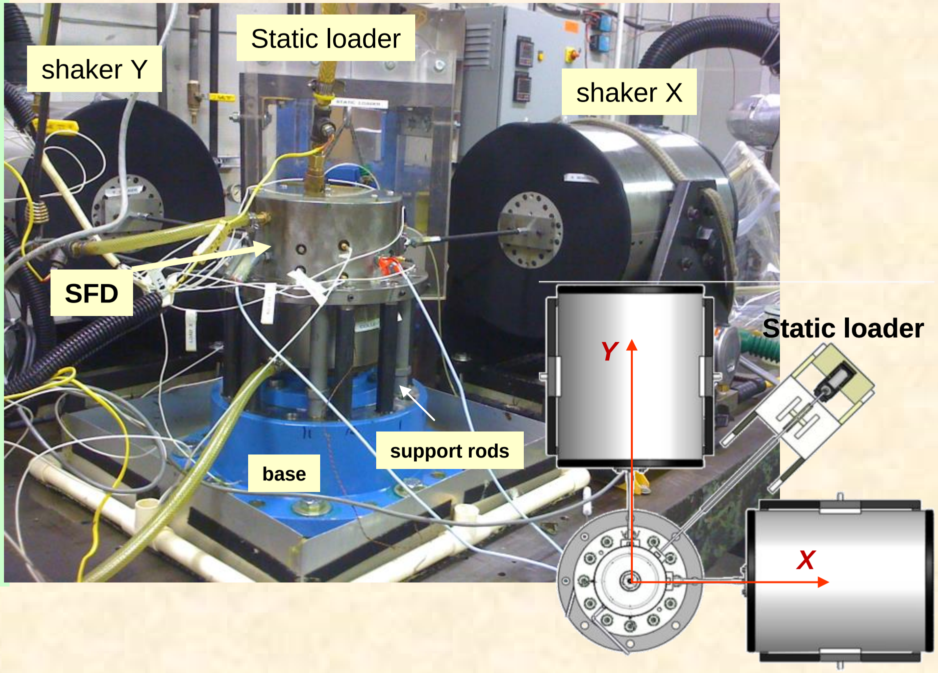
An example of parameter identification

Identification of force coefficients in a test damper



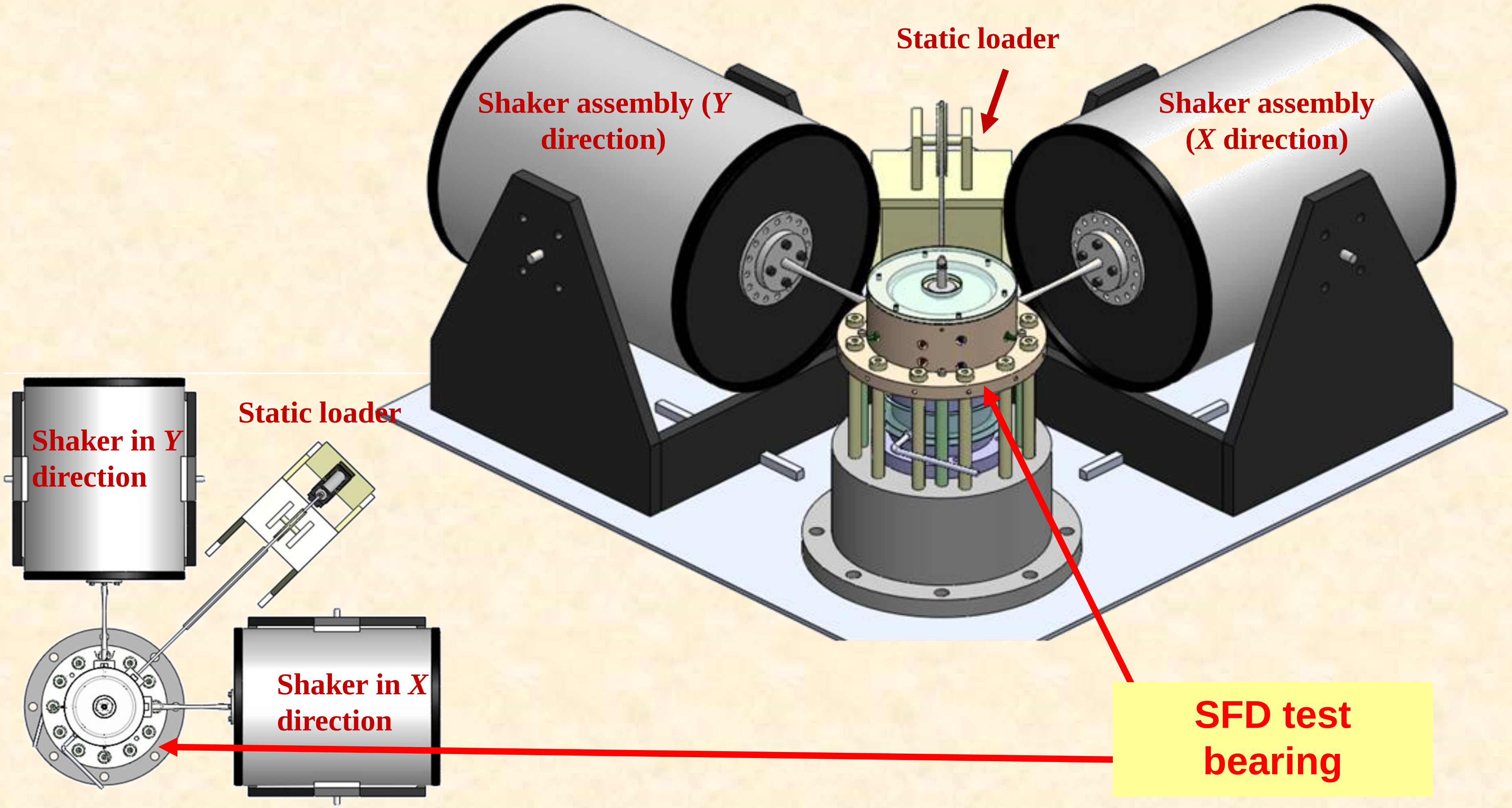
Luis San Andrés

Sanjeev Seshagiri, Paola Mahecha

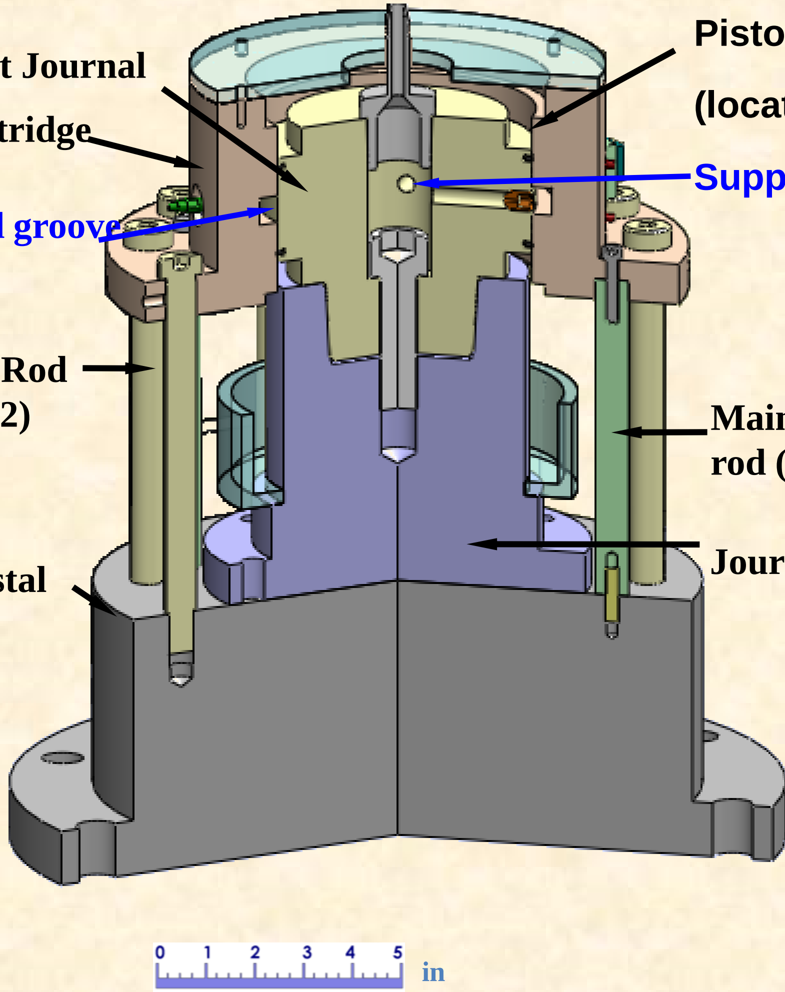
Research Assistants

Sponsor: Pratt & Whitney Engines

