Identification of SFD force coefficients

Large Clearance Open Ends SFD

TRC-SFD-01-2012

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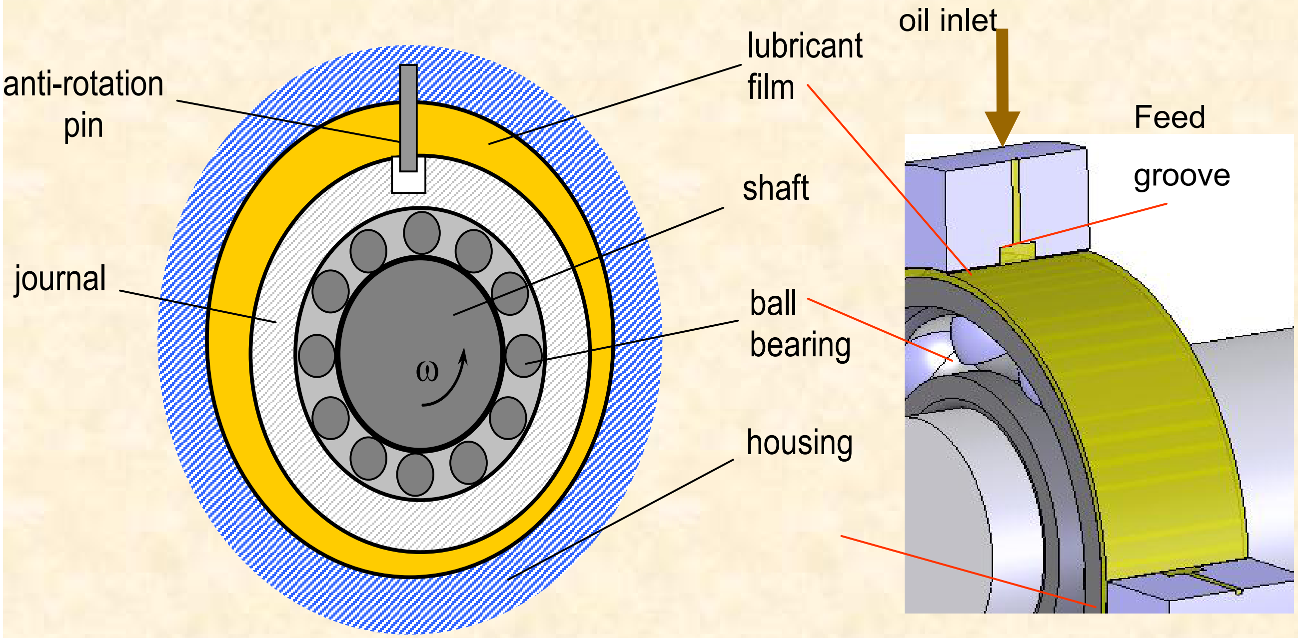
TRC Project 32513/1519FB

Linear Nonlinear Force Coefficients for SFDs

SFD with a central groove



Typical squeeze film damper (SFD) with a central groove

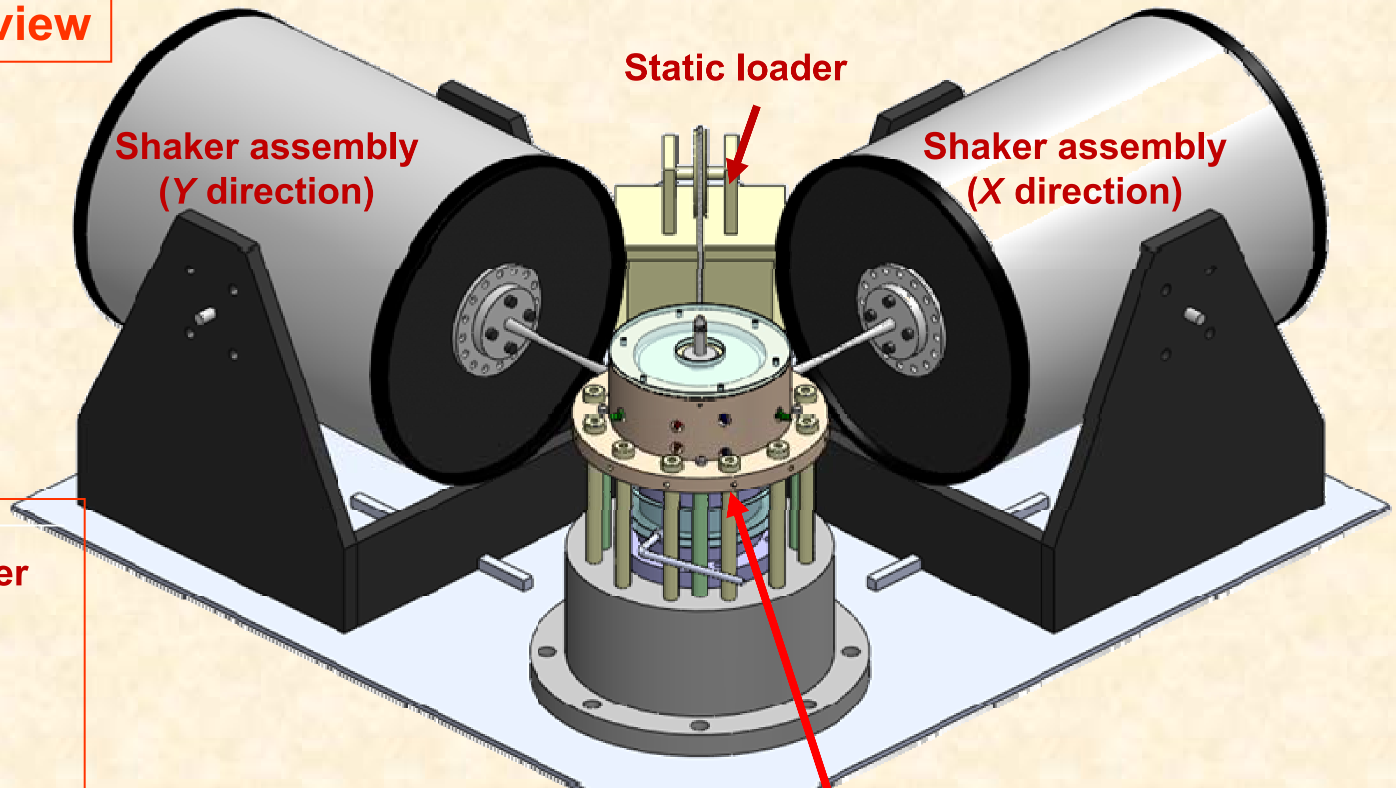
Conventional knowledge regards a groove is indifferent to the kinematics of journal motion, thus effectively isolating the adjacent film lands.

Pressurized lubricant flows through a central groove to fill the squeeze film lands.

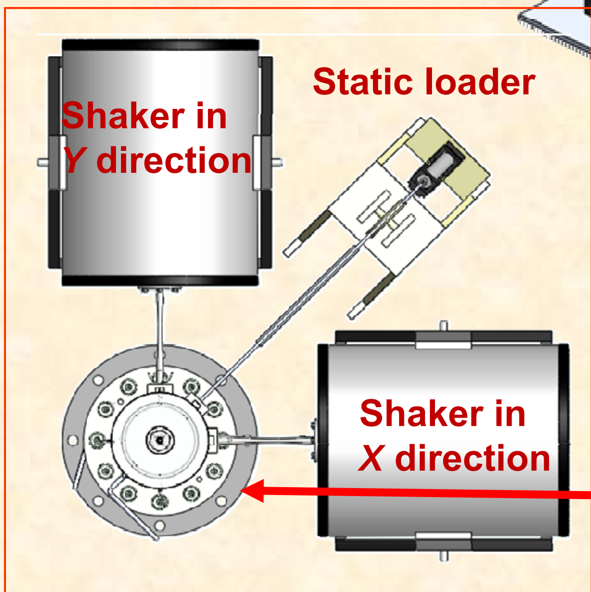
Dynamic pressures in the film lands generate reaction forces aiding to damp excessive amplitudes of rotor whirl motion.

