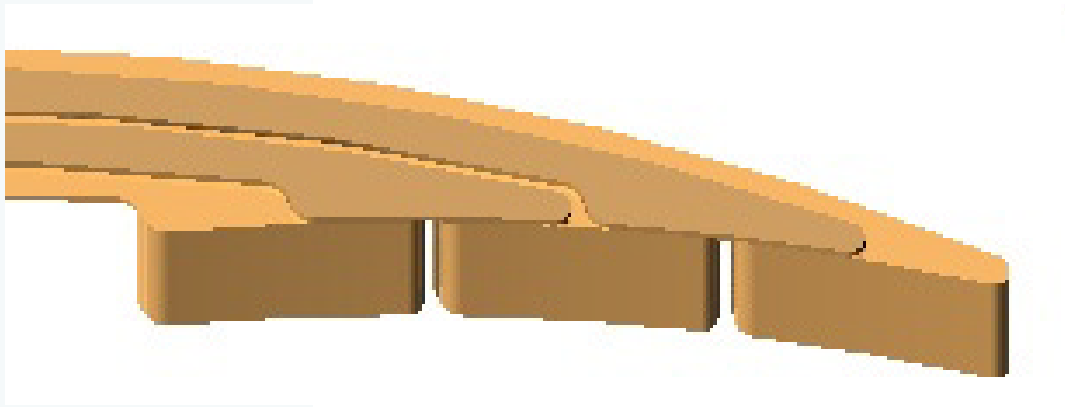
Two rows of basic fingers and pad



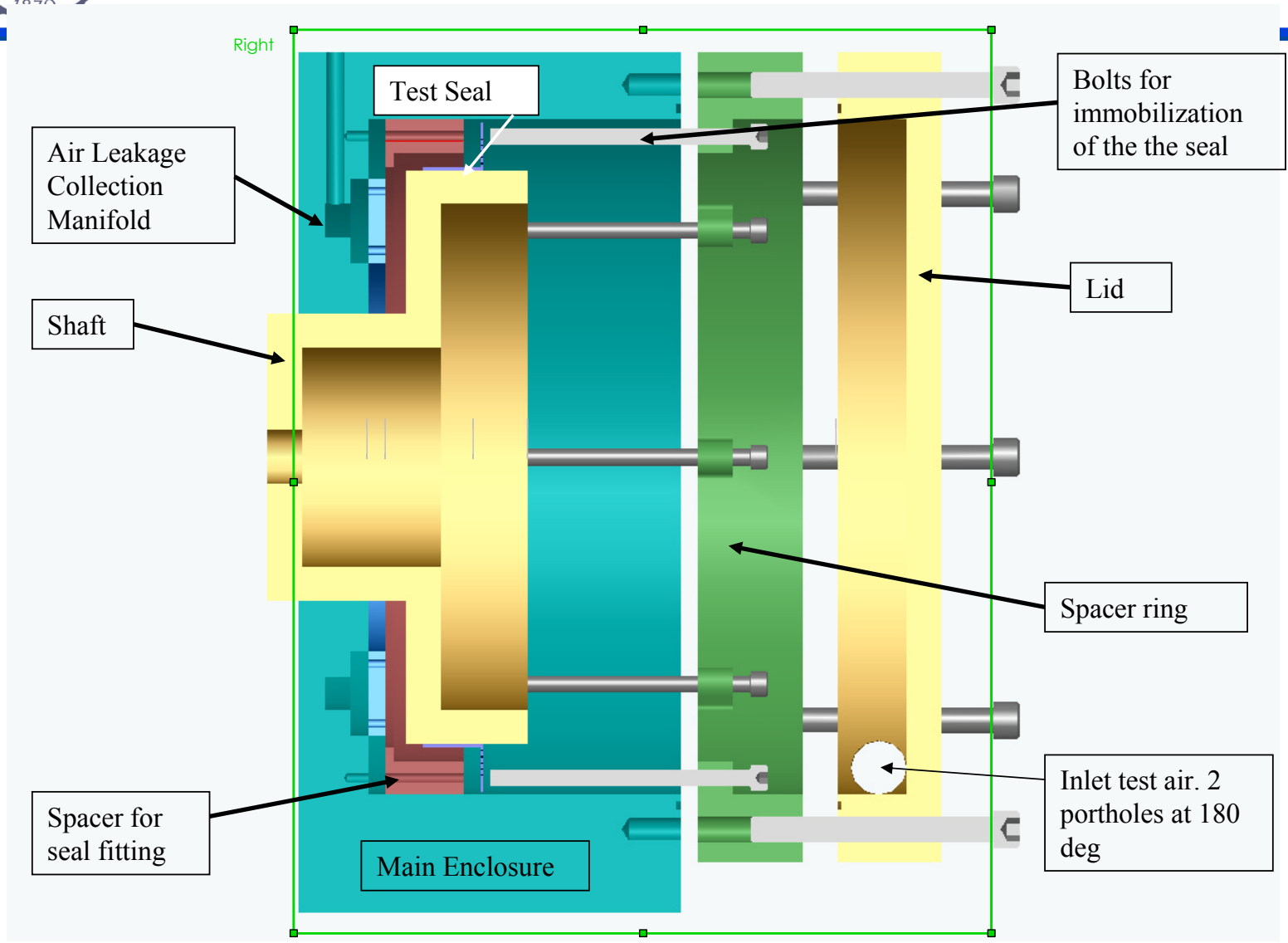
56 fingers in the rim geometry with Rayleigh pad