P&W SFD test rig

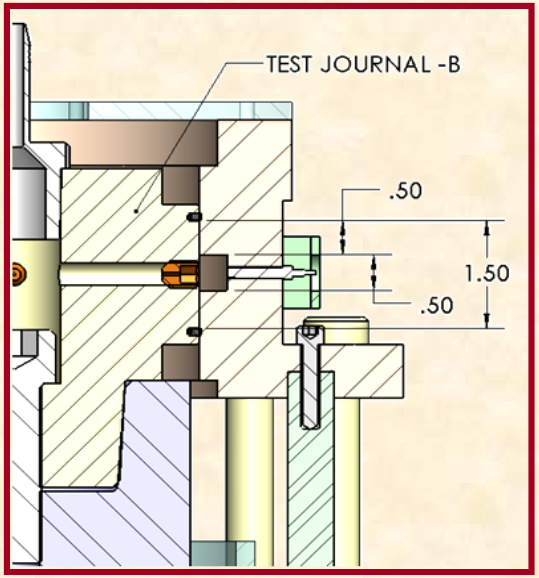


P & W SFD Test Rig – Cut Section



Test rig main features

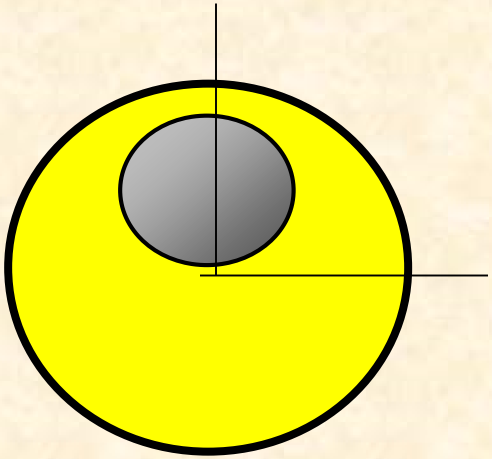
Journal diameter: 5.0 inch
Film clearance: 5.1 mil
Film length: 2 x 0.5 inch
Support stiffness: 22 klbf/in



Objective & task

Evaluate dynamic load performance of SFD.