P&W SFD test rig

Isometric view

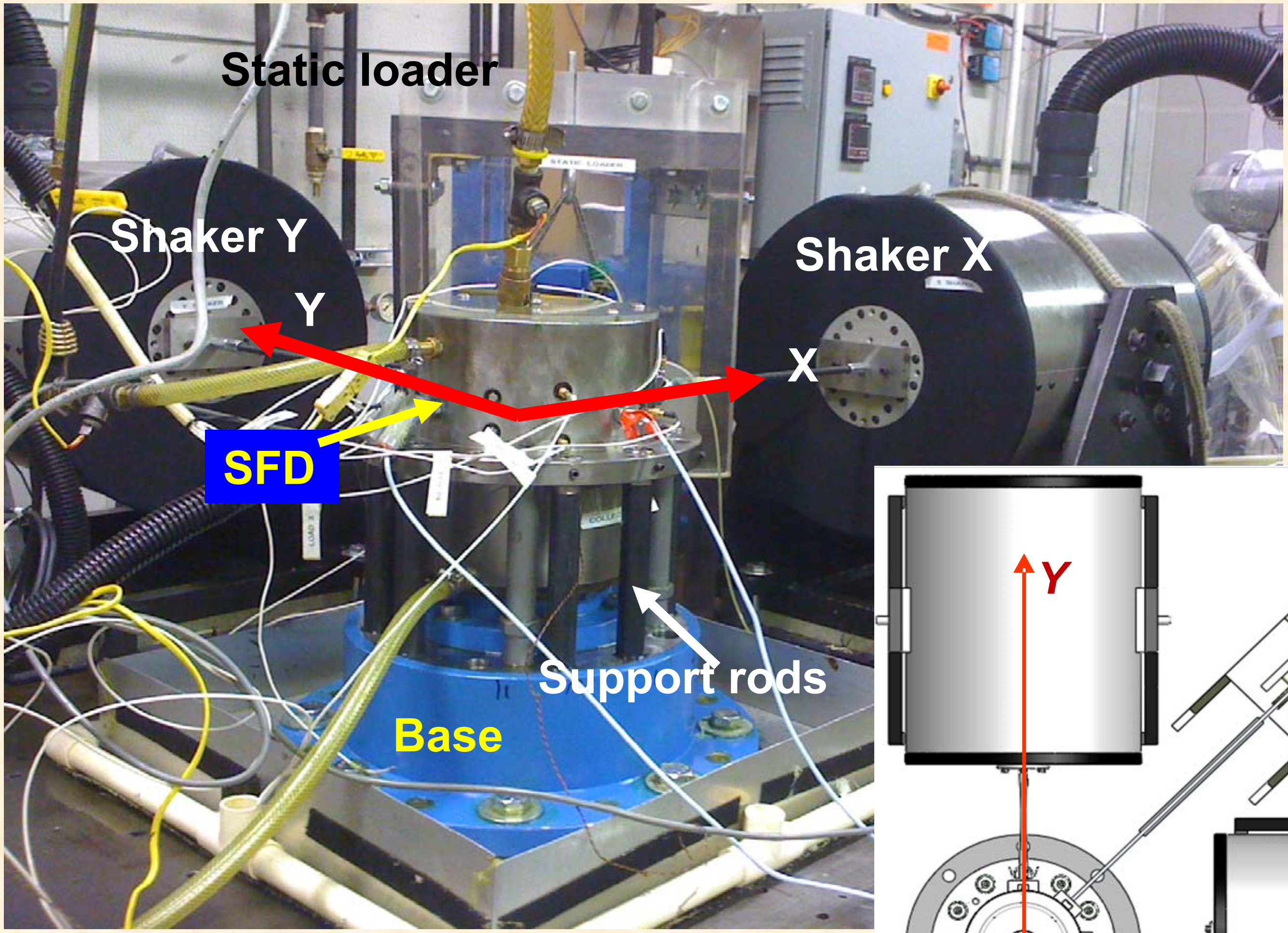


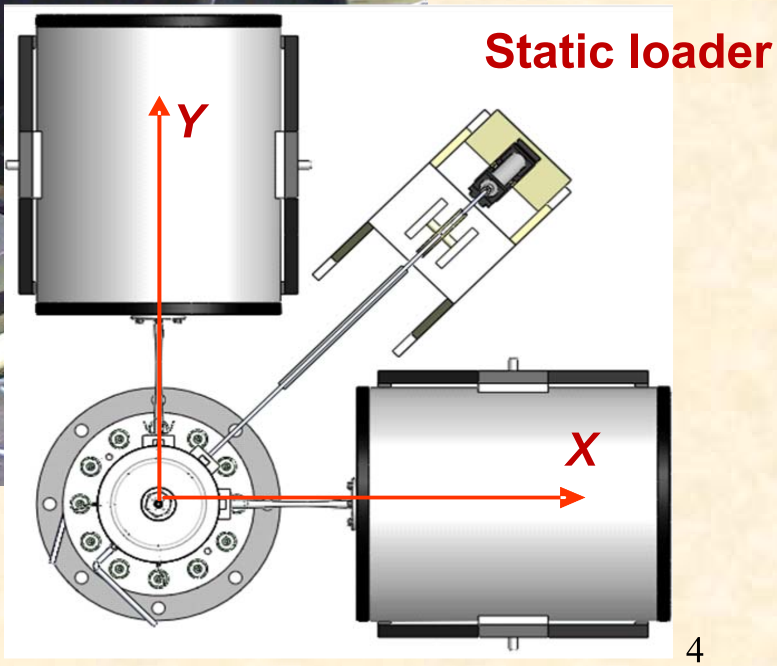
Top view



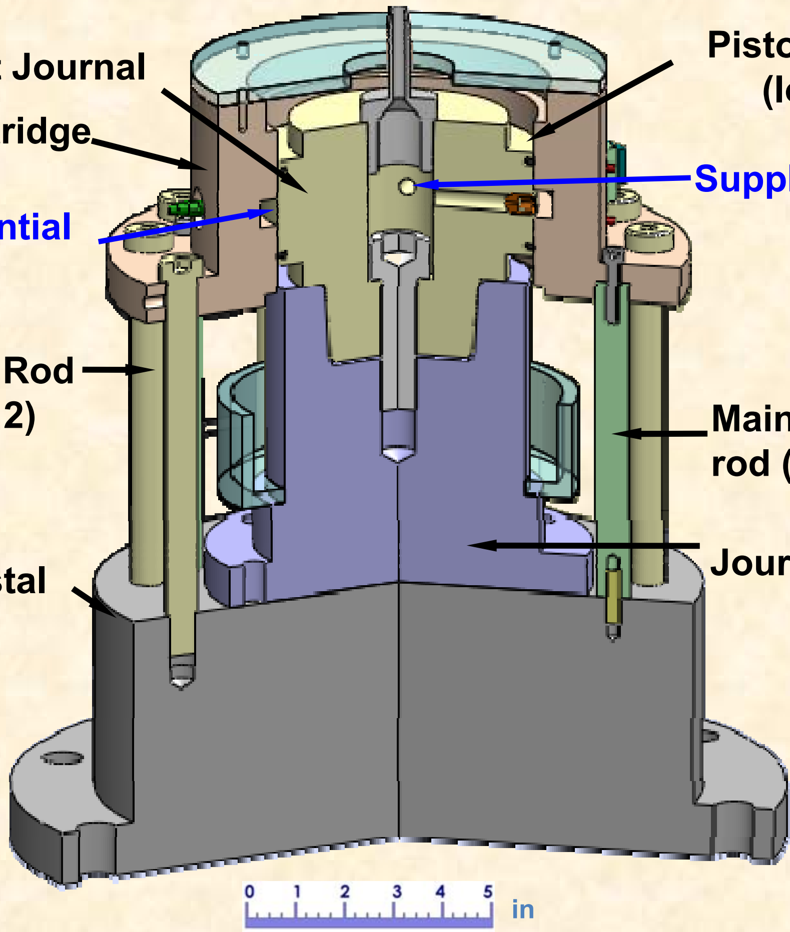
SFD test bearing

Test rig description

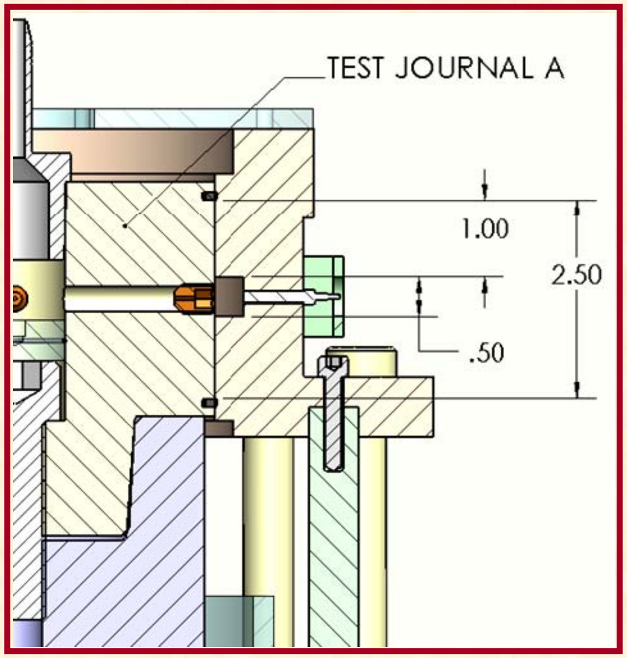




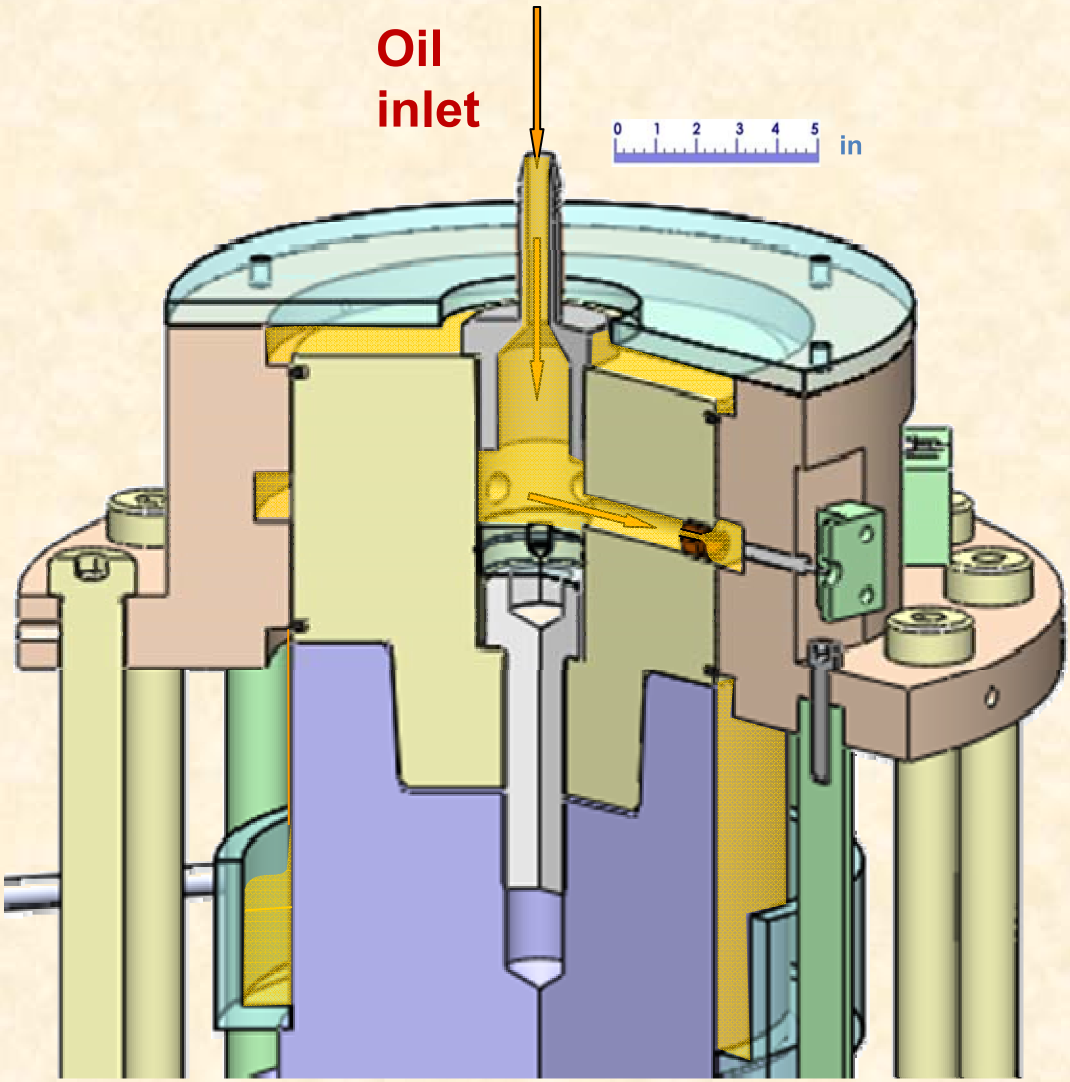
SFD Test Rig – cut section

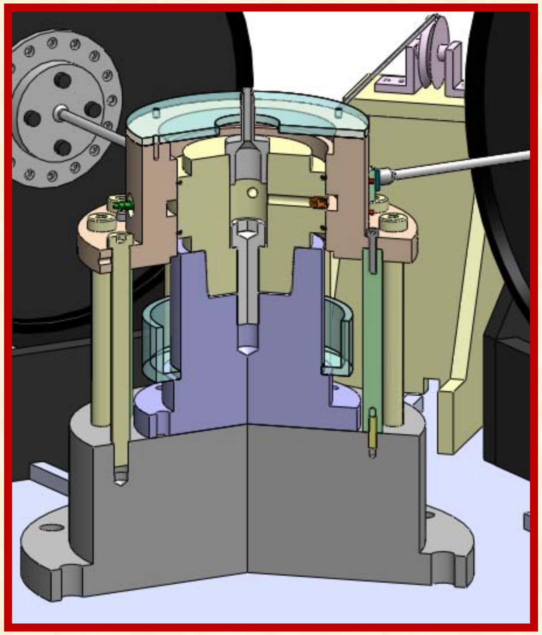


Test rig main features
Journal diameter: 5.0 inch
Film clearance: 9.9 mil
Film length: 2 x 1 inch
Support stiffness: 100 klbf/in



Lubricant flow path



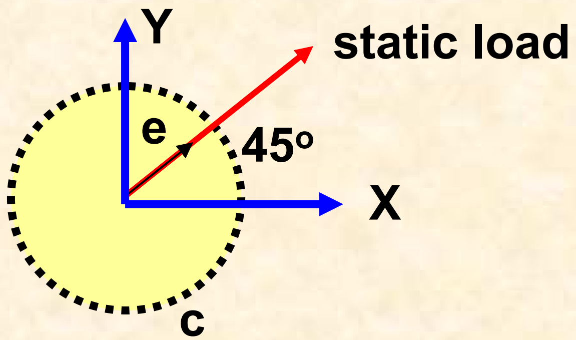


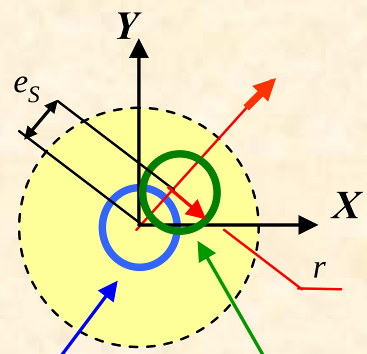
ISO VG 2 oil

Objective & tasks

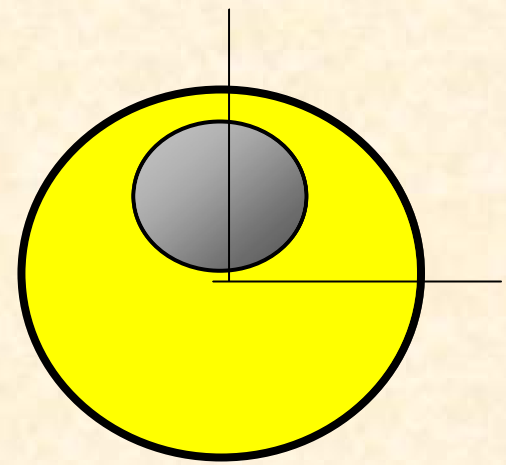
Evaluate dynamic load performance of SFD with a central groove.