Assembly of HP and LP fingers

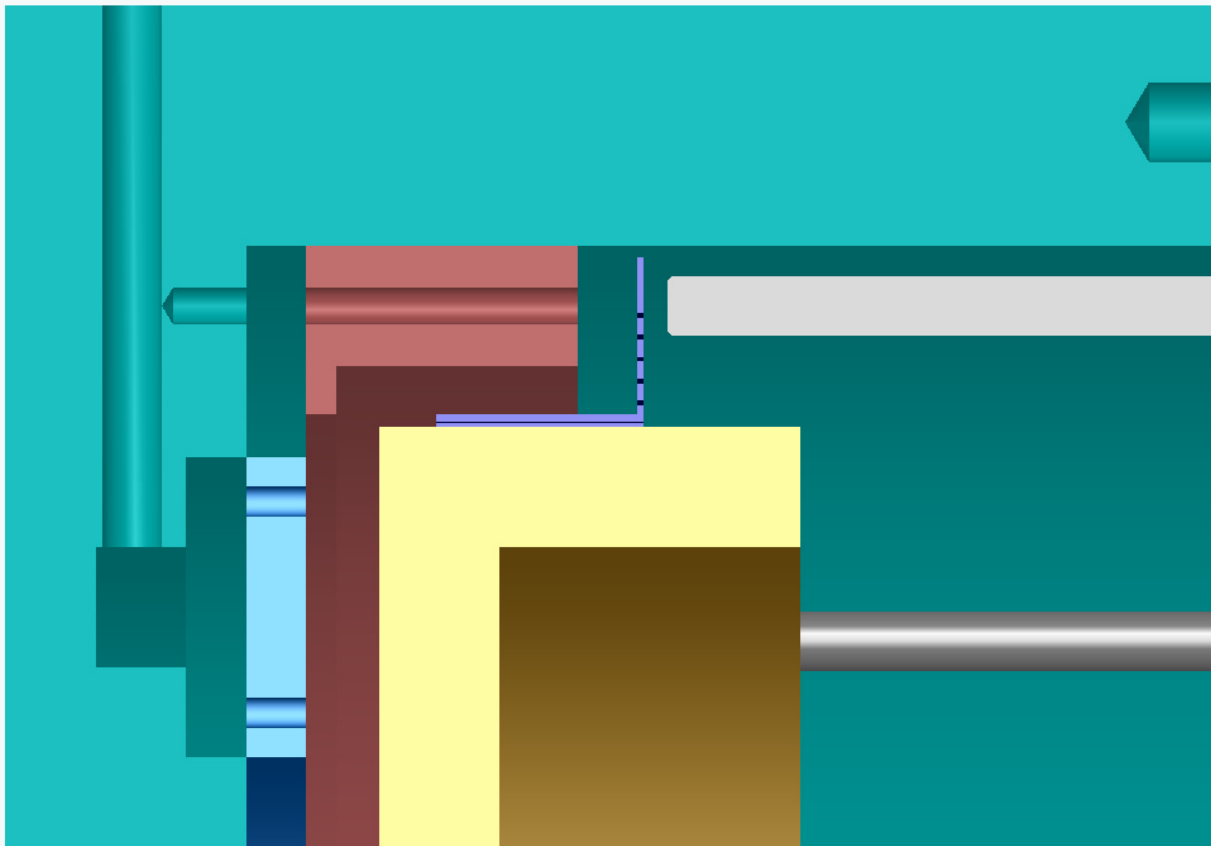




Test Section Cross Longitudinal Section



Detail of the Seal Location



SOME SOLID MODELING USING ALGOR

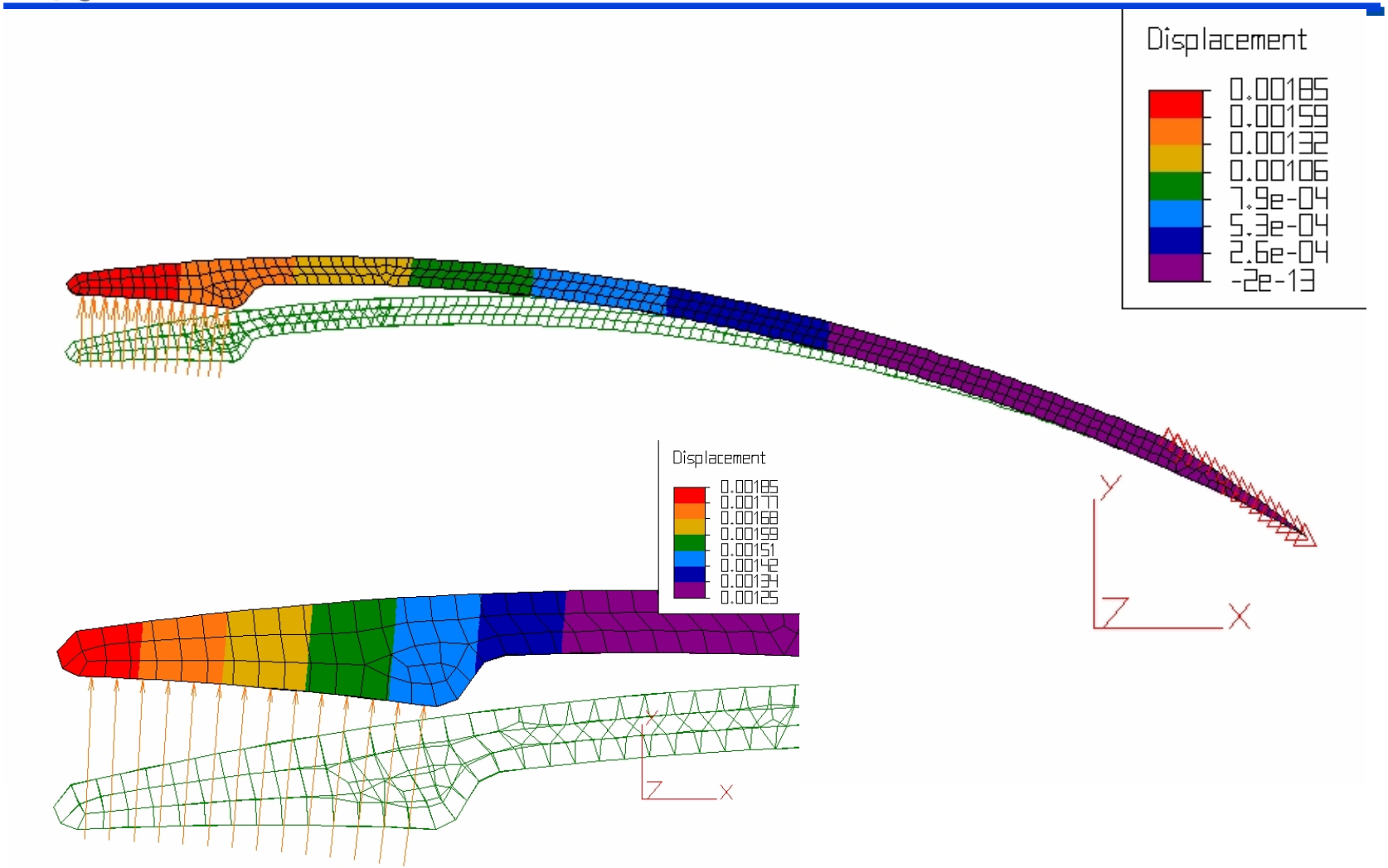
Algor modeling of the finger

These runs were also used to verify the FEMSTRESS results in general

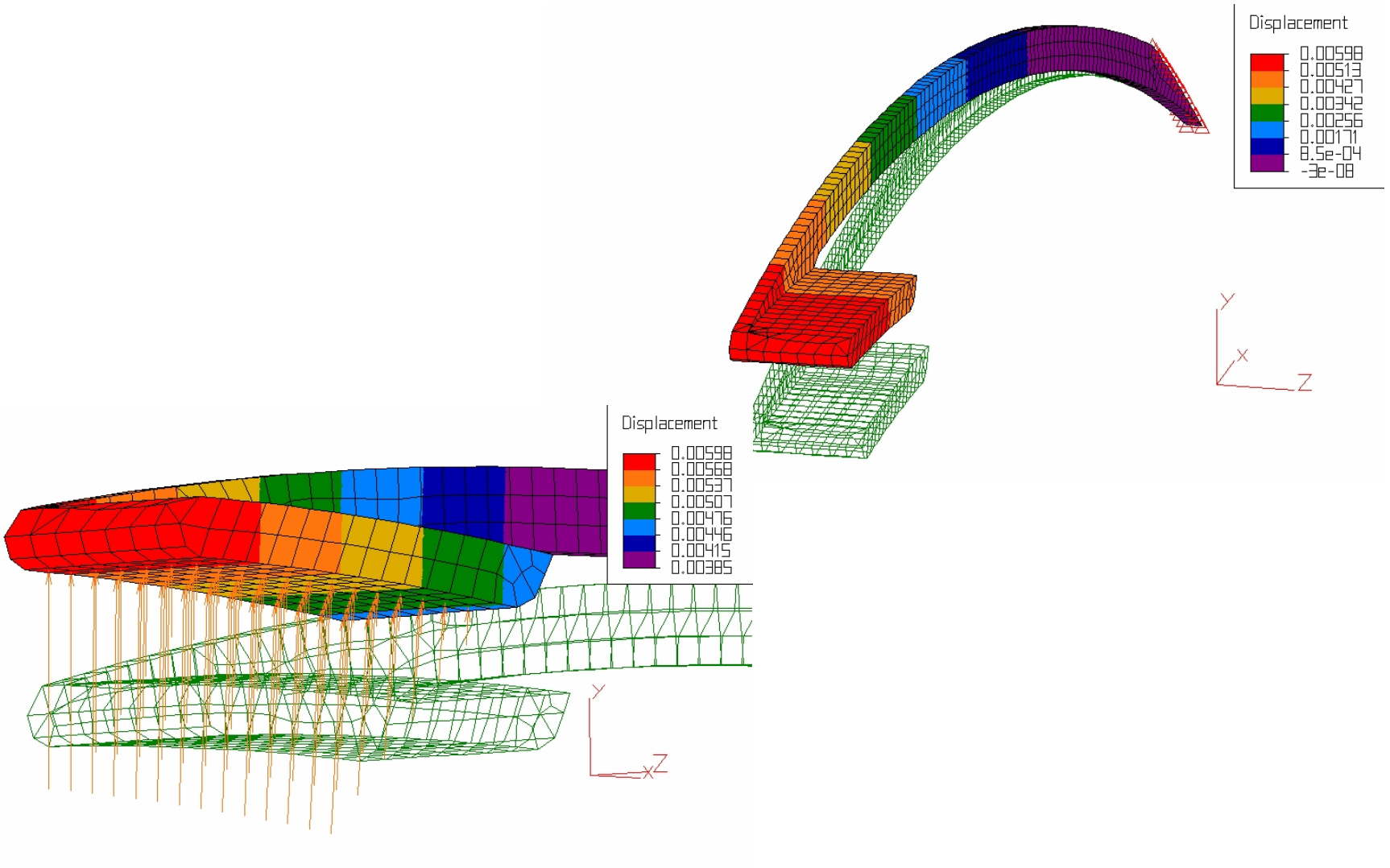
no pad –Existing geometry, ID=5.090 in

the short pad→0.1 in long pad

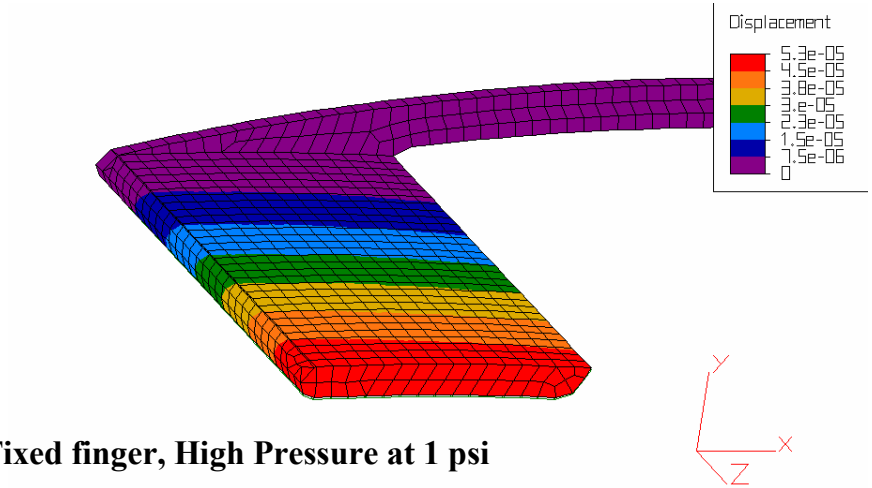
PAD=none
Pressure =1psi



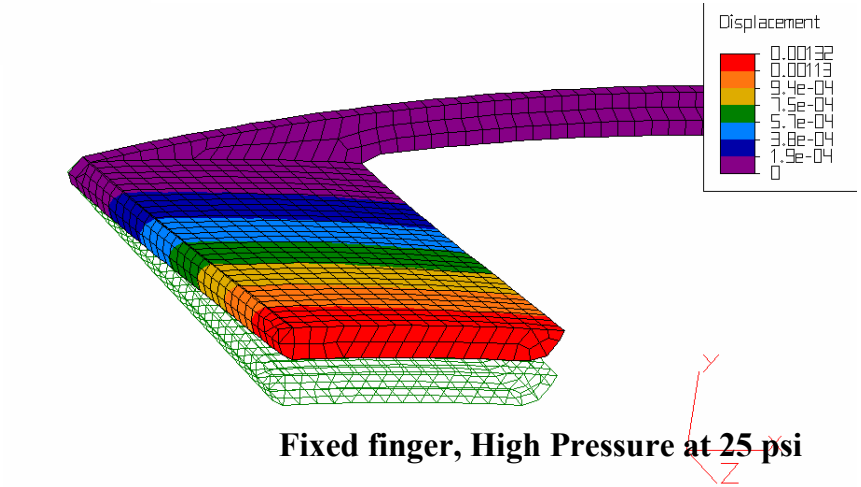
PAD=0.1in long
Pressure =1psi max



Pad deformation; fixed stick; no fillet

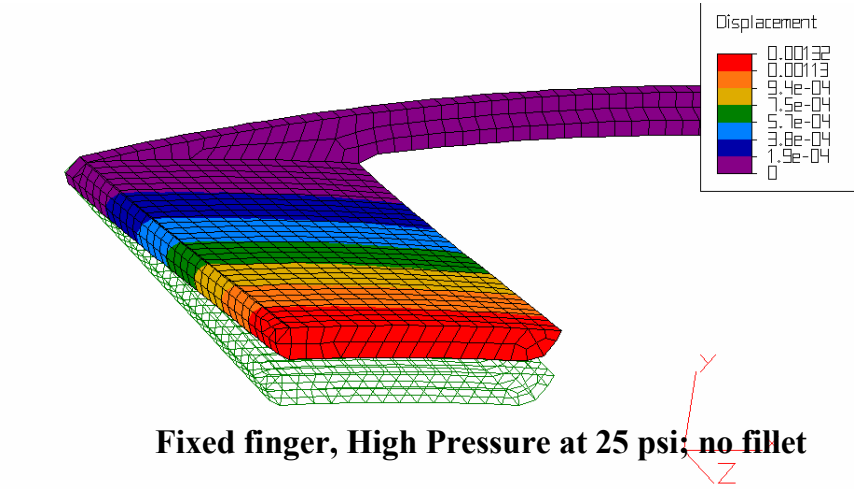


Fixed finger, High Pressure at 1 psi

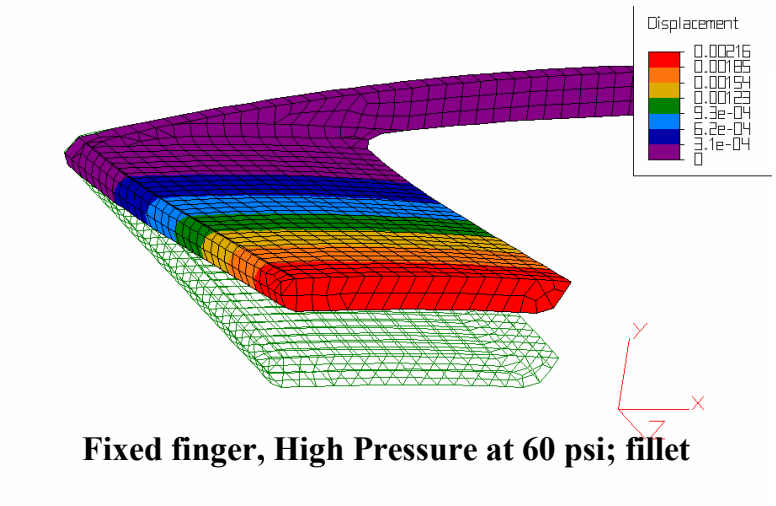


Fixed finger, High Pressure at 25 psi

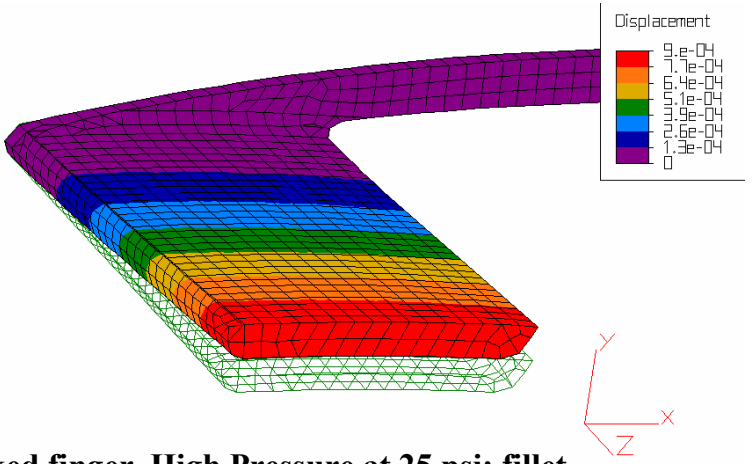