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

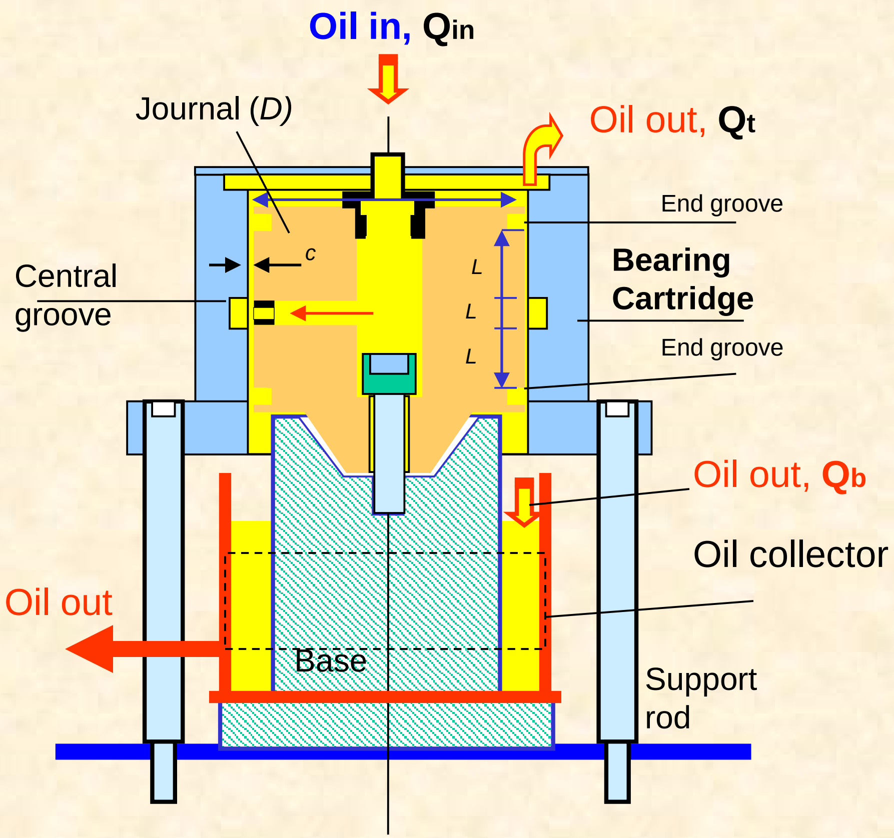


Circular orbit tests

- **Frequency range: 5-85 Hz**
- **Centered and off-centered, e_S/c = 0.20, 0.40, 0.60**
- **Orbit amplitude r/c = 0.05 – 0.50**