Dynamic load measurements: **circular orbits (centered and off centered) and identification of test system and SFD force coefficients**



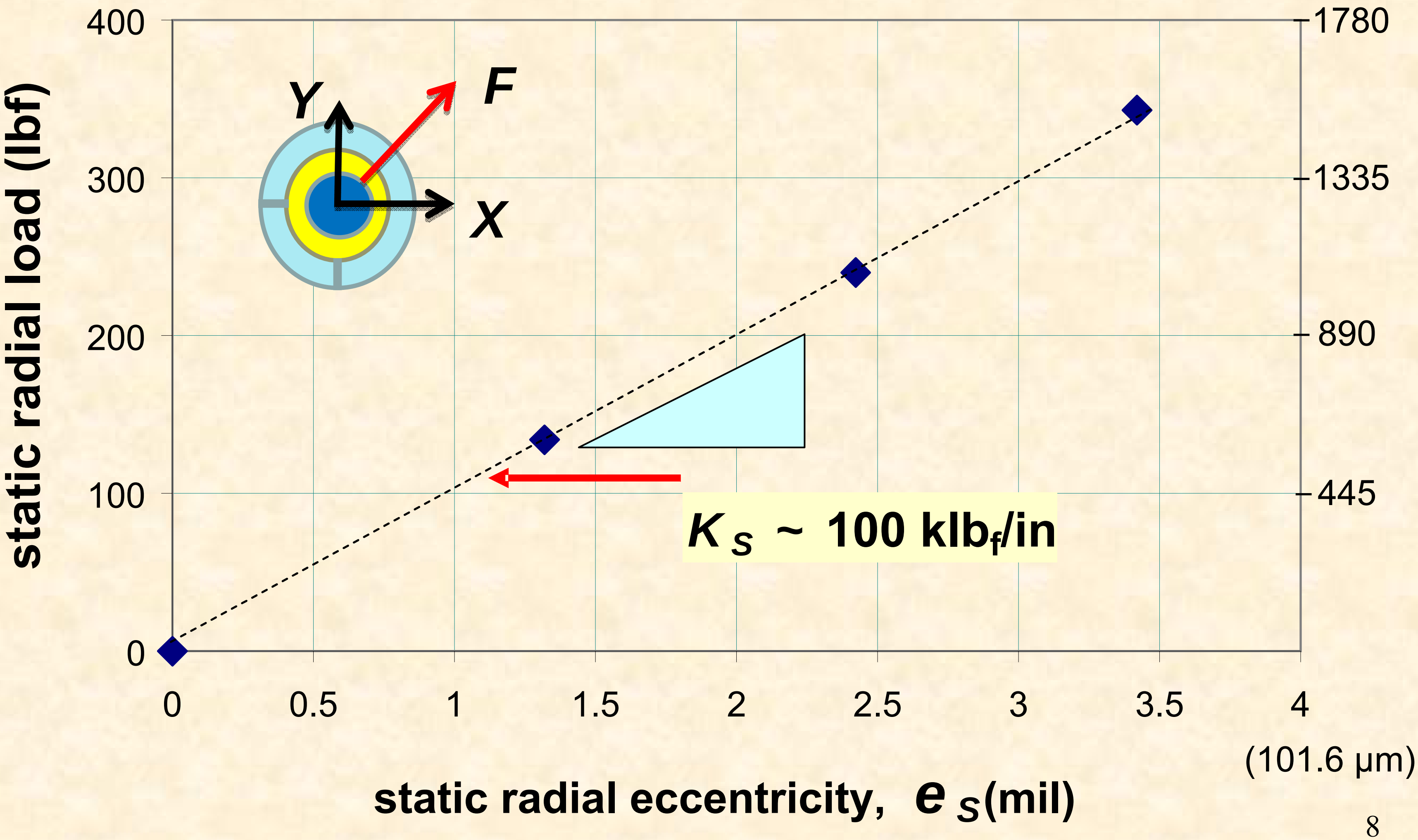


centered and off-centered circular orbits



Structure static stiffness

•Pull test using static loader to determine static structure stiffness



Structural parameters

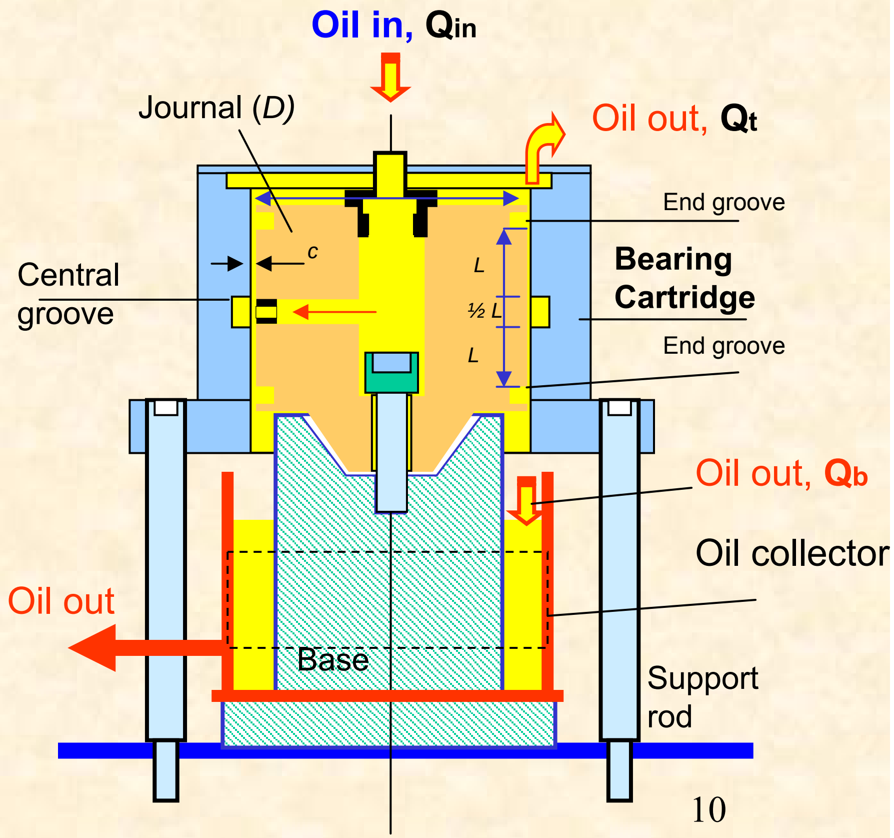
- Dry test system
- Circular Centered Orbits
- Frequency 50-210 Hz

		Direct			
		XX		*YY*	
		US	SI	US	SI
Stiffness	K_s	107 klbf/in	19 MN/m	120 klbf/in	21 MN/m
Damping	C_s	8 lbf-s/in	1.4 kN-s/m	9 lbf-s/in	1.6kN-s/m
Mass	M	-4 lb	-2 kg	-3 lb	-1 kg
System Mass	M_{BC}	48 lb	22 kg	48 lb	22 kg
Natural frequency	f_{ns}	148Hz		156Hz	
Damping ratio	ξ_s	4%		4%	

SFD dimensions & operating conds.

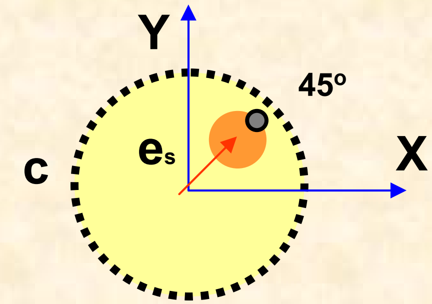
- **Maximum static load 324 lbf**
- **Centered and off-centered, e_S= 1, 2, and 3 mil**
- **Frequency range: 50-210 Hz, Orbit amplitude r = 0.5 mil**

ISO VG 2 Oil	
Viscosity at 73 °F [cPoise]	3.10
Density [kg/m³]	785
Inlet pressure [psig]	1.6
Outlet pressure [psig]	0
Radial Clearance [mil]	9.9
Journal Diameter [inch]	5.0
Central groove length [inch] & depth	0.500 0.375
Land length, *L* [inch]	1.0 x 2
Total Length [inch]	2.5



SFD force coefficients

IVFM parameter identification method



SFD Difference between lubricated system and dry system (baseline) coefficients

$$C_{SFD}=C_{lubricated} - C_s$$

$$M_{SFD}=M_{lubricated} - M_{BC}$$

$$K_{SFD}=K_{lubricated} - K_s$$

DRY system parameters

K_s = 100 klbf/in
M_{BC} = 48 lb
C_s= 8-9 lbf-s/in
Nat freq = 148-156 Hz
Damping ratio = 4%

SFD force coefficients - theory

Centered journal (e_s=0), no lubricant cavitation
Two film lands separated by a plenum: central groove has no effect on squeeze film forces.

Damping ➡

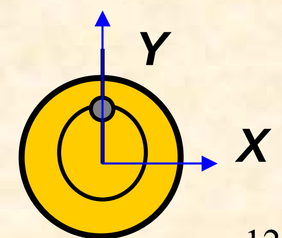
$$C^* = C^*_{XX} = C^*_{YY} = 2 \times 12 \pi \mu L \left(\frac{R}{c} \right)^3 \left[1 - \frac{\tanh\left[L/D \right]}{L/D} \right]$$

Inertia ➡

$$M^* = M^*_{XX} = M^*_{YY} = 2 \times \frac{\pi \rho L R^3}{c} \left[1 - \frac{\tanh\left[L/D \right]}{L/D} \right]$$

Stiffness ➡

$$K_{XX} = K_{YY} = K_{XY} = K_{YX} = 0$$



SFD force coefficients - theory

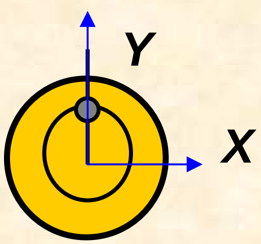
Damping

$$C^* = C^*_{XX} = C^*_{YY} = 2\times 12\pi\,\mu L\left(\frac{R}{c}\right)^3\left[1-\frac{\tanh\left[{L}/{D}\right]}{{L}/{D}}\right]$$

c=5.5 mil	C_* = 7,121 N.s/m (40.7 lb_f.s/in)
c=9.9 mil	C_* = 1,255 N.s/m (7.16 lb_f.s/in)

Inertia

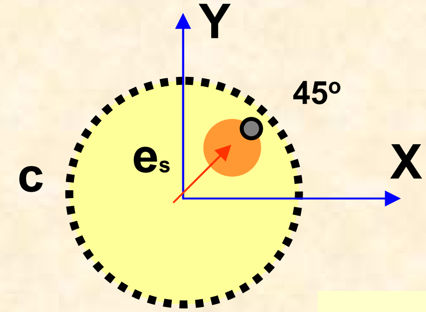
$$M^* = M^*_{XX} = M^*_{YY} = 2\times\frac{\pi\,\rho L R^3}{c}\left[1-\frac{\tanh\left[{L}/{D}\right]}{{L}/{D}}\right]$$