Pad displacement comparison when fillet is added



Fixed finger, High Pressure at 25 psi; no fillet

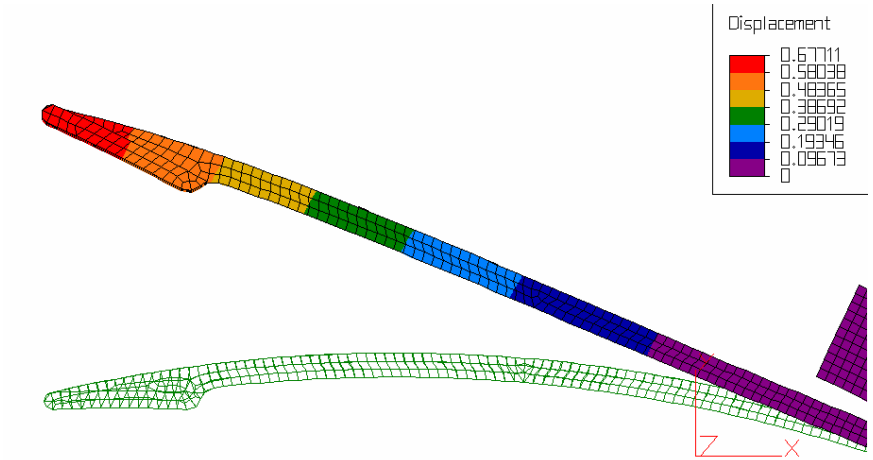


Fixed finger, High Pressure at 60 psi; fillet

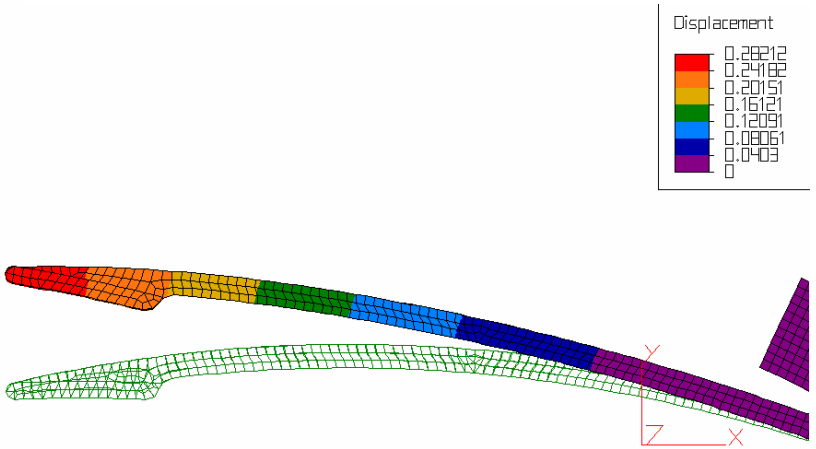


Fixed finger, High Pressure at 25 psi; fillet

Finger motion when Coulomb type friction is applied at 25 and 60 psi pressure differential



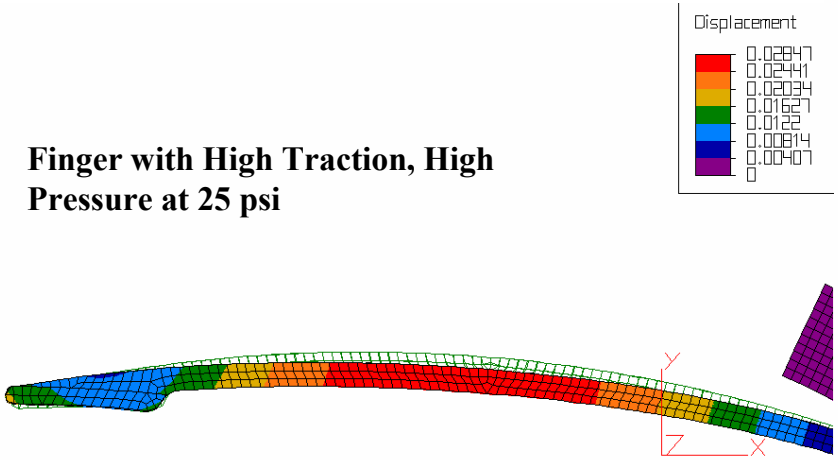
Finger with Traction, High Pressure at 60 psi



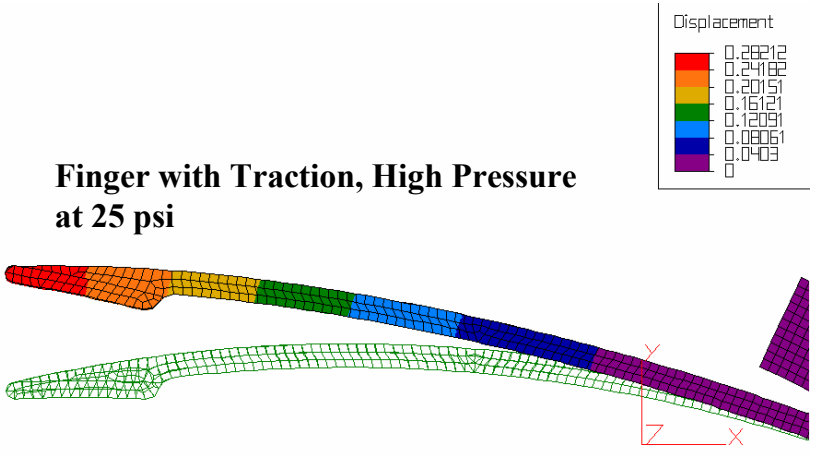
Finger with Traction, High Pressure at 25 psi

Comparison of stick motion with one and two layered fingers.

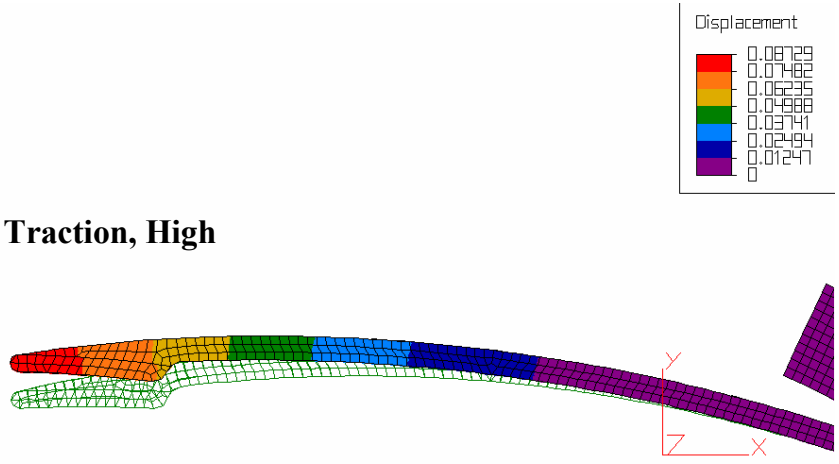
Finger with High Traction, High Pressure at 25 psi



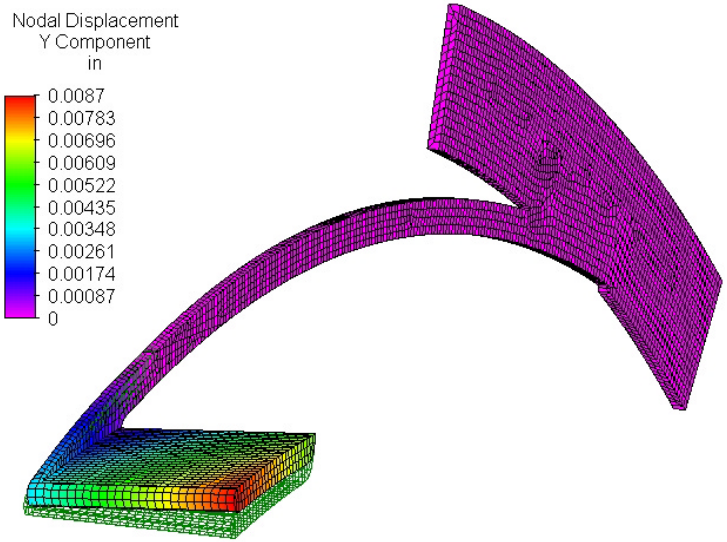
Finger with Traction, High Pressure at 25 psi