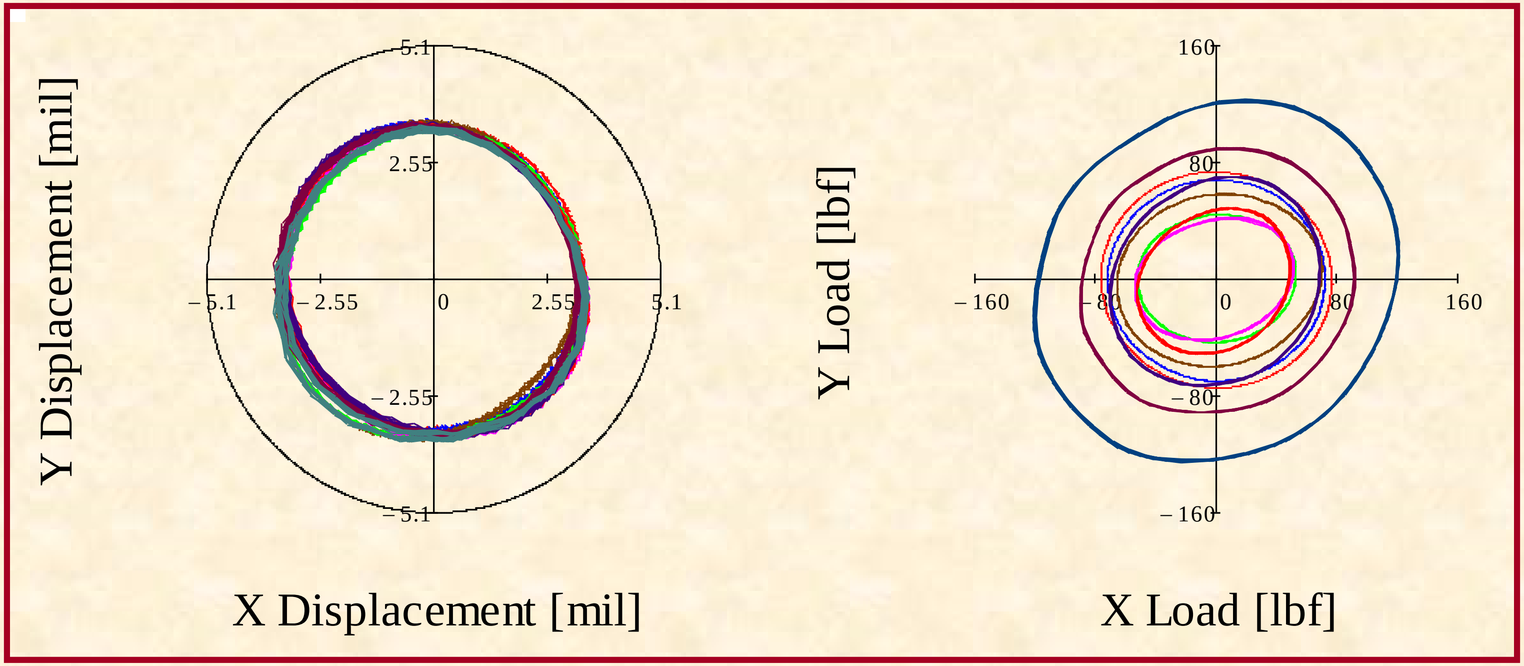
ISO VG 2 Oil

Viscosity at 73.4 °F [cPoise]	**2.95**
Density [kg/m³]	**784**
Inlet pressure [psig]	**7.5**
Outlet pressure [psig]	**0**
Radial Clearance [mil]	***c***
Journal Diameter [inch]	**5.0**
Central groove length [inch]	***L***
Land length, L [inch]	***L***
Total Length [inch]	**3*L***



Typical circular orbit tests

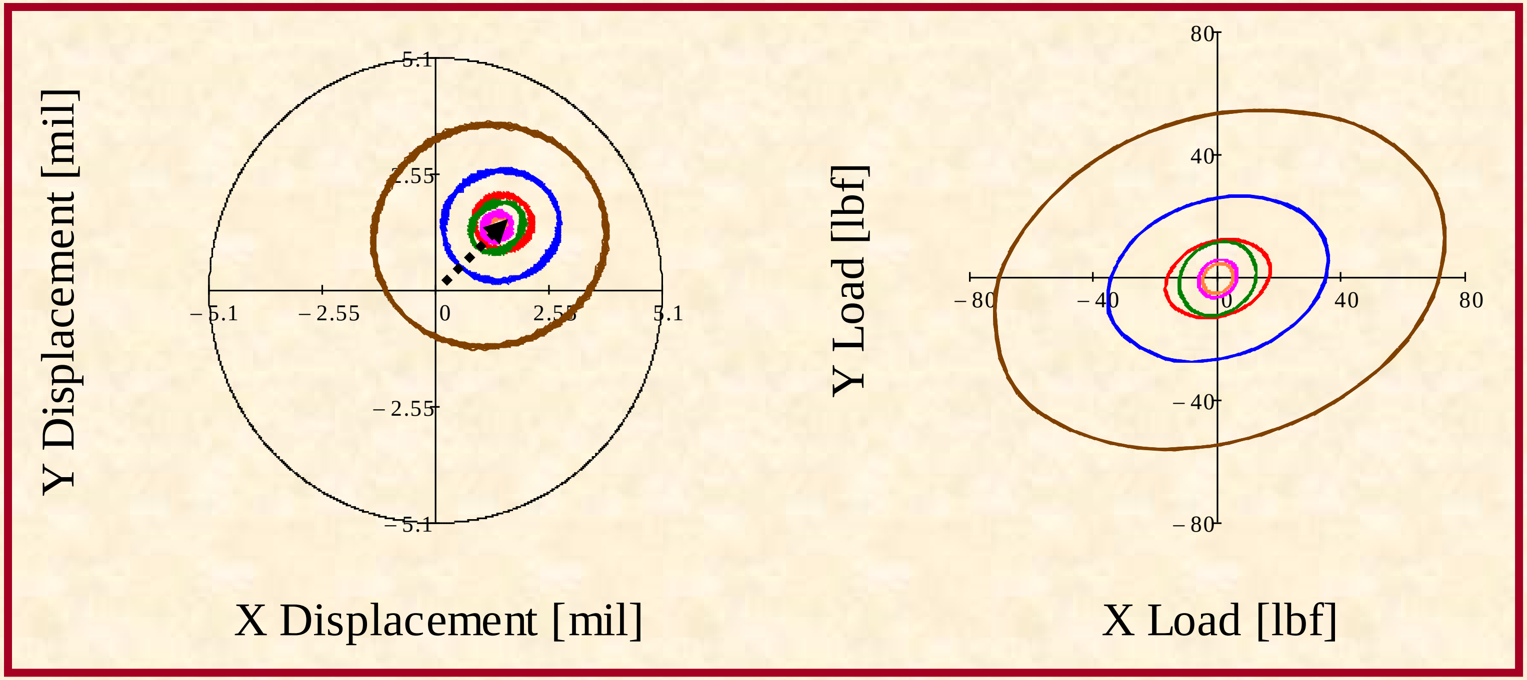
- **Frequency range: 5-85 Hz**
- **Centered e_S=0**
- **Orbit amplitude r/c=0.66**