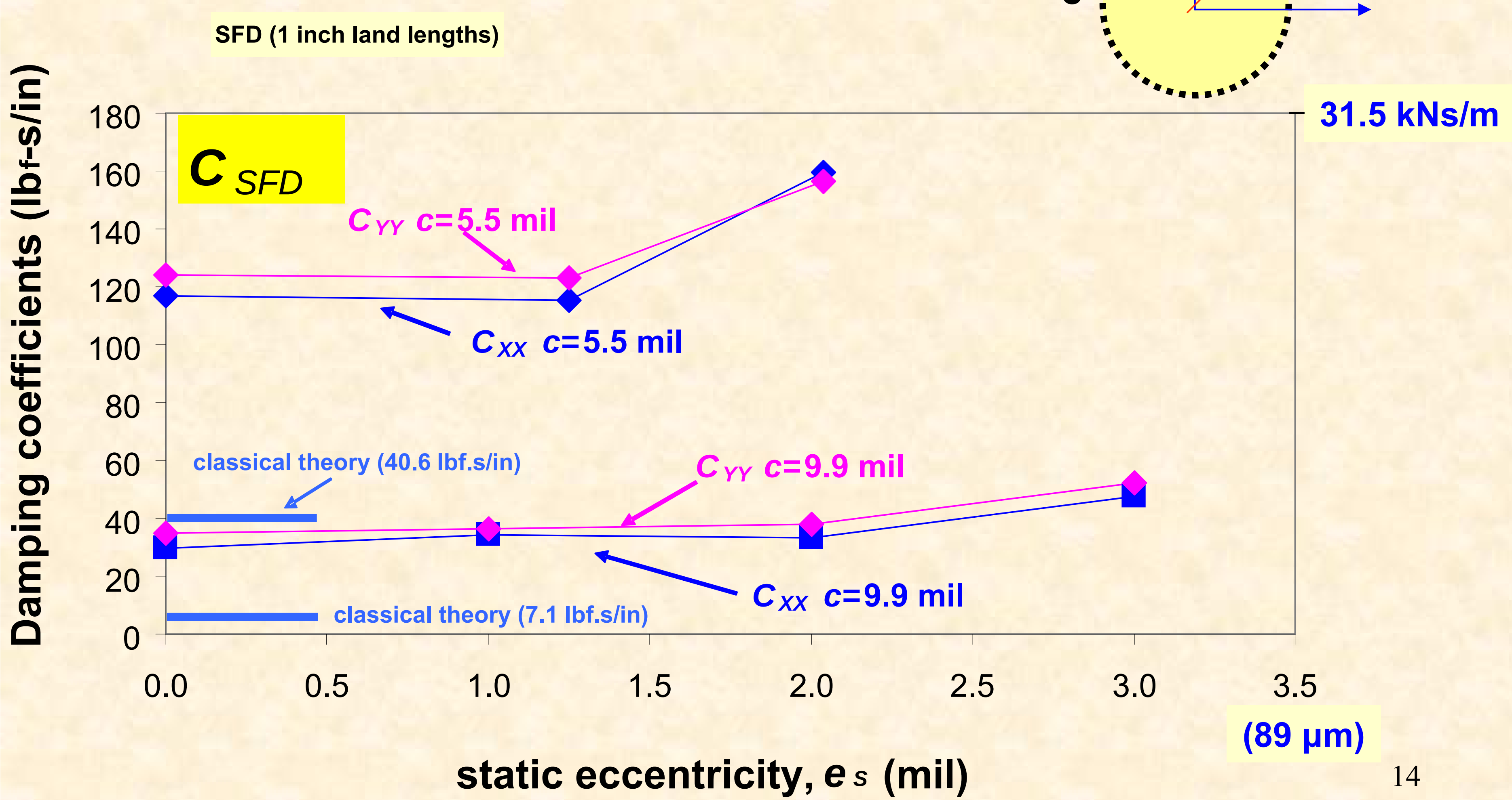


c=5.5 mil	M_* = 2.98 kg (6.58 lb_m)
c=9.9 mil	M_* = 1.67 kg (3.69 lb_m)

Experimental SFD damping coeffs.

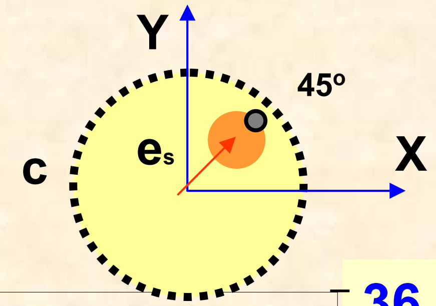
- Open ends SFD
- Circular orbits (r = 0.5 mil)

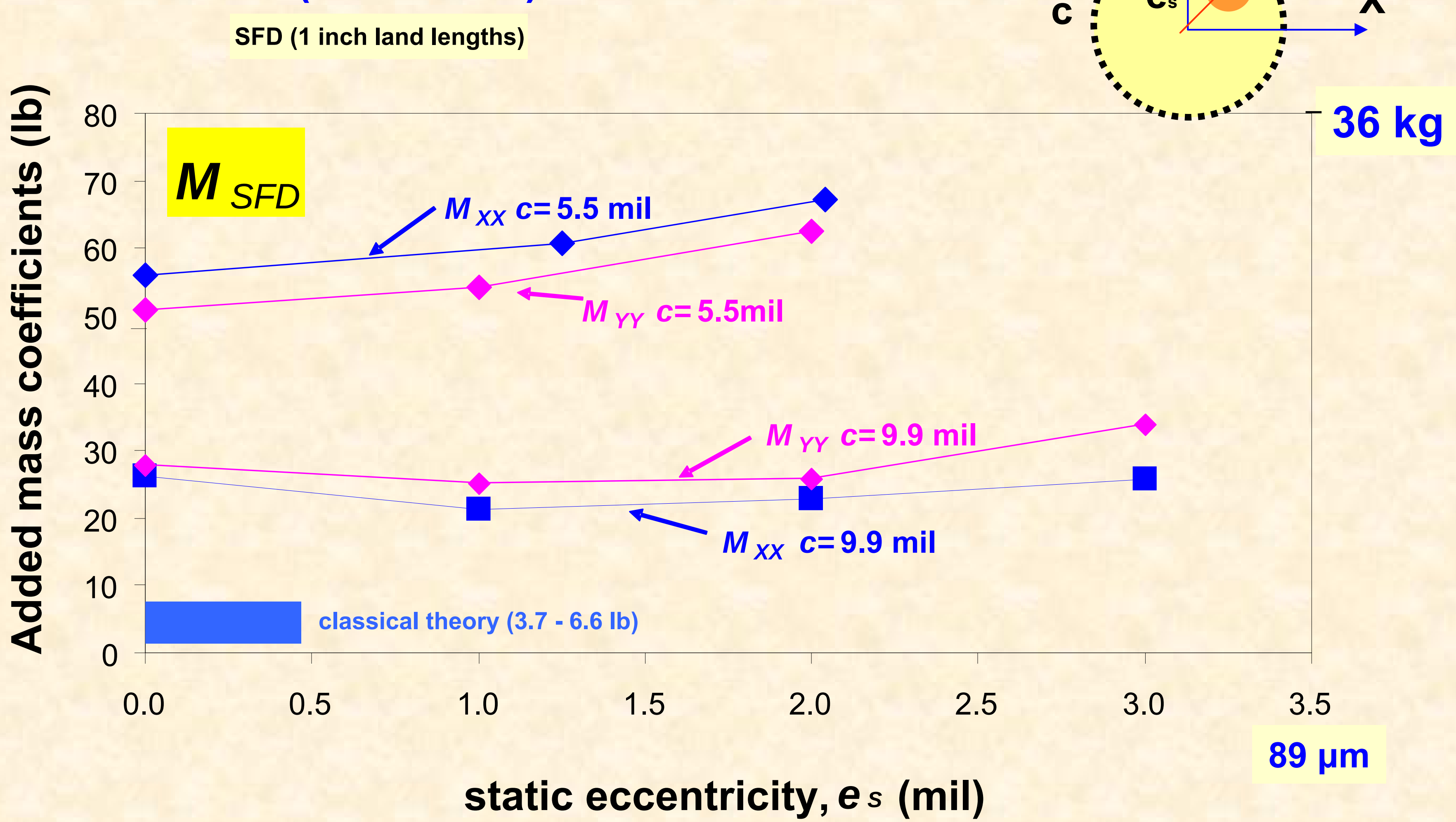




Experimental SFD inertia coeffis.

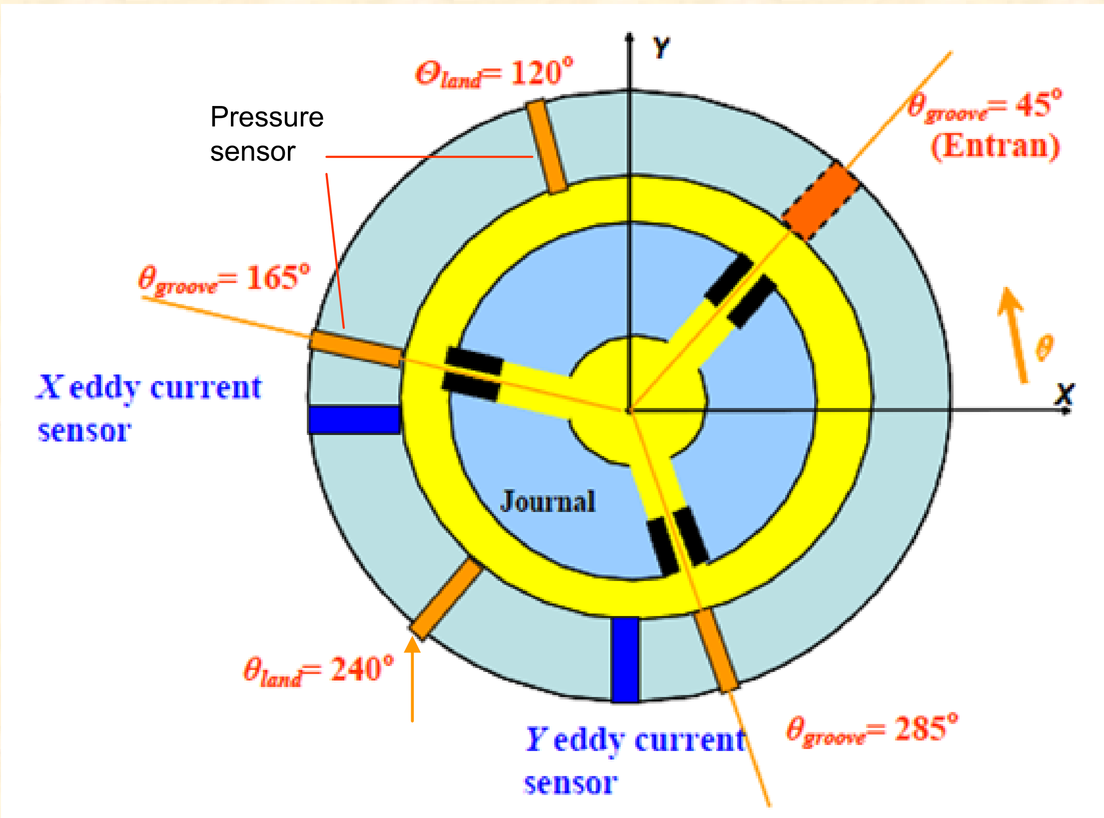
- Open ends SFDs
- Circular orbits **(r = 0.5 mil)**