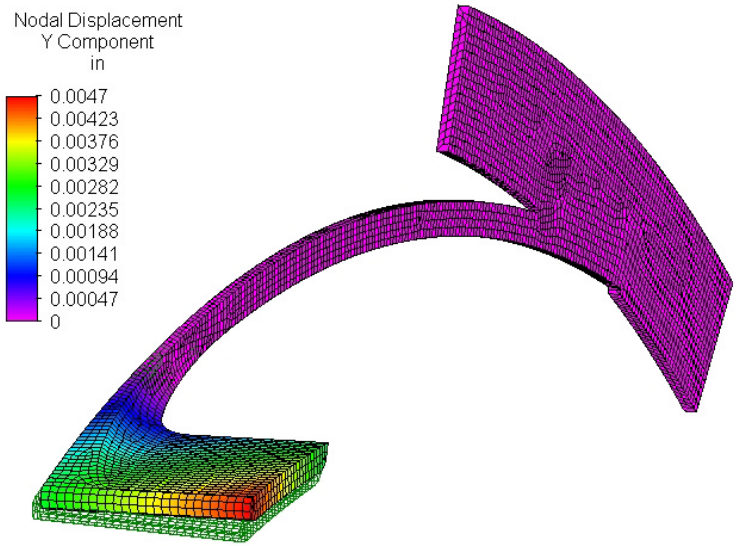
Two layered Finger with Traction, High Pressure at 25 psi



Fully constrained stick; motion of the pad; 25 psi

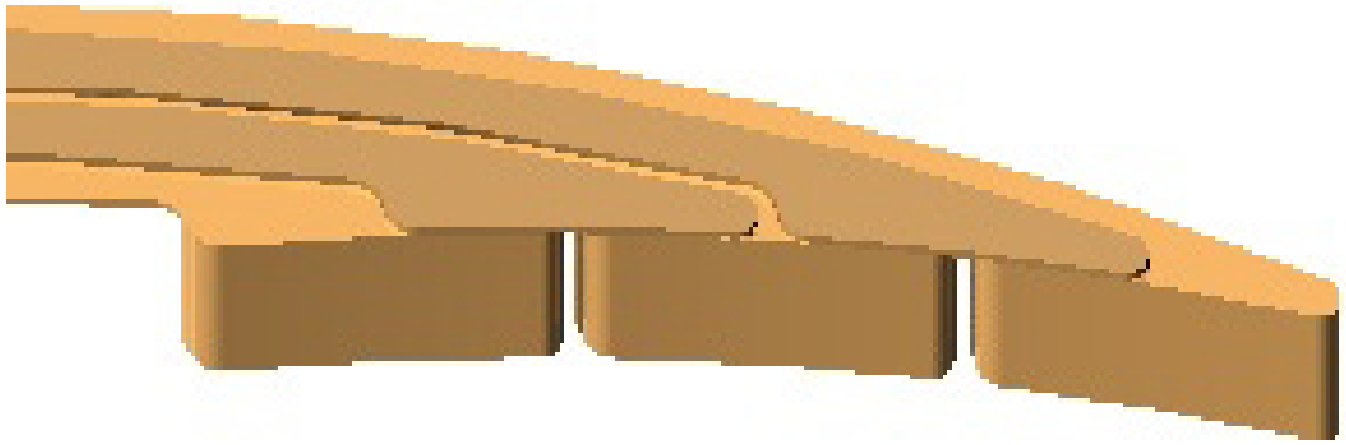


1 low pressure finger fully constrained without fillet

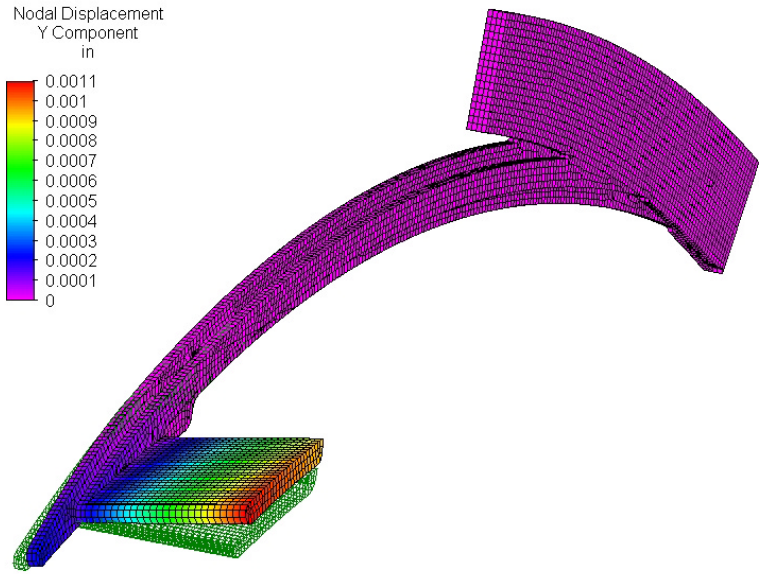


1 low pressure finger fully constrained with fillet

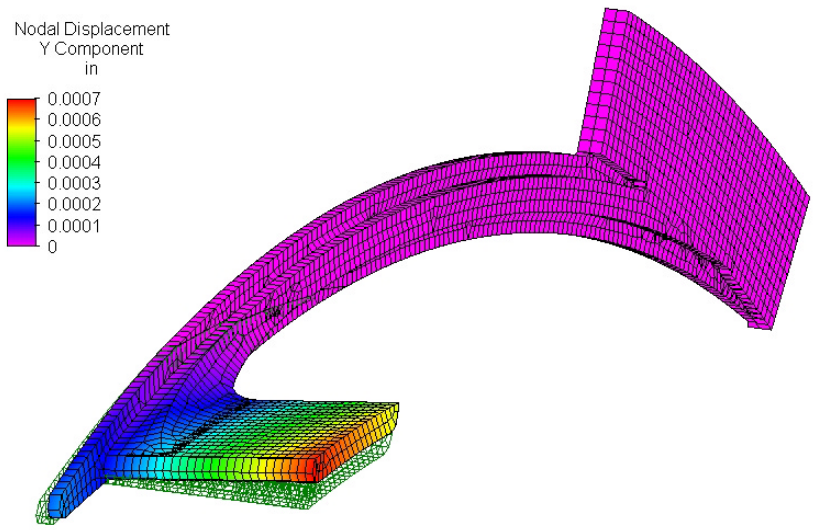
Finger Assembly: 2 HP + 3 LP



Finger Assembly: 2 HP +1 LP; 25 psi; constraint is applied 0.100in from rotor surface

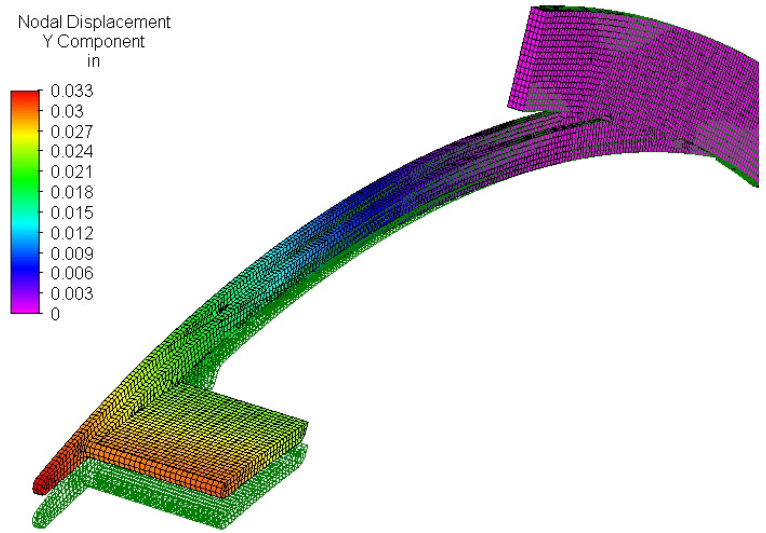


2 high pressure, 1 low pressure fingers fully constrained without fillet

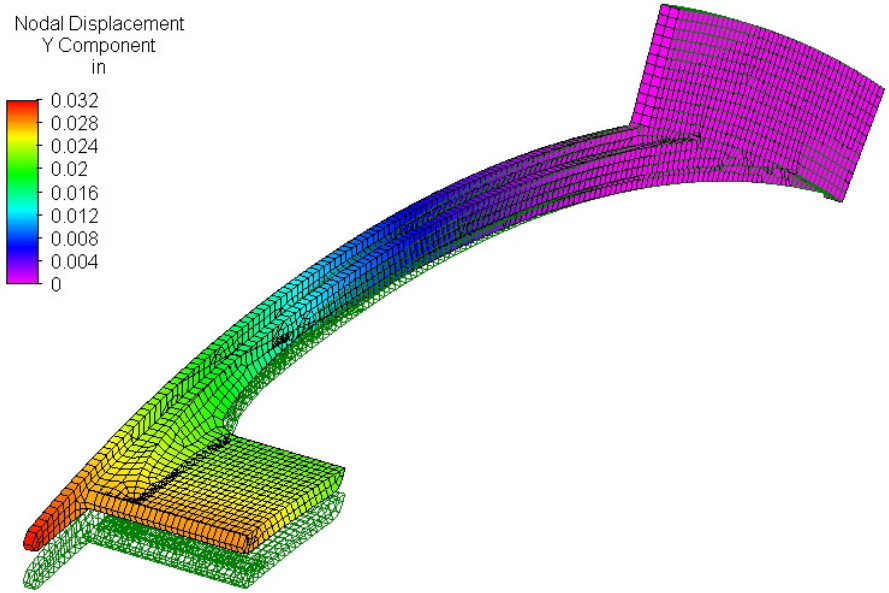


2 high pressure, 1 low pressure fingers fully constrained with fillet

Finger Assembly: 2 HP +1 LP; 25 psi; constraint is applied 0.100in from rotor surface. Traction is 0.3 P



2 high pressure, 1 low pressure finger with traction, without fillet



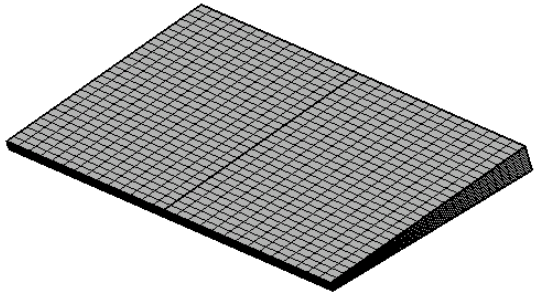
2 high pressure, 1 low pressure finger with traction with fillet

Contents

- Radial Wedge Geometry - without pad/stick deformation
- Radial Wedge Geometry - with deformation
- Radial Wedge Geometry – Two Fingers +Washer
  (with restriction & with contact friction)

Pressure Distribution and Forces the Pad

at 10..20,000 rpm



Operational Parameters

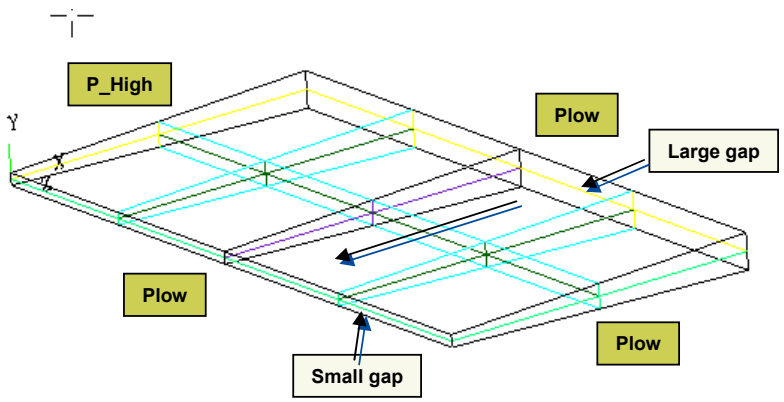
We consider linear runner velocities of 30,60,100 (15,000 rpm) and 135 m/s (20,000 rpm).