motion (y vs. x)   **Forces (f_y vs. f_x)**

Typical circular orbit tests

- Frequency: 85 Hz
- Off-centered at e_S/c= 0.31
- Orbit amplitude r=0.05 – 0.5



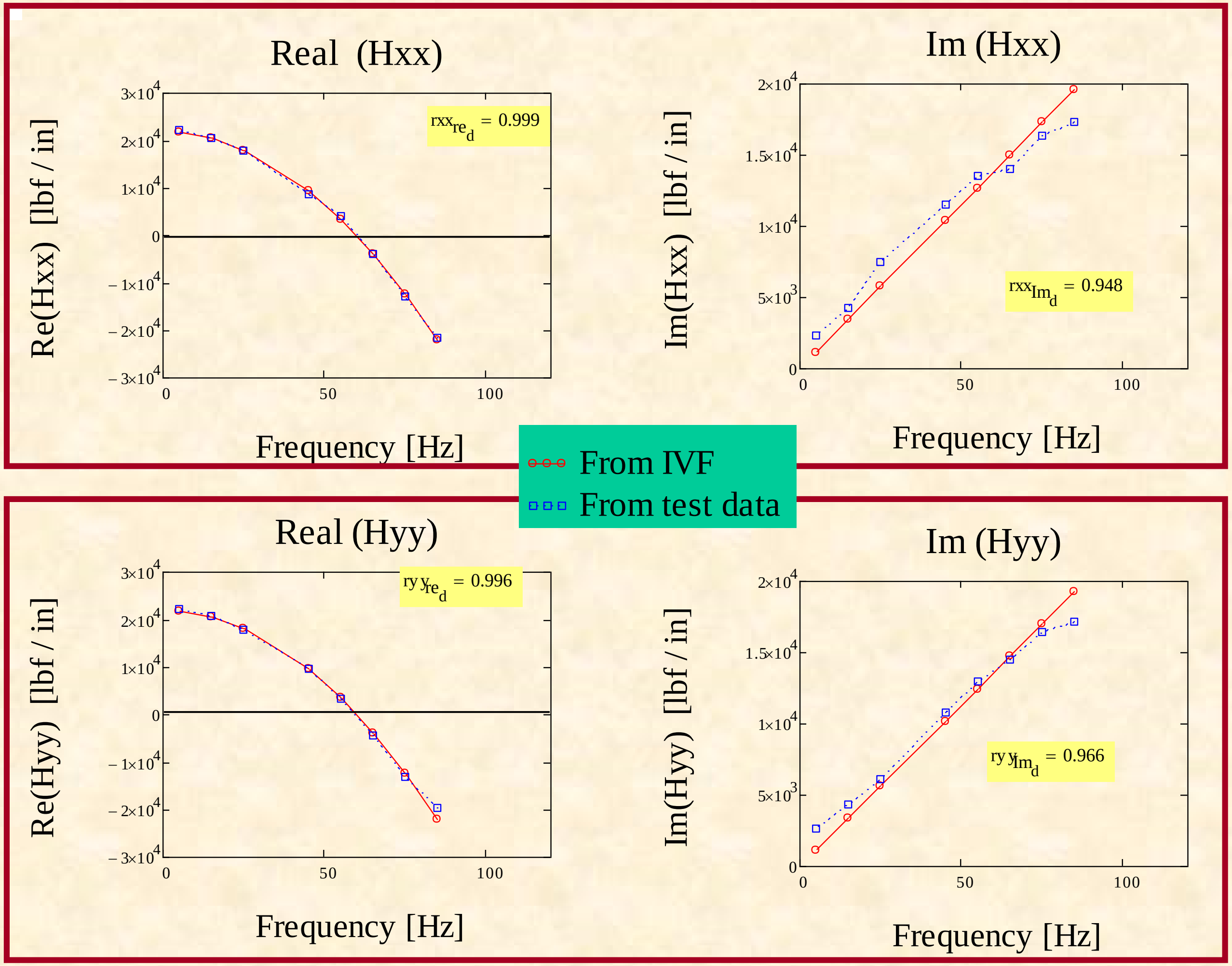
motion (y vs. x)

Forces (f_y vs. f_x)