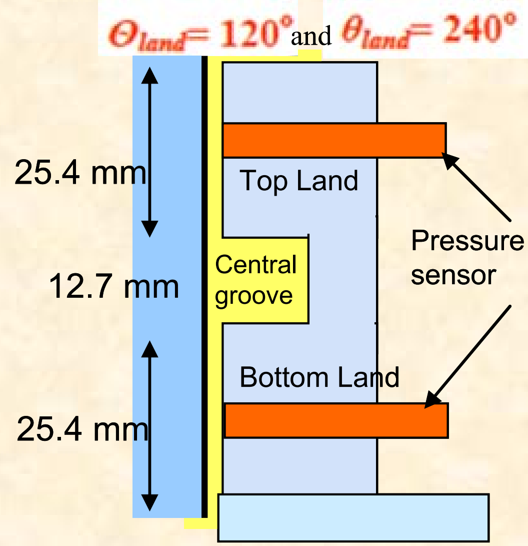


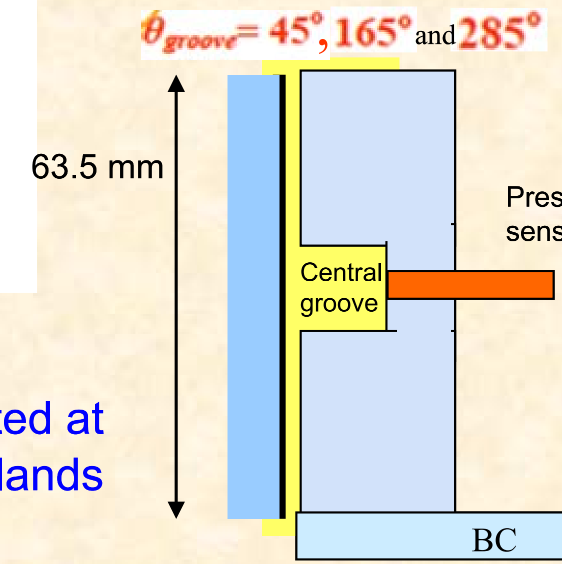
Pressure sensors in bearing

Top view: Sensors around bearing circumference



Pressure sensor locations





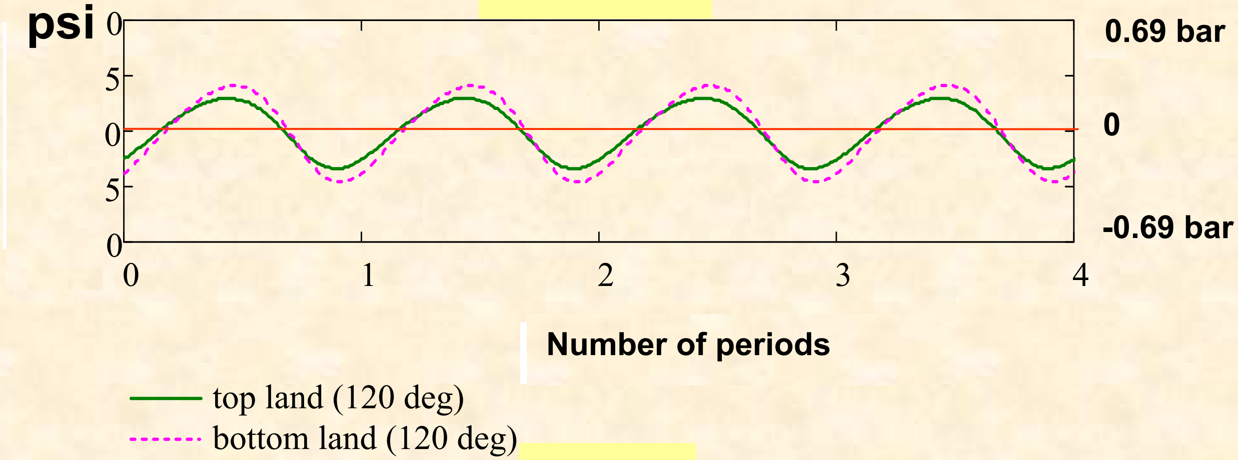
Side view: Sensors located at middle plane of film lands

Dynamic pressures: films & groove

Whirl frequency 130 Hz

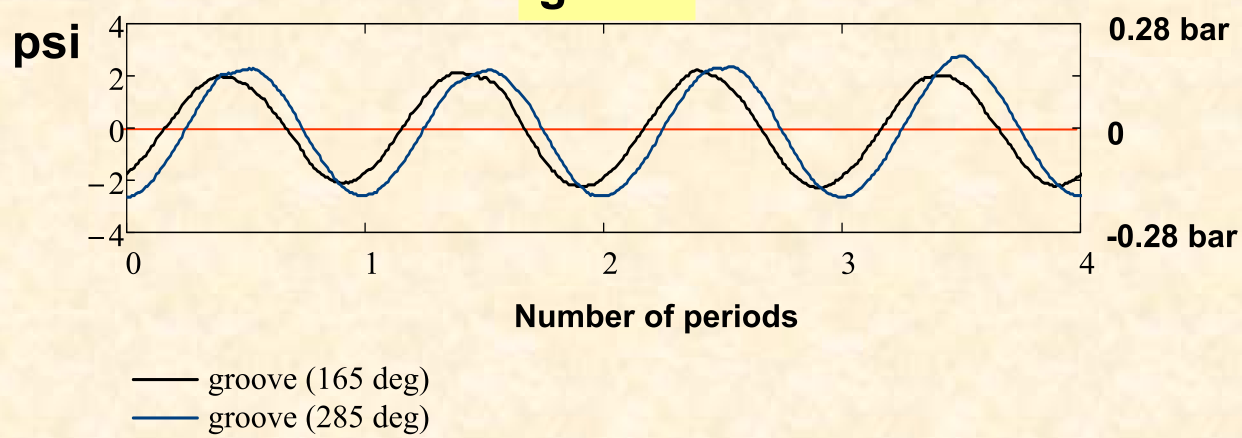
ASME GT2012-68212

film lands



Top and bottom film lands show similar pressures.

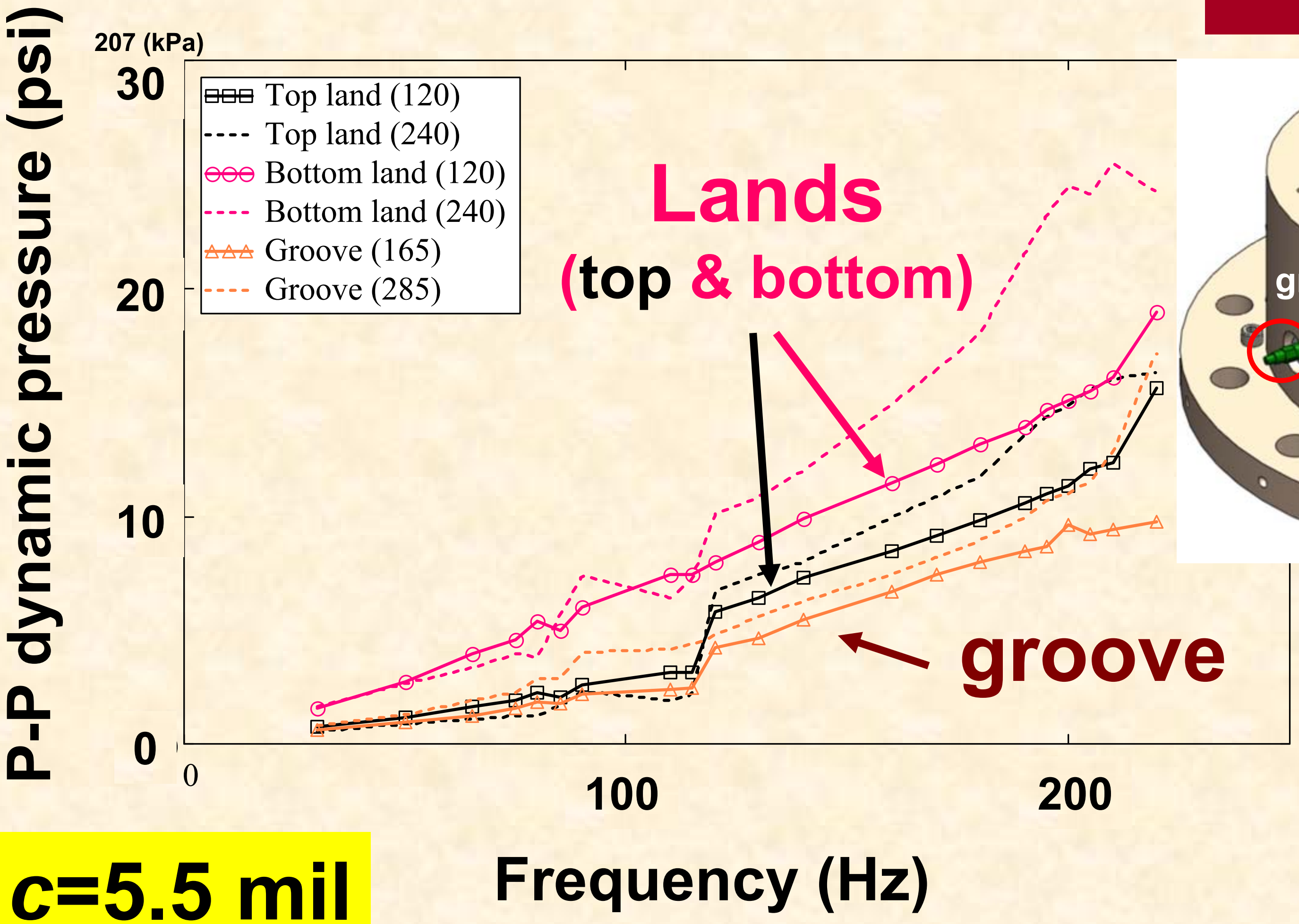
groove



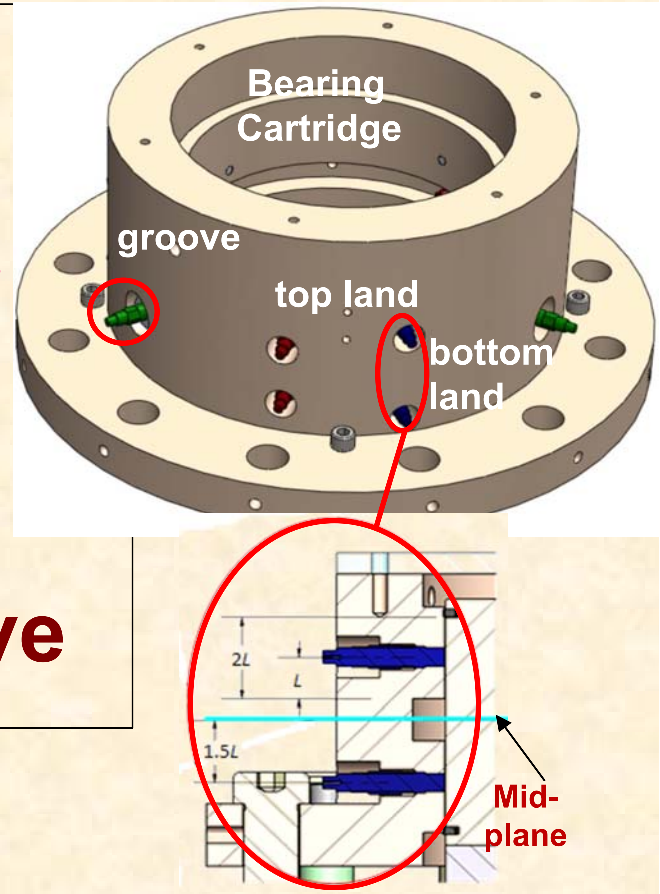
Dynamic pressure in the groove is not zero!

e_s=0, circular orbit r=0.5 mil. Groove pressure P_G = 0.72 bar

Peak-peak lubricant pressures



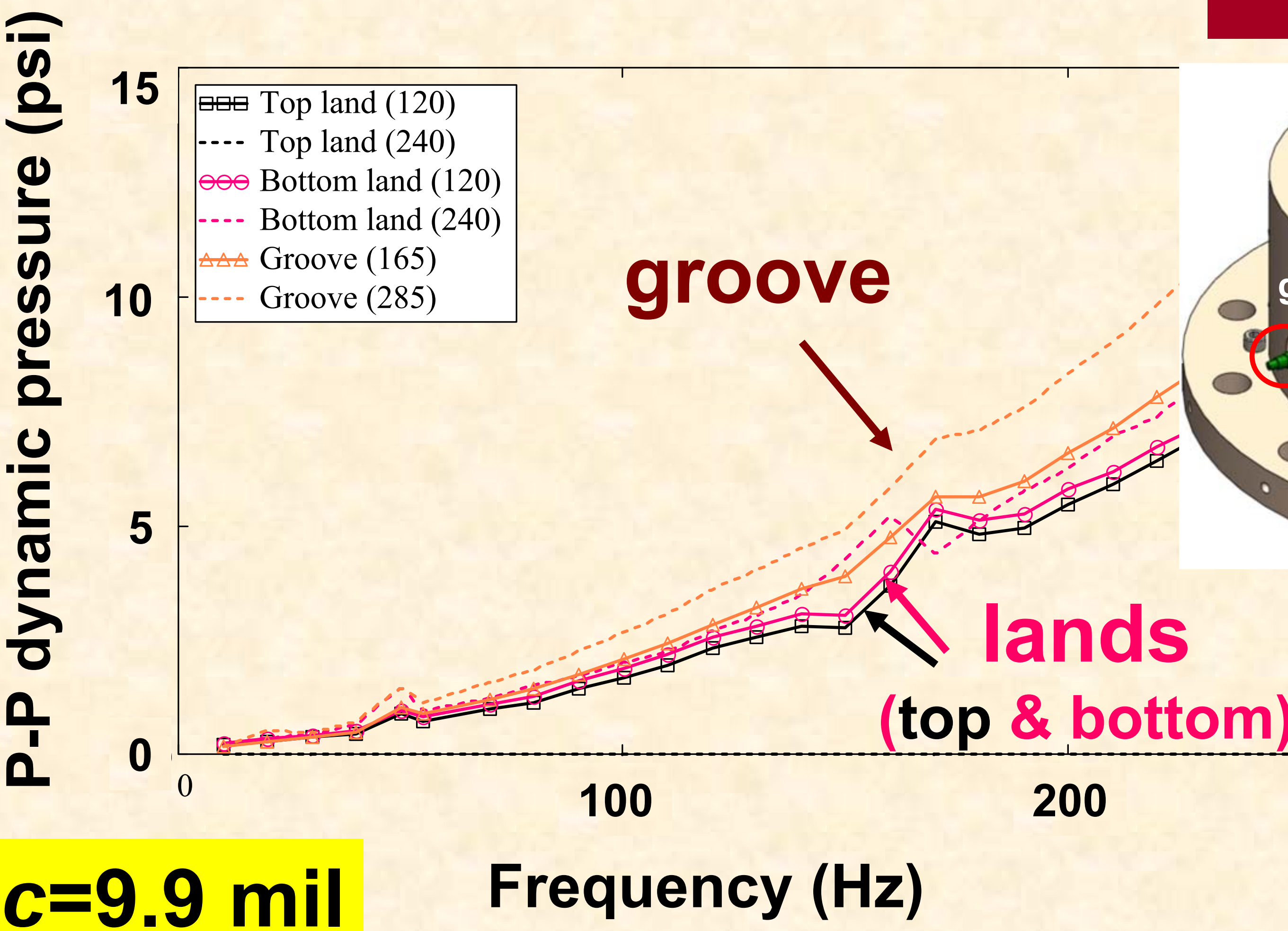
Piezoelectric pressure sensors (PCB) location