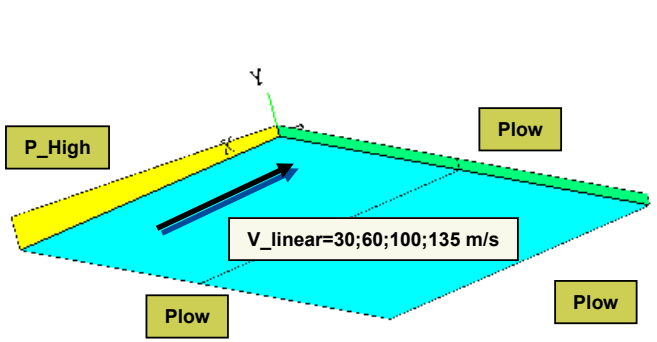
Basic pad surface area is 0.8 cm2 and at average pressure of 10,000 N/m2 (Pa) this constitutes 0.8 Newton (or equivalent of 80 grams of weight). From our previous FEA and FLUID calculations one may expect that average forces on the pad got to be in this ballpark to lift it.

	Parameters:								
R(inches)	R(cm)	L(cm)	RPM1	RPM2	RPM3	V1(m/s)	V2	V3	
2.545	6.4643	40.5958	10000	15000	20000	67.65967	101.4895	135.3193	
	Geometry				10000Pa	30000Pa			
Length	Wleg	W	Area,m2	Area,cm2	Force,N	Force,N			
0.5	0.015	0.24	7.97418E-05	0.797418	0.797418	2.392253			

Radial Wedge –Cartesian Geometry

Small gap=0.25 mil; large gap =0.75 mil;

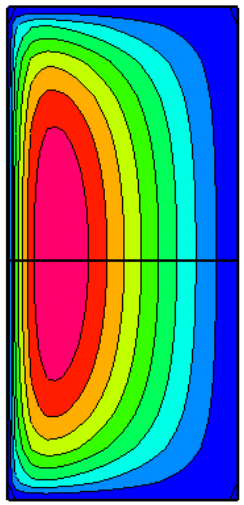


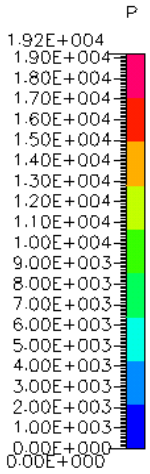


SUMMARY OF FORCES: F=F1+F2		
V=135 m/s	Force1_Y=1.14 N	Force2_Y=1.23 N
V=100 m/s	Force1_Y=0.935 N	Force2_Y=1.006 N
V= 60 m/s	Force1_Y=0.624 N	Force2_Y=0.672 N
V= 30 m/s	Force1_Y=0.335 N	Force2_Y=0.361 N

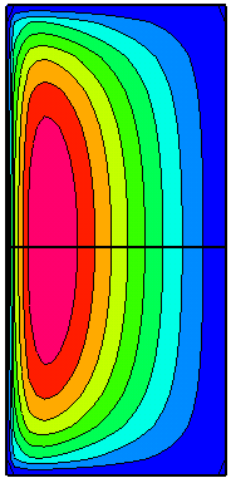
Radial Wedge – Pressures due to Rotation

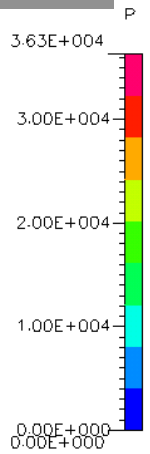
30 m/s



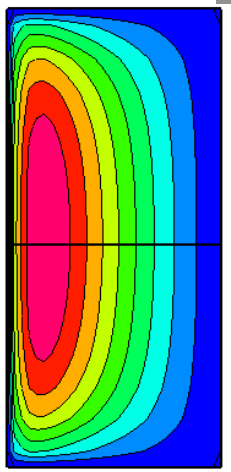


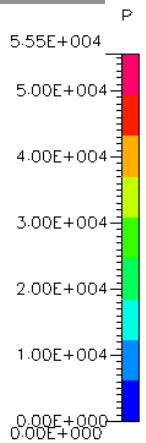
60 m/s



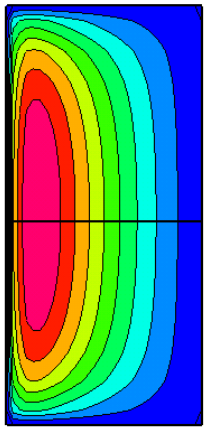


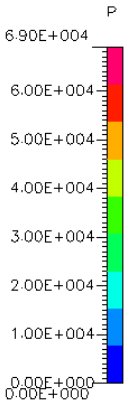
100 m/s





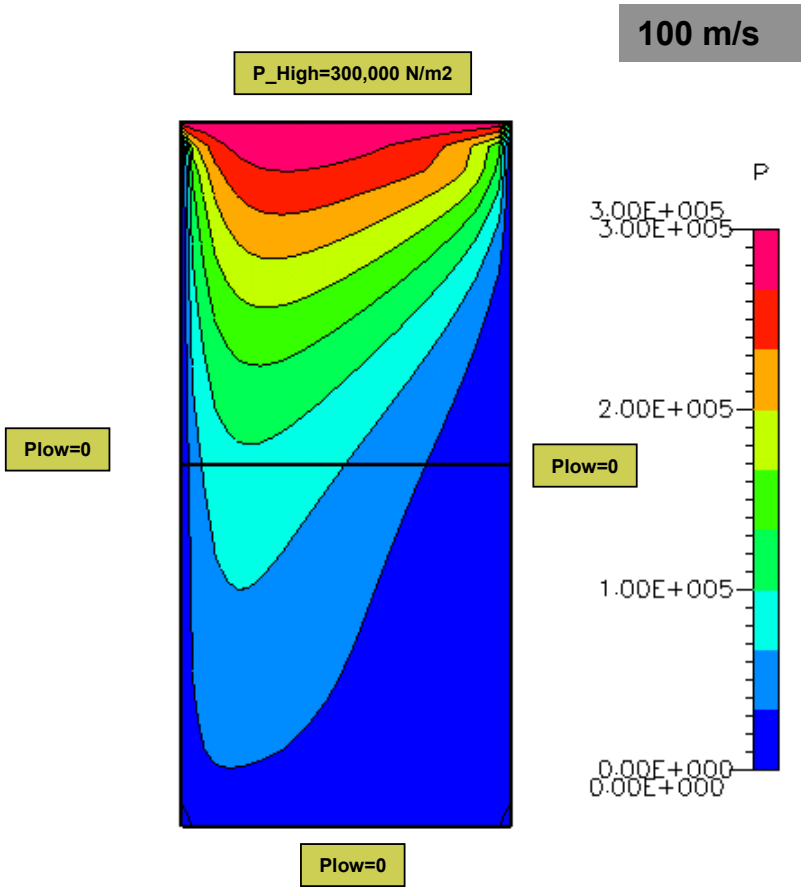
135 m/s



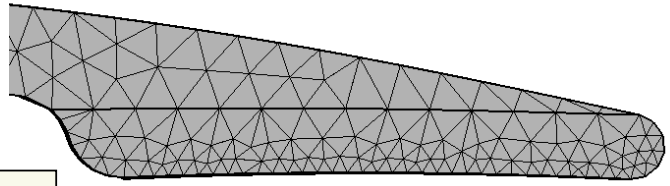


Pressures are in Pa [N/m2], 101000 Pa=14.7 psi

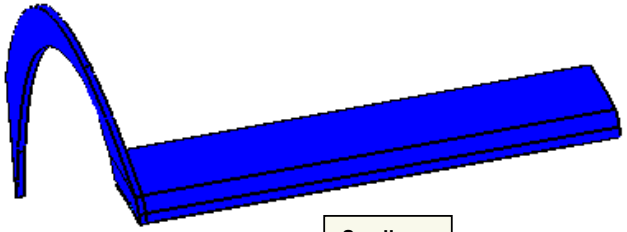
Radial Wedge – pressures due to rotation+ P_high