Typ system direct complex stiffnesses

H_{XX}

H_{YY}

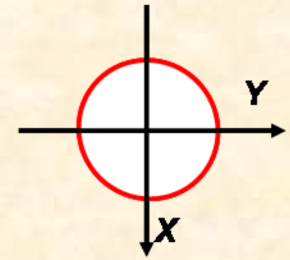


Real part

Imaginary part

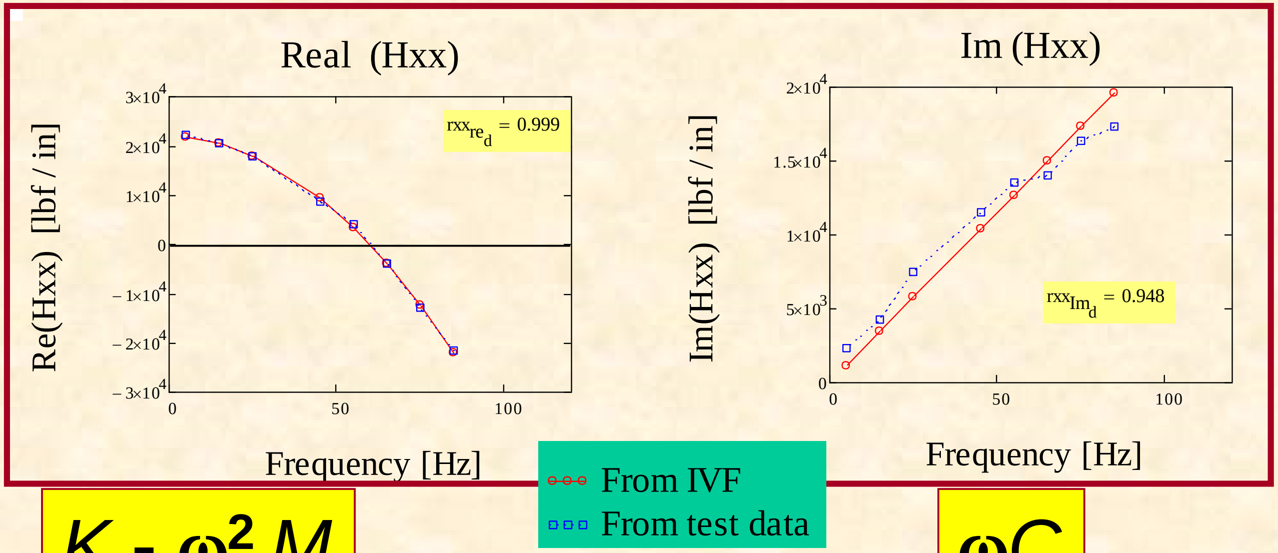
***r/c*= 0.66,**

centered e_s=0

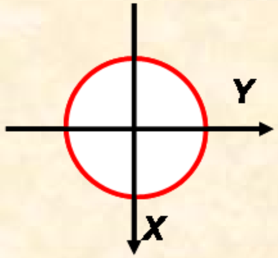


Typ. system direct complex stiffness H_{XX}

H_{XX}



r/c= 0.66,
centered e_s=0



$K - \omega^2 M$

ωC

Excellent correlation between test data and physical model

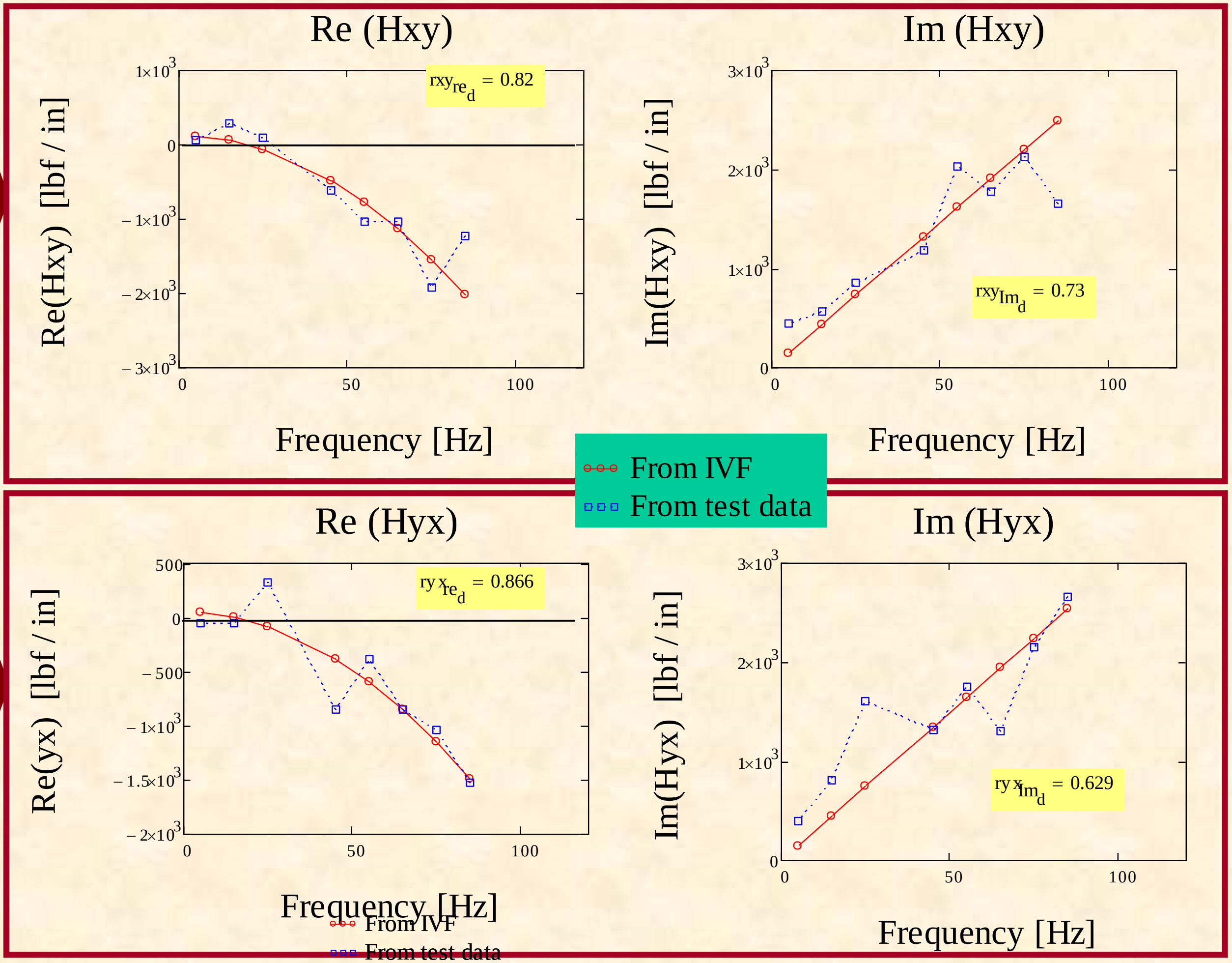
REAL PART = dynamic stiffness

IMAGINARY PART proportional to viscous damping

Test cross-coupled complex stiffnesses

H_{XY}

H_{YX}