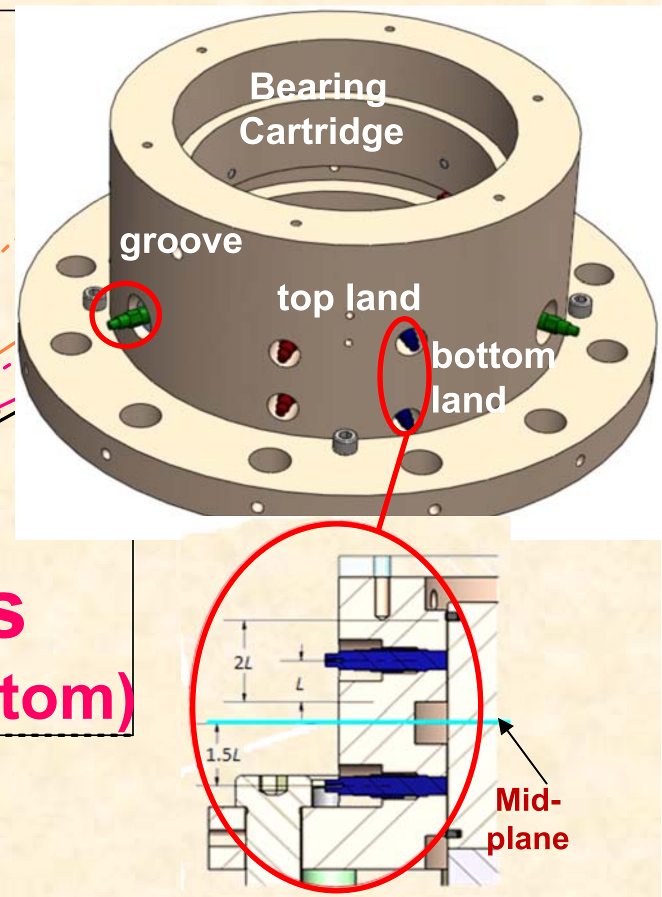


c=5.5 mil

Peak-peak lubricant pressures

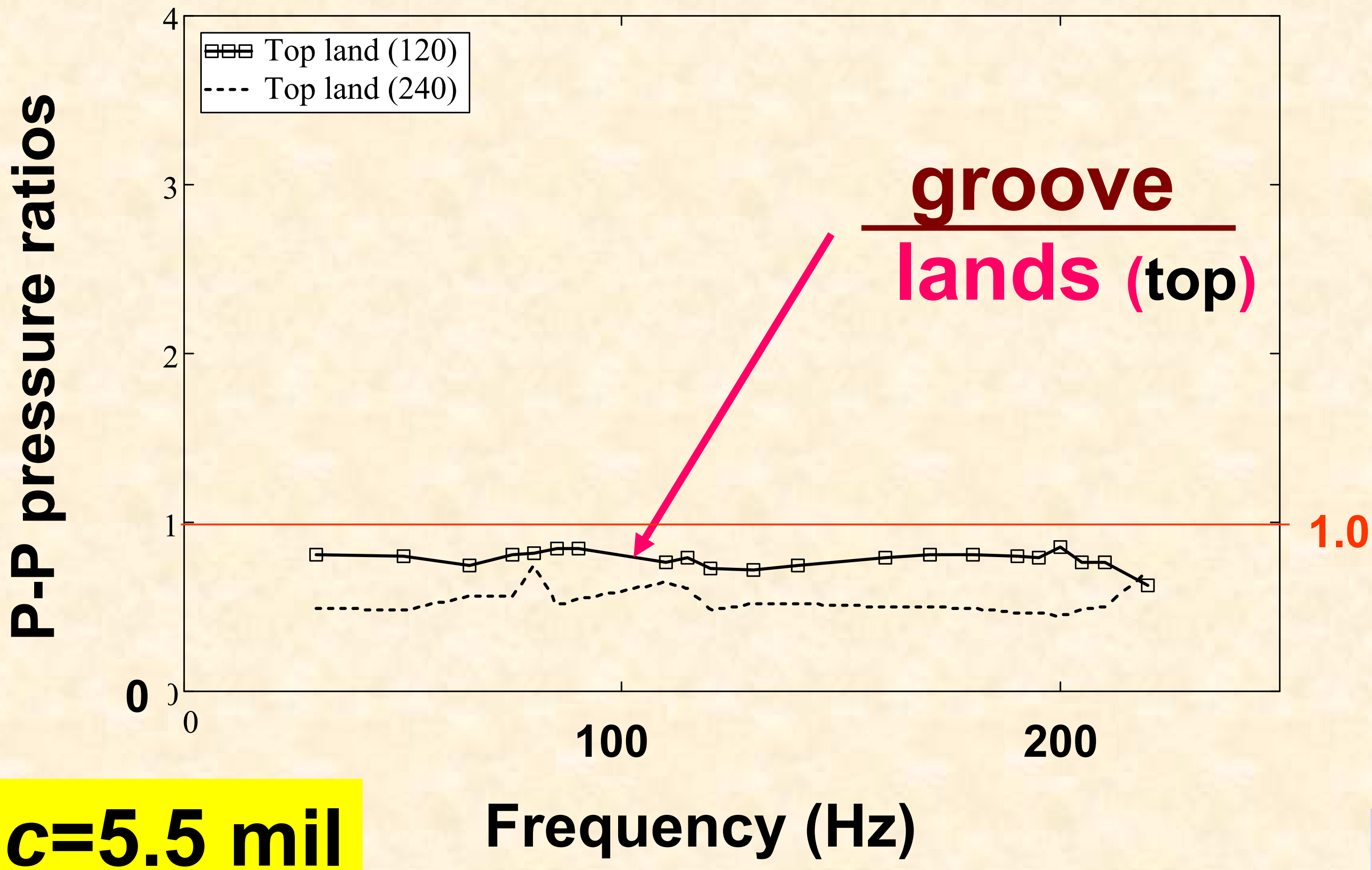
Piezoelectric pressure sensors (PCB) location





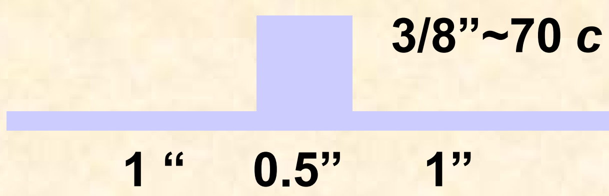
***c*=9.9 mil**

Ratio of groove/film land pressures

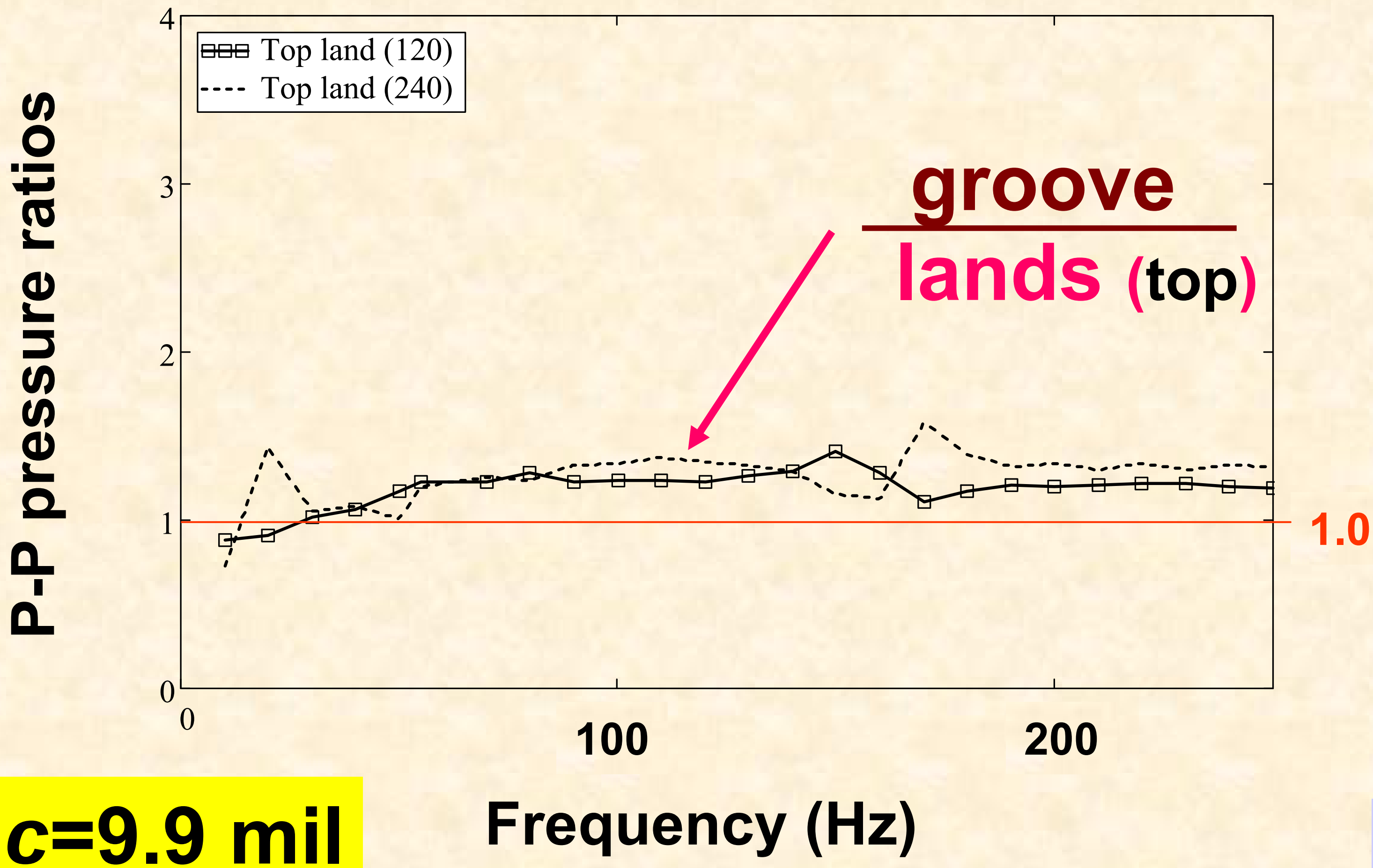


Groove generates large hydrodynamic pressures!