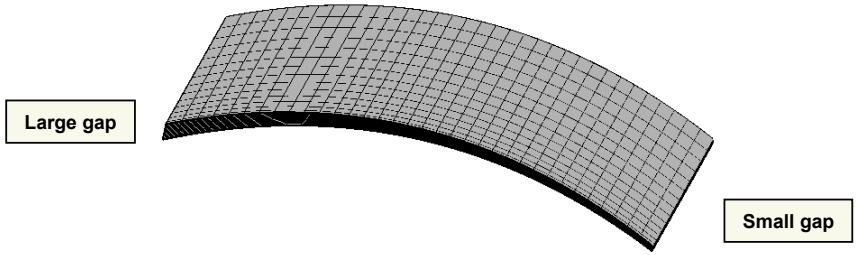
Radial Wedge – Cylindrical Geometry



Radial Wedge Under Pad

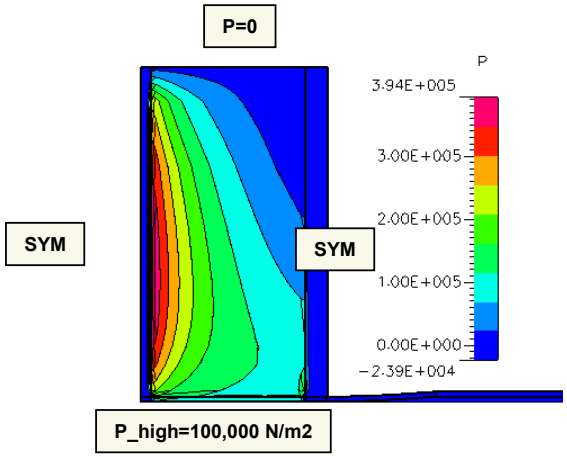


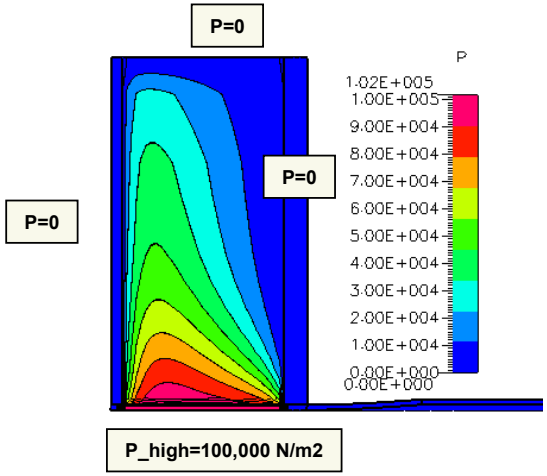
Geometry: pad is not moving

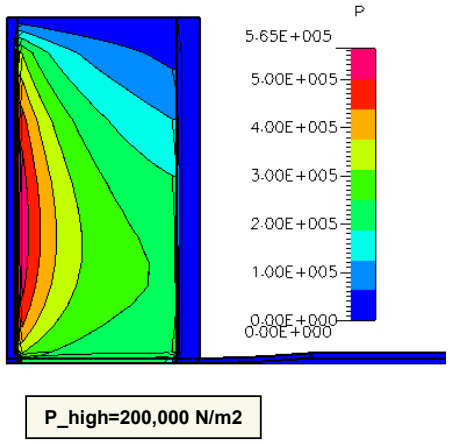


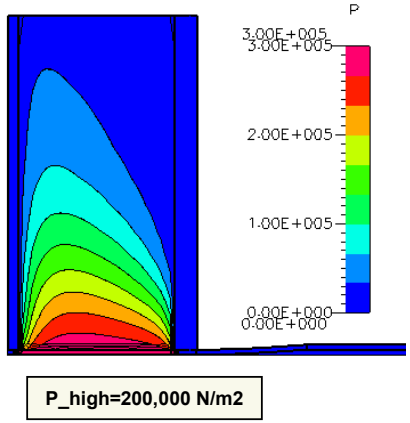
3d film zone-scaled

Radial Wedge – Results



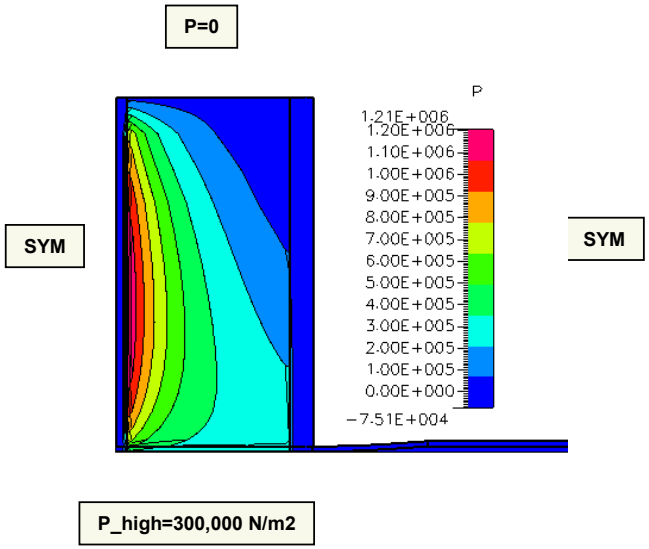


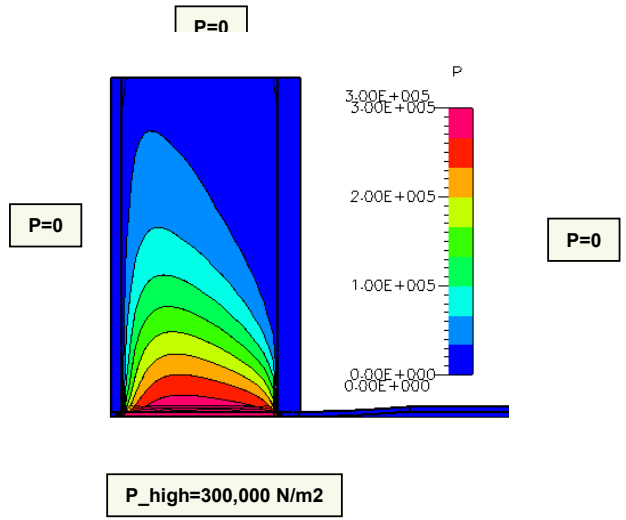




20,000 RPM

Radial Wedge – Results



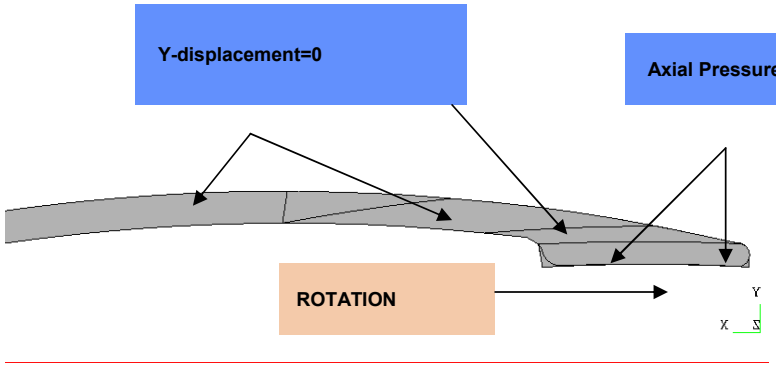


20,000 RPM

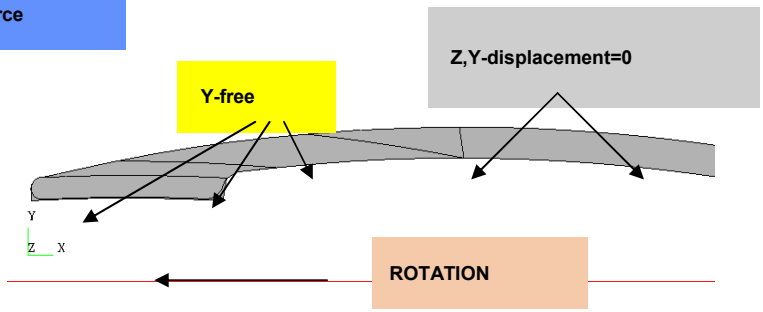
Moving Finger Simulation

Omega=2000 rad/sec (19108 RPM),
Phigh=8000 Pa (1 psi)
axial wedge (basic)
Pad L=0.25 inch, stick=15 mil
Film: 0.25 to 0.75 mils thick wedge

Boundary Conditions



Front view: high P side

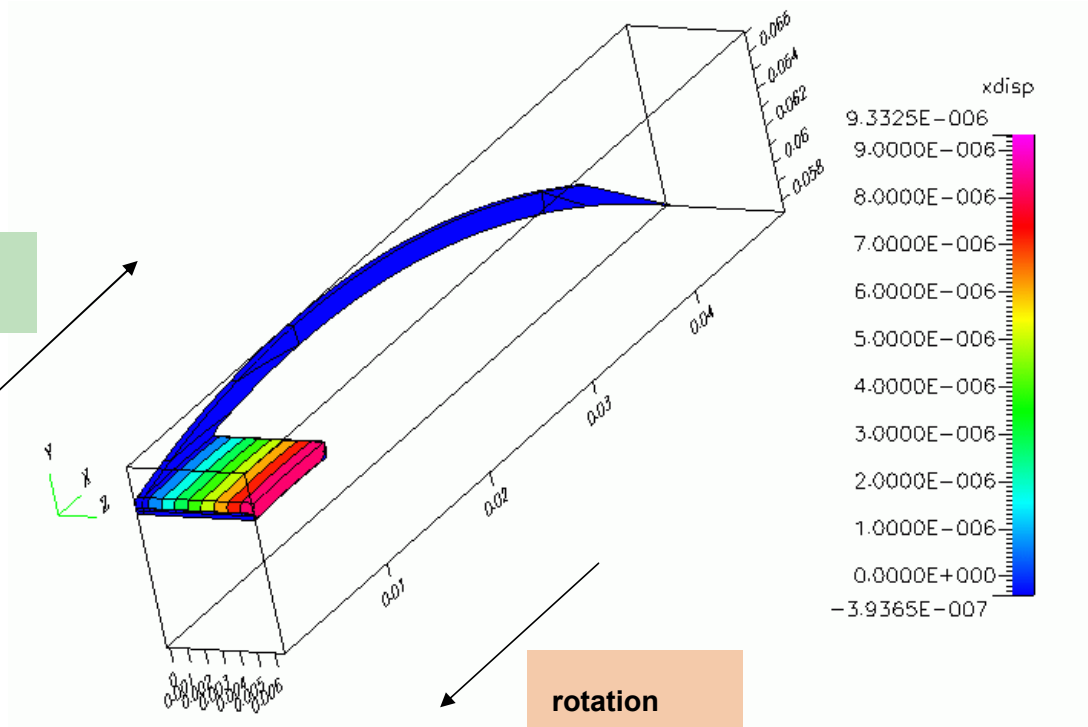


Rear view: low P side

Thus most of front surface is restricted for vertical displacement, except for 30 mils (radially,pad thickness) where axial pressure force of 1 psi is applied. On the rear end side we restrict both Y and Z(axial) displacement simulating backside support with strong friction. Support is not extended all the way and zone of 100 mils (radially) does not have any restrictions, free to deflect under axial forces from Phigh side.

X-Displacement (circumferential)