***r/c*= 0.66,**
centered e_s=0

One order of magnitude lesser than direct impedances = Negligible cross-coupling effects

Real part **Imaginary part**

SFD force coefficients

SFD

Difference between lubricated system and dry system (baseline) coefficients

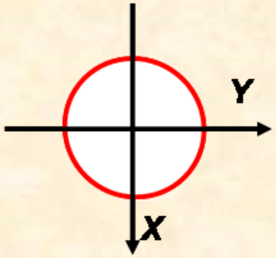
$$C_{SFD} = C_{lubricated} - C_s$$

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

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

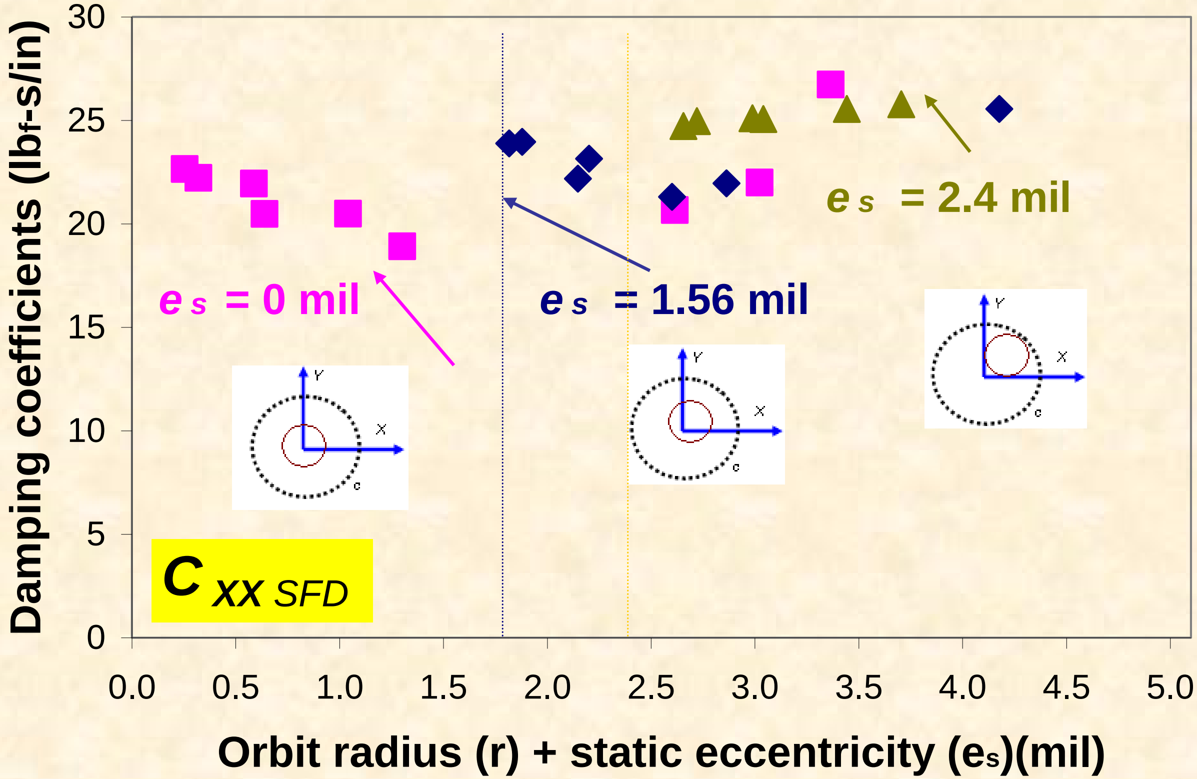
DRY system parameters

K_s = 21 klbf/in
M_s = 40 lb
C_s = 7 lbf-s/in
Nat freq = 73-75 Hz
Damping ratio = 0.04