c=5.5 mil

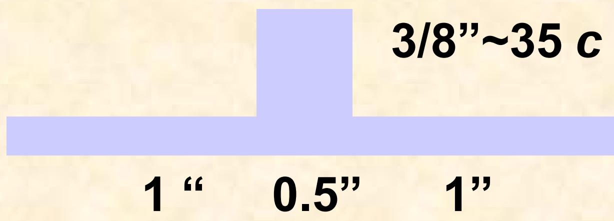


Ratio of groove/film land pressures



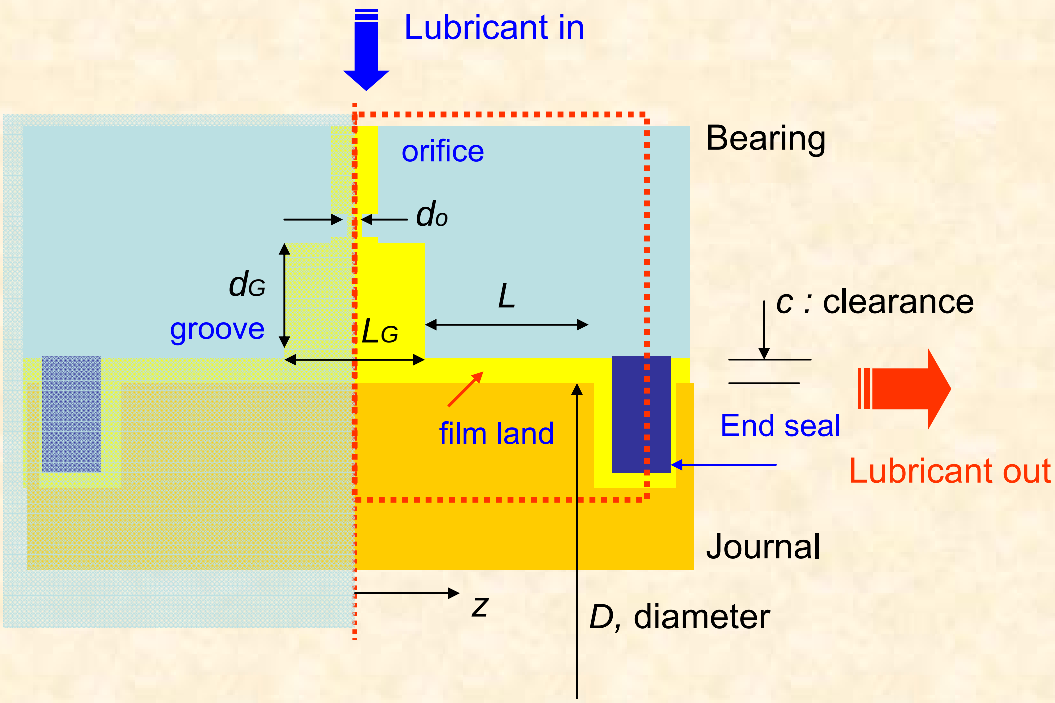
Groove generates larger hydrodynamic pressures!! Larger than in the film!

c=9.9 mil

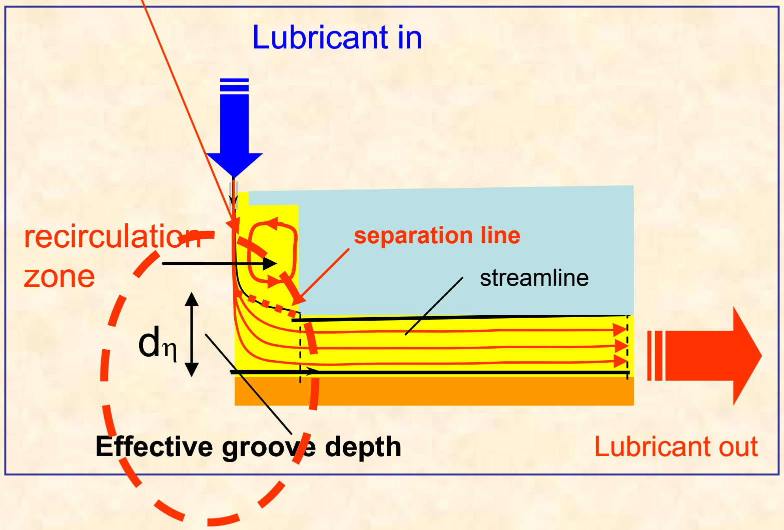


Model SFD with a central groove

SFD geometry and nomenclature



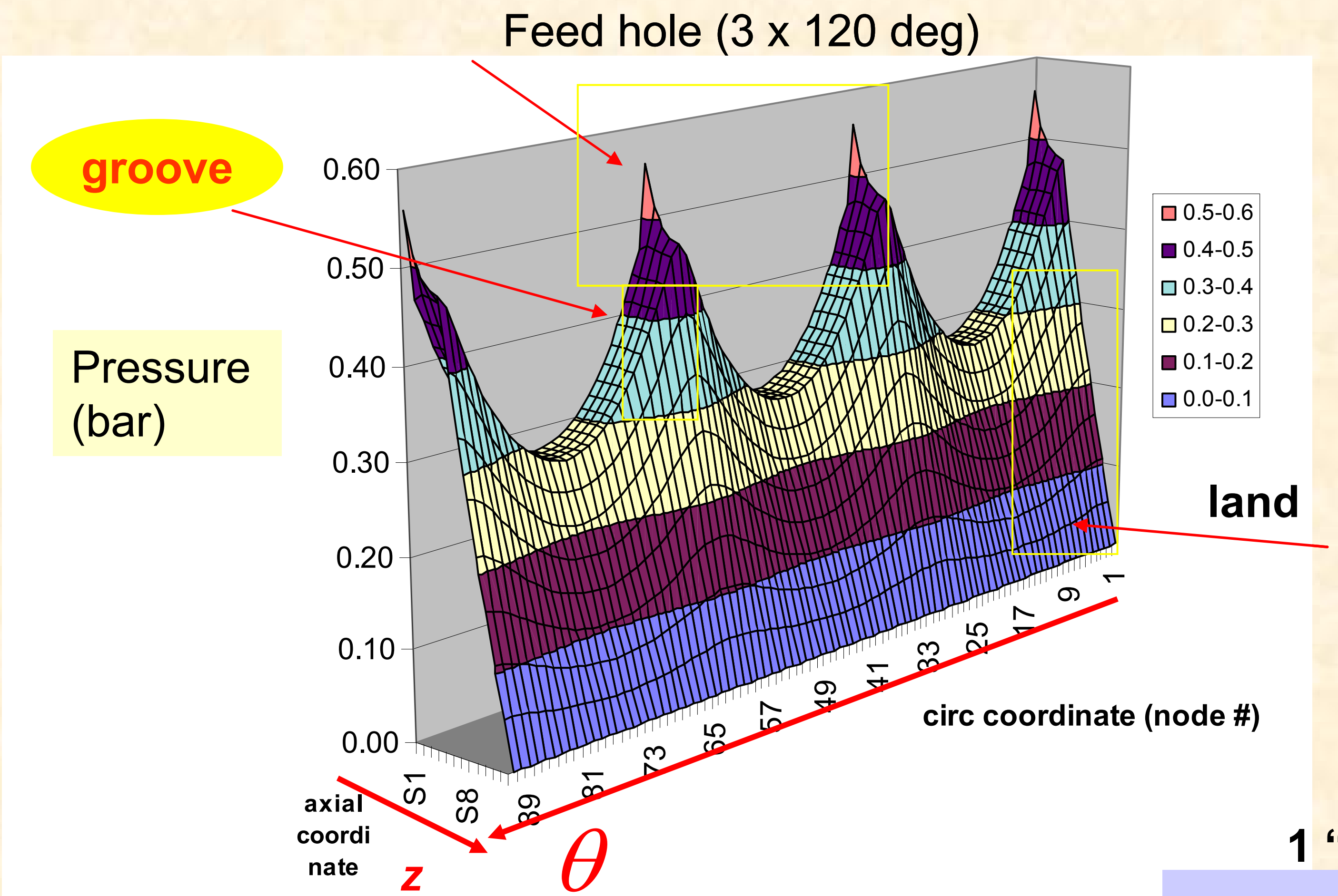
Use effective depth
d_η=1.6c



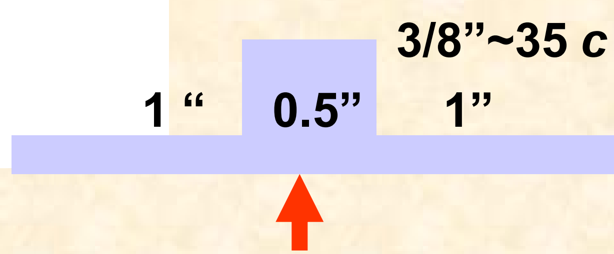
Solve modified Reynolds equation (with fluid inertia)

$$\frac{\partial}{R\partial\Theta}\left(h^3\frac{\partial P}{R\partial\Theta}\right)+\frac{\partial}{\partial z}\left(h^3\frac{\partial P}{\partial z}\right)=12\mu\frac{\partial h}{\partial t}+\rho h^2\frac{\partial^2 h}{\partial t^2}$$

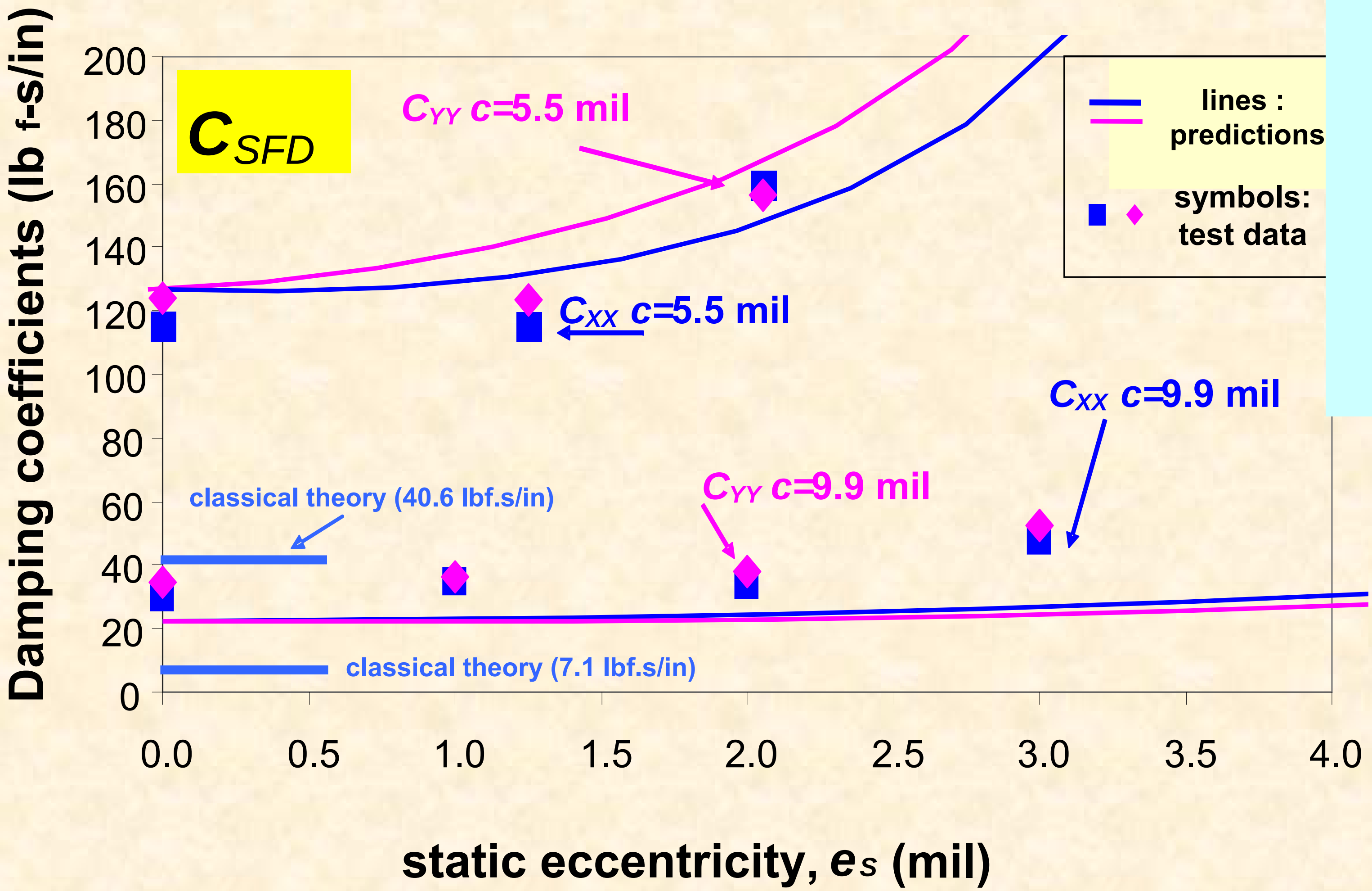
Example predicted pressure field



Static pressure at groove shows circumferential variation due to feed holes spacing