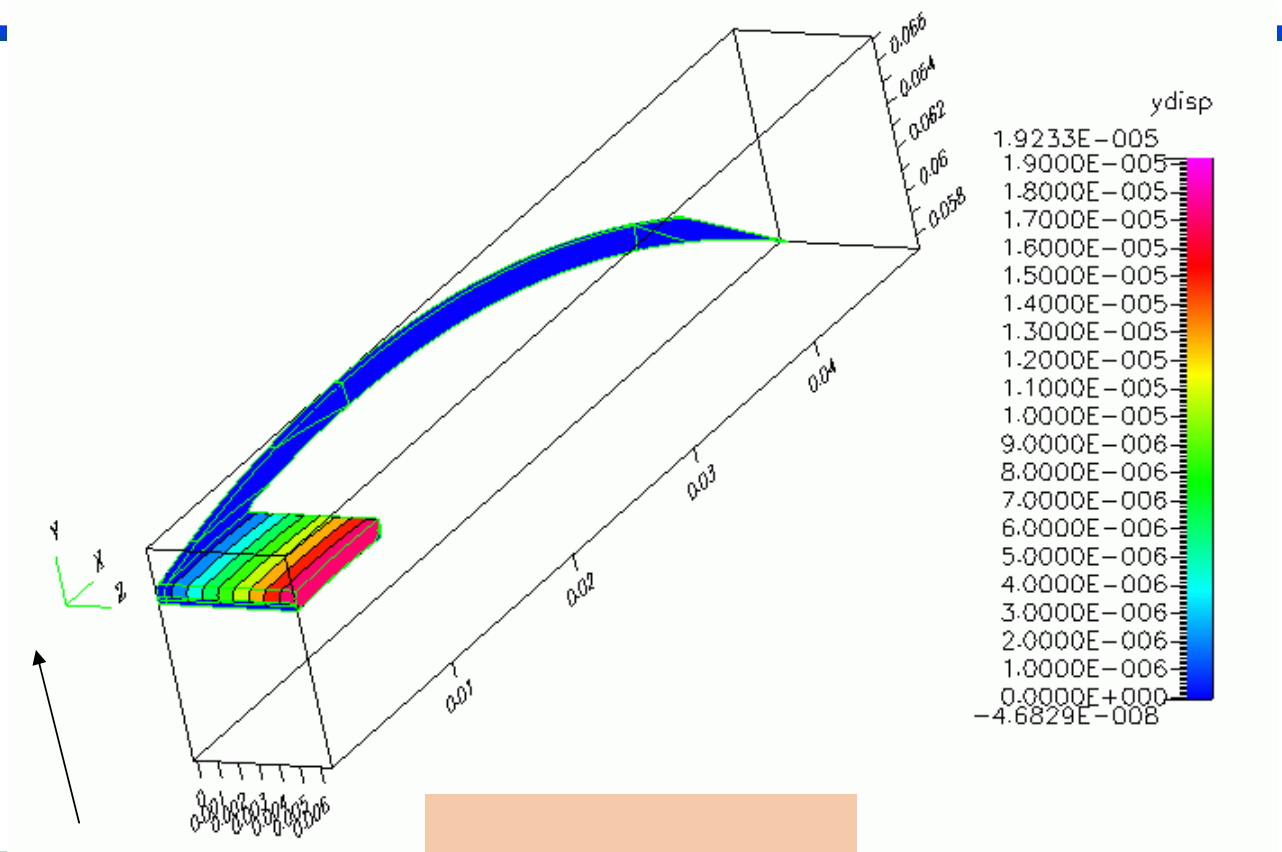
X –direction
circumferential



rotation

all displacements are in METERS
1mil=2.54E-5 m, so max displacement =0.33 mil

Y-Displacement (radial lift-off)

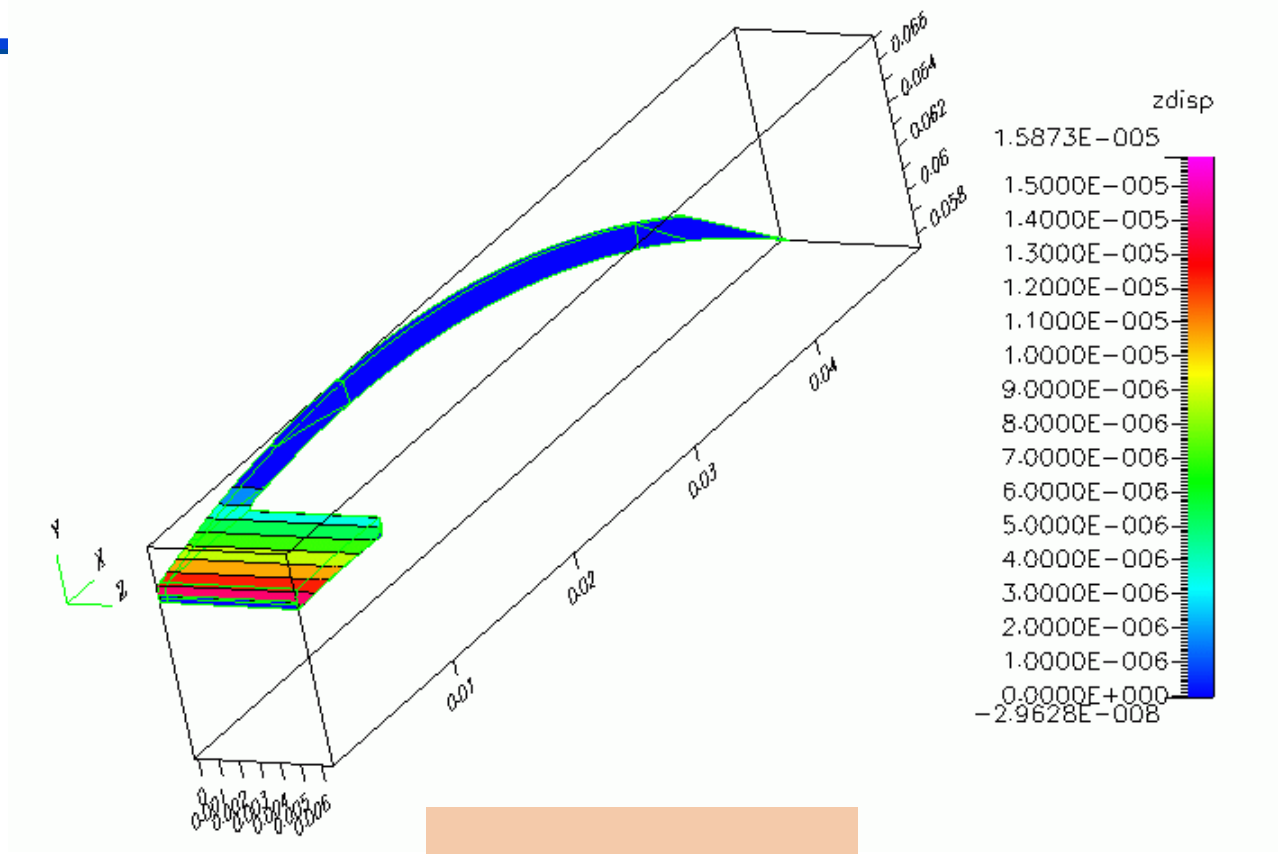


Y -direction

MOST radial DISPLACEMENT AT THE REAR TIP OF THE PAD (as expected) since stick is strongly restricted

all displacements are in METERS
1mil=2.54E-5 m, so max displacement =0.75 mil

Z-Axial Displacement



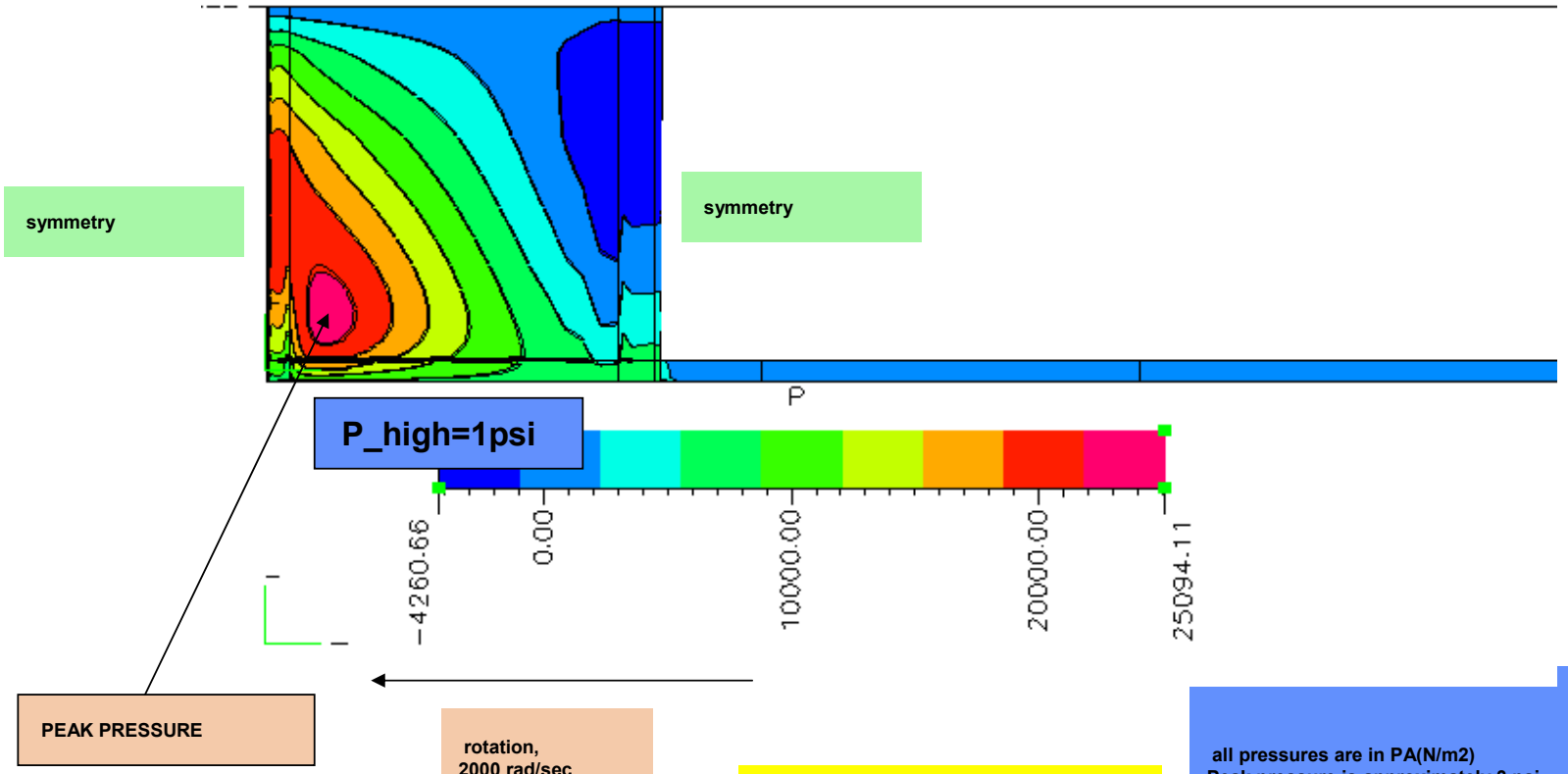
Z -direction

MOST DISPLACEMENT
AT THE front TIP
(largest momentum)

all displacements are in METERS
1mil=2.54E-5 m, so max displacement =0.6 mil

RADIAL PRESSURE FORCE (film)