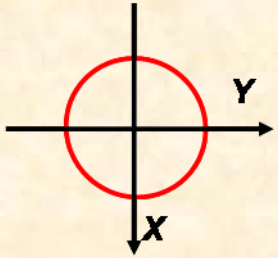


SFD damping coefficients

C_{XX}



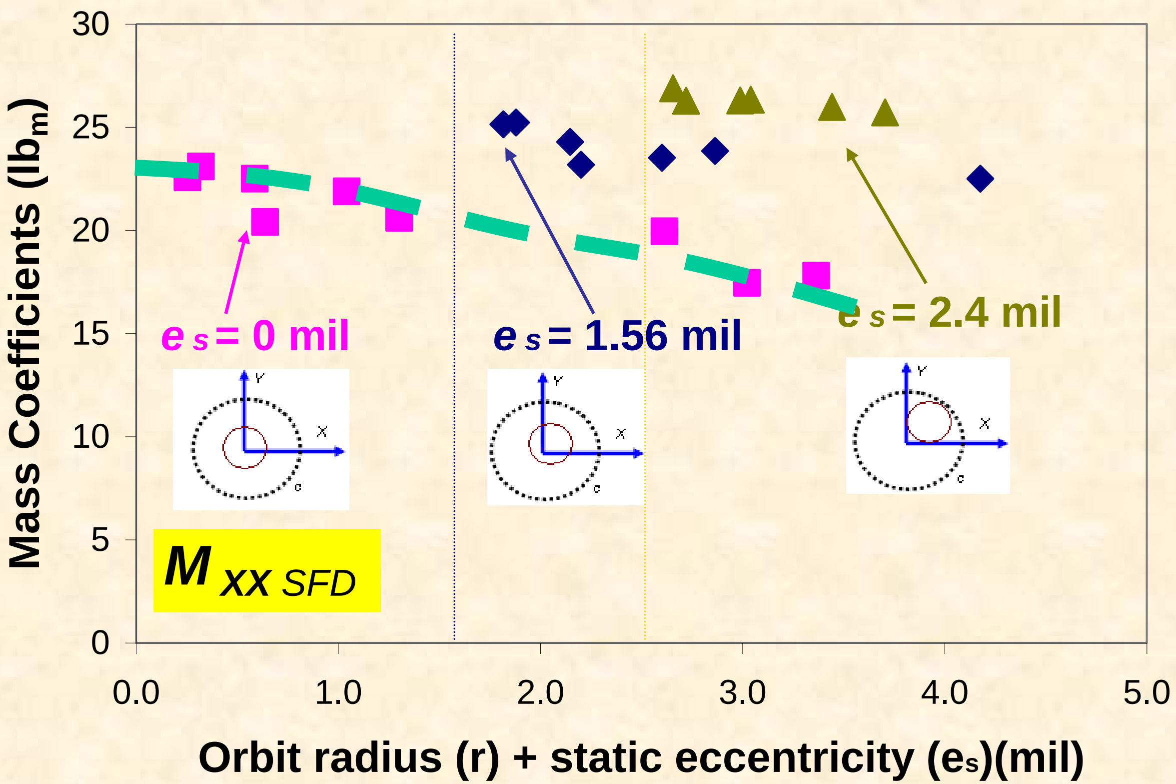
Damping increases mildly as static eccentricity increases



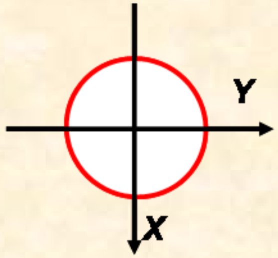
C_{YY} ~ C_{XX} for circular orbits, independent of static eccentricity

SFD mass coefficients

M_{XX}



M_{XX} ~ M_{YY} **decreases with orbit radius (*r*) for centered motions. Typical nonlinearity**



Conclusions

- **SFD test rig: completed measurements of dynamic loads inducing small and large amplitude orbits, centered and off-centered.**

- **Identified SFD damping and inertia coefficients behave well. IVFM delivers reliable and accurate parameters.**

- **Comparison to predictions are a must to certify the confidence of numerical models.**

References

Fritzen, C. P., 1985, "Identification of Mass, Damping, and Stiffness Matrices of Mechanical Systems," ASME Paper 85-DET-91.

Massmann, H., and R. Nordmann, 1985, "Some New Results Concerning the Dynamic Behavior of Annular Turbulent Seals," Rotordynamic Instability Problems of High Performance Turbomachinery, Proceedings of a workshop held at Texas A&M University, Dec, pp. 179-194.

Diaz, S., and L. San Andrés, 1999, "A Method for Identification of Bearing Force Coefficients and its Application to a Squeeze Film Damper with a Bubbly Lubricant," STLE Tribology Transactions, Vol. 42, 4, pp. 739-746.

San Andrés, L., 2012, "Damping And Inertia Coefficients for Two Open Ends Squeeze Film Dampers with a Central Groove: Measurements and Predictions," ASME paper GT2012-68212

San Andrés, L..., and Delgado, A., 2012, "A Novel Bulk-Flow Model for Improved Predictions of Force Coefficients in Grooved Oil Seals Operating Eccentrically," ASME J. Eng. Gas Turbines Power, vol. 134 (May), 022509

Questions (?)

Learn more: http:/rotorlab.tamu.edu