Damping coefficients: test & predictions

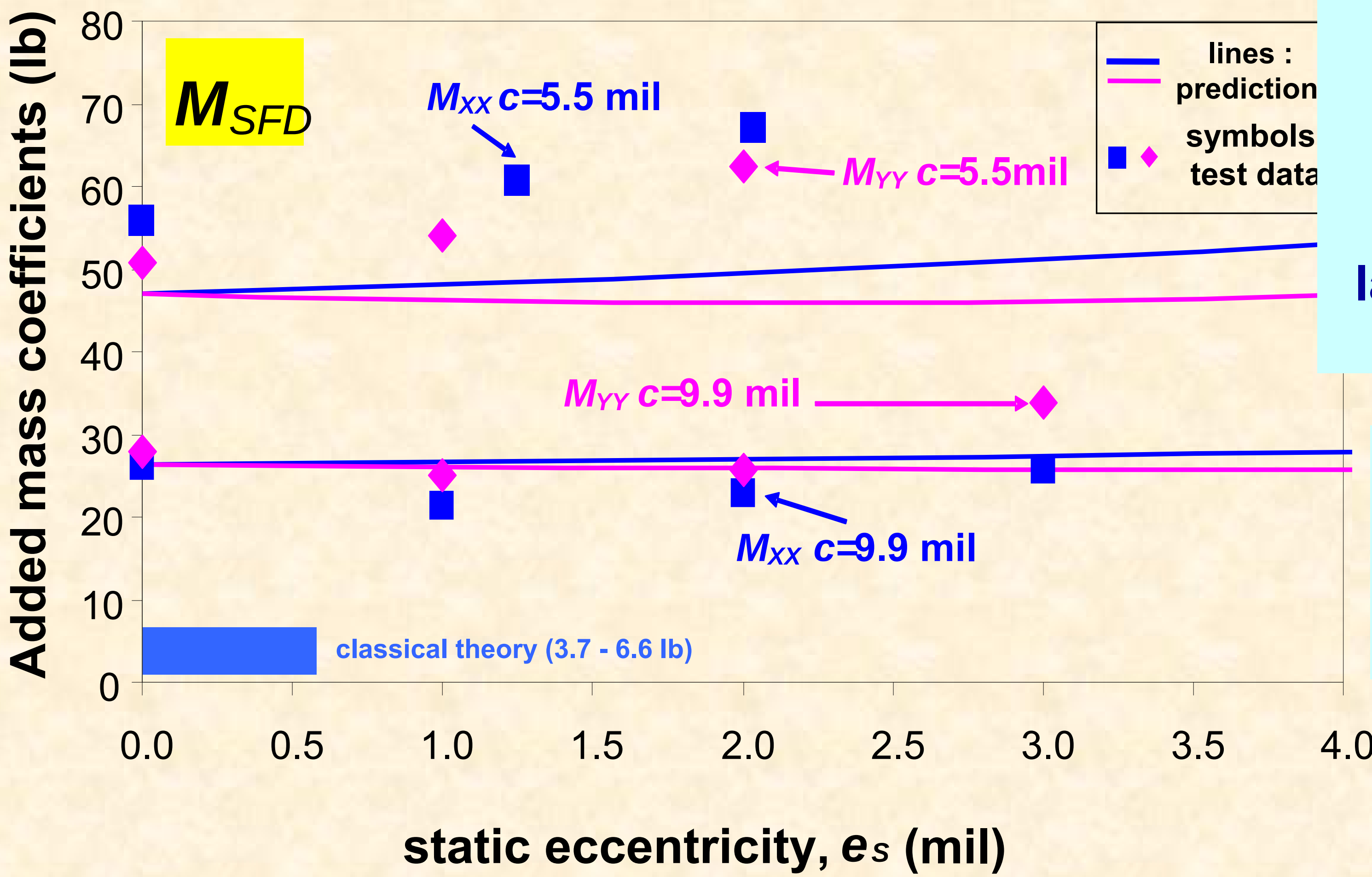


Model predicts

small *c* SFD: larger damping coefficient than test values

large *c* SFD: less damping than test values

Inertia coefficients: test & predictions



Model predicts

small *c* SFD: less inertia than test values

Large *c* SFD: larger inertia than test values

Classical theory predicts ~ 1/7 of test values

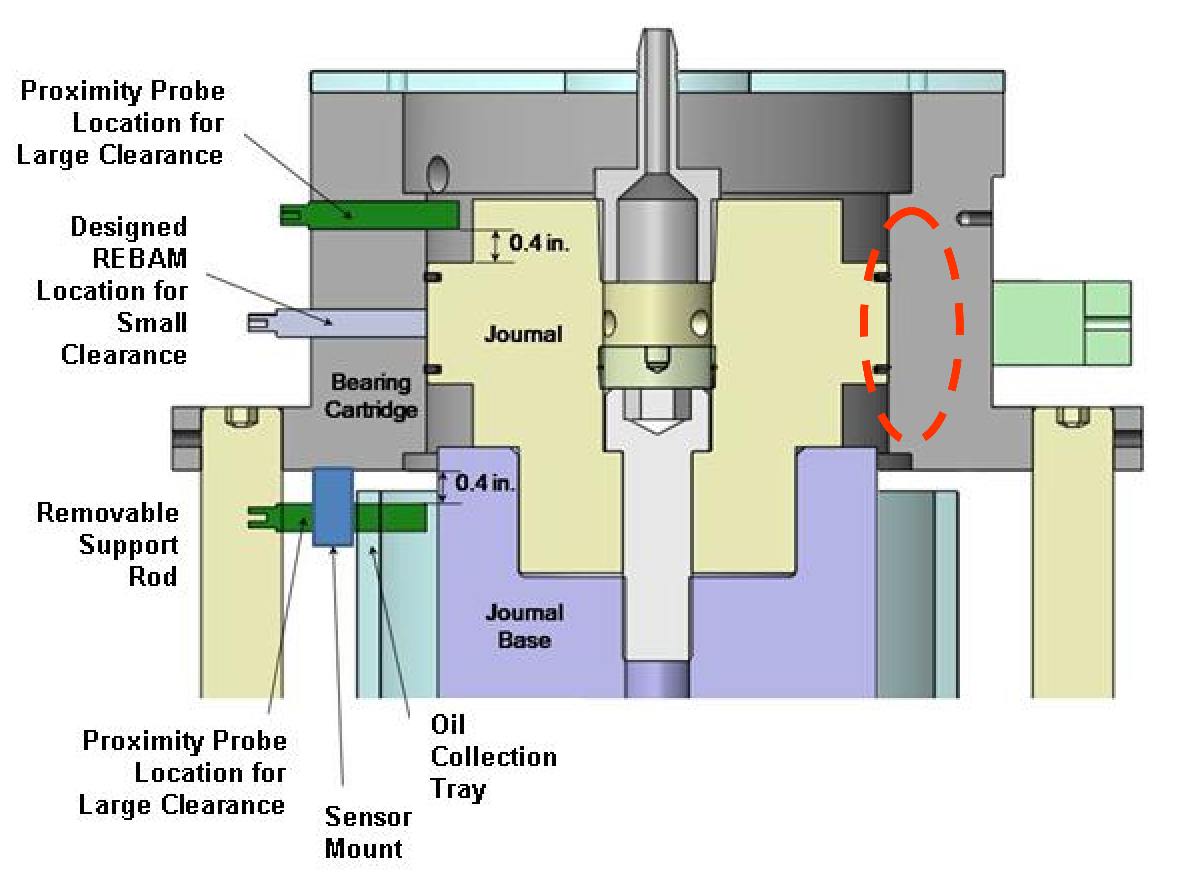
Conclusions

Conducted measurements of dynamic load response in large clearance (*c*=9.9 mil) open ends SFD with circular orbits, centered and off-centered.

- **Central grove is NOT a zone of constant pressure: dynamic pressures as large as in film lands.**
- **Classical theory predicts too low SFD added masses: 1/7 of test values**
- **Using an effective shallow groove depth, new model predictions agree well with test results.**

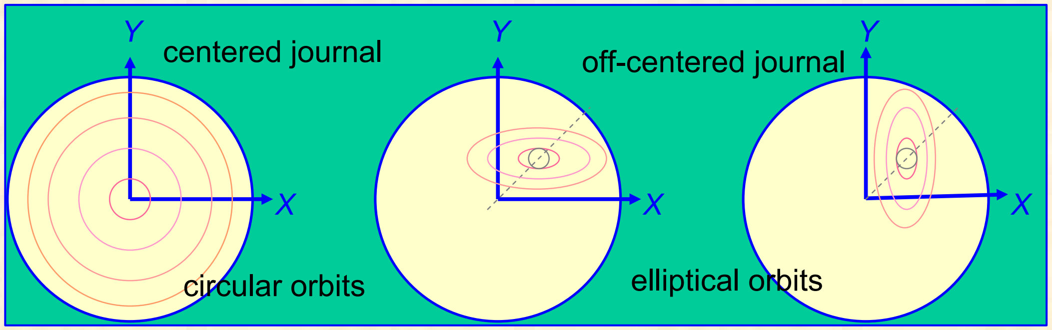
P&W funded project (2012)

Modify test rig and construct SFD w/o a central groove, conduct measurements of film pressures and identify force coefficients.



Proposed tasks TRC (2012-13)

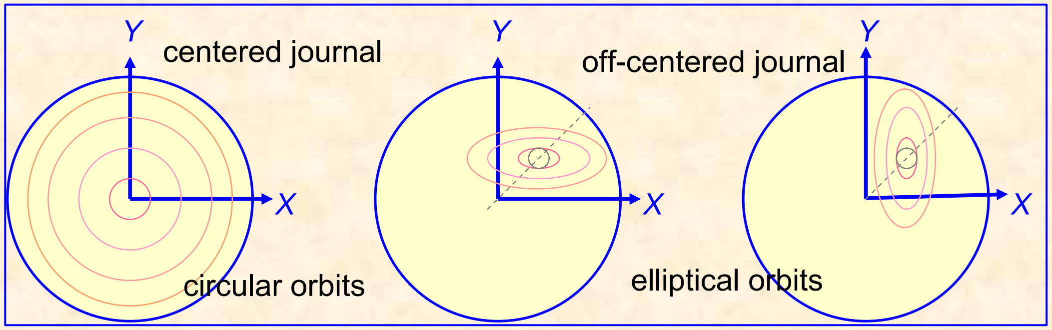
1. Test damper w/o groove with dynamic loads (20-300 Hz) inducing off-centered elliptical orbital motions to reach 0.8*c*.
2. Identify SFD force coefficients from test impedances, and correlate coefficients with linear force coefficients and experimental coefficients for smallest whirl amplitude (0.05*c*).
3. Perform numerical experiments, similar to the physical tests, to extract linearized SFD force coefficients from the nonlinear forces. Quantify goodness of linear-nonlinear representation from an equivalence in mechanical energy dissipation.



TRC Budget (2012-13)

eight months	Year II
Support for graduate student (20 h/week) x $ 2,200 x 8 months	$ 17,600
Fringe benefits (0.6%) and medical insurance ($197/month)	$ 1,682
Travel to (US) technical conference	$ 1,200
Tuition three semesters ($227 credit hour x 15 ch x 1.7 fees multiplicative factor)	$ 5,789
Supplies for test rig	$ 2,200
Total Cost:	**$ 28,470**

Year I started on Jan 2012



Acknowledgments

Thanks to

- Pratt & Whitney Engines
- Turbomachinery Research Consortium
- Sung-Hwa Jeng, RA for making the presentation

Learn more

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Questions (?)