Plow=0

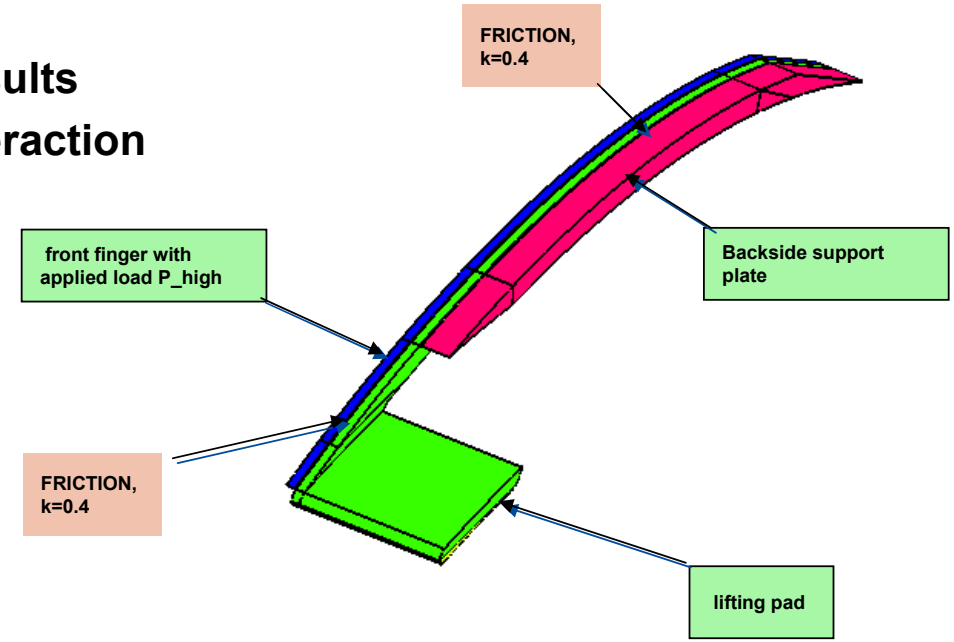


all pressures are in PA(N/m2)
Peak pressure is approximately 3 psi

NOTE THAT DUE TO INTERACTION OF STRONG ROTATION AND UPSTREAM AXIAL PRESSURE P_max in the film > P_high

Two Finger Geometry with Friction

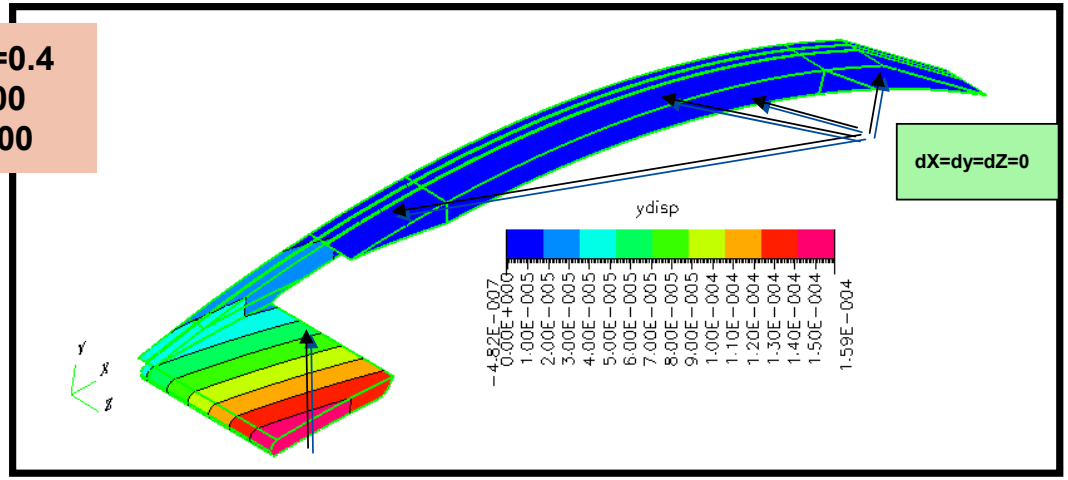
- Solid Modeling Results
- Fluid-Structure Interaction (FSI) Results

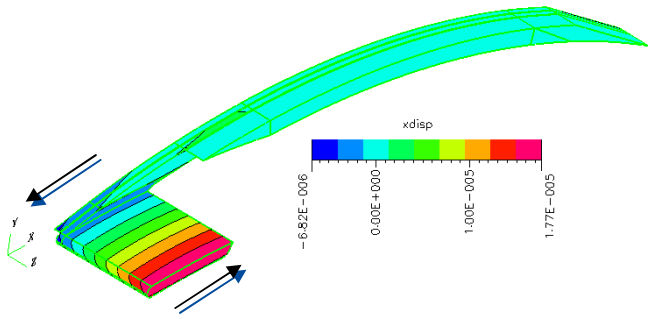


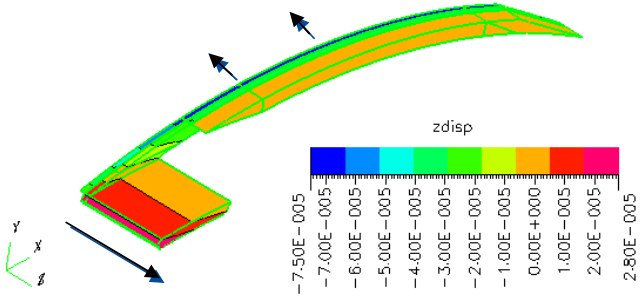
For solid models (stress only) we specified several load values, i.e. P_high=15,000 to 300,000 Pa and P_pad= 15,000 to 300,000 Pa.
For FSI analysis we calculated pressure distributions under the pad and accounted for finger/pad deformation under these forces.

Rear Washer restricted

FRICTION, k=0.4
Phigh=300,000
P_pad=300,000

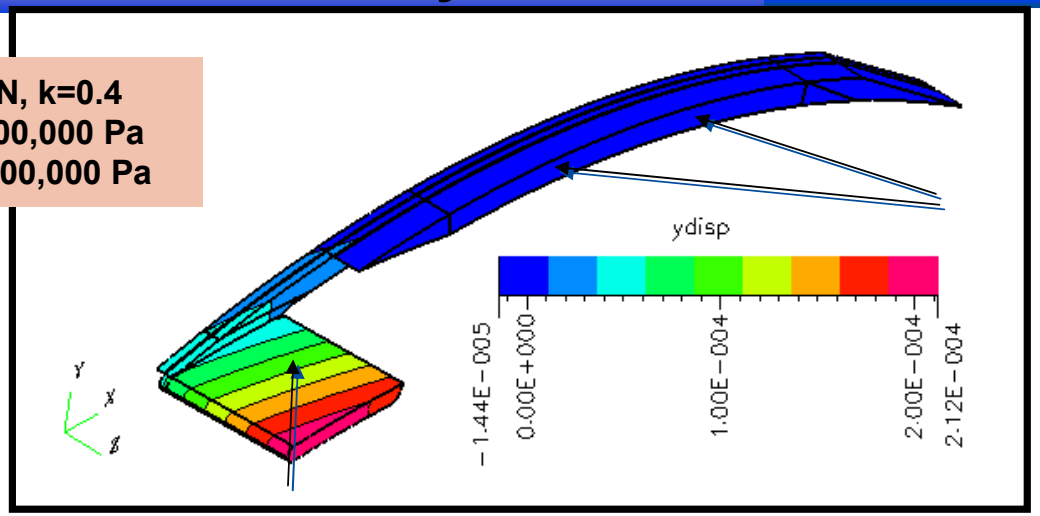




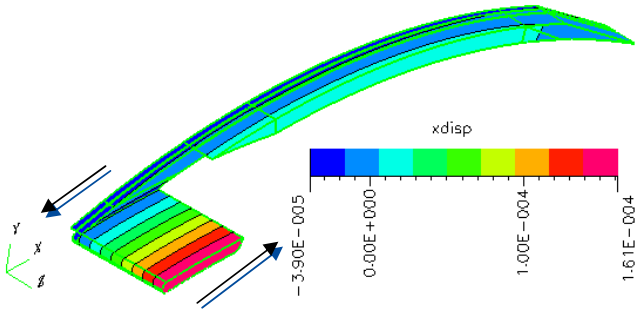


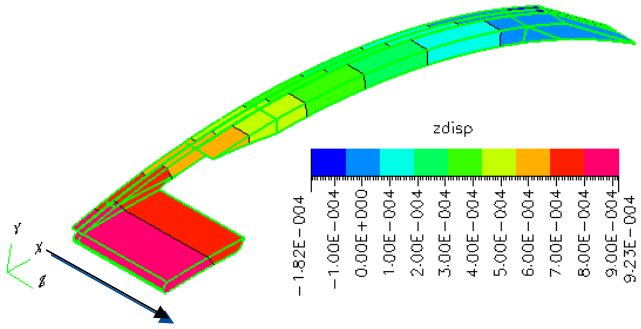
Rear Washer Partially restricted

FRICTION, k=0.4
Phigh=300,000 Pa
P_pad=300,000 Pa



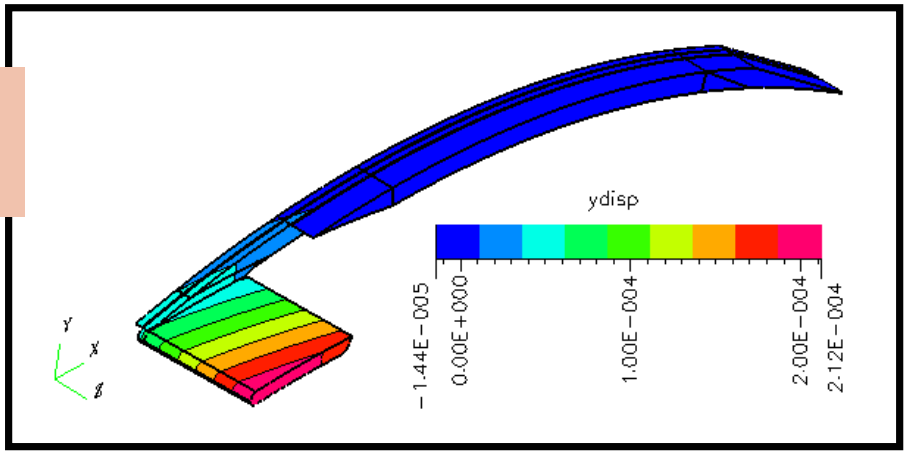
dY=0, only Y restriction, on rear surface only



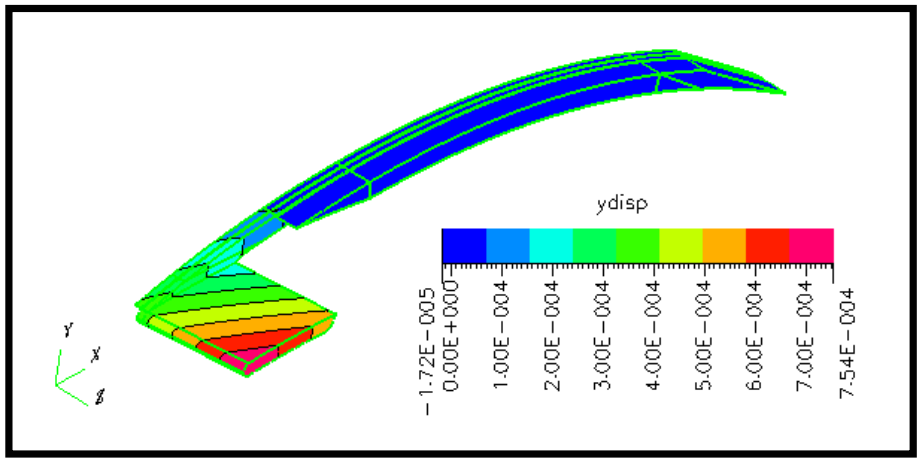


First vs Second FEA

FRICTION, k=0.4
P-high=300,000 Pa
P_pad=300,000 Pa



Radial Displacement -max 8.3 mil



Radial Displacement -max 29 mil