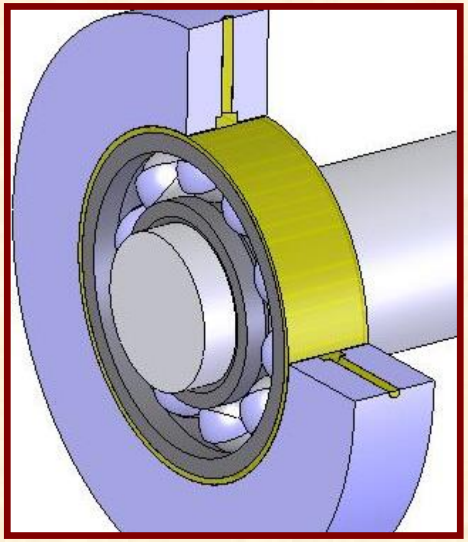
Presentation to Tianjin University

Squeeze Film Dampers
Do’s and Don’ts

May 11 2018



Dr. Luis San Andres
Mast-Childs Chair Professor
Mechanical Engineering Department
Texas A&M University

TURBOMACHINERY LABORATORY
TEXAS A&M ENGINEERING EXPERIMENT STATION

Luis San Andrés

Mast-Childs Chair Professor at Texas A&M University – Turbomachinery Laboratory. He Performs research in lubrication and rotordynamics. Luis is a Fellow of ASME and STLE, a member of the Industrial Advisory Committees for the Texas A&M Turbomachinery Symposia, a Chair of the 2018 Global Power and Propulsion Society Forum, and a Technical Advisor for the Chinese International Turbomachinery Conference.

Dr. San Andres is an Associate Editor for Tribology Transactions and a former Associate Editor for various ASME journals. Luis and students have published over 260 journal and conference papers. Several papers are recognized as best in various international conferences.

Most common problems in rotordynamics

1. Excessive steady state synchronous vibration levels:

Improve balancing.

Modify rotor-bearing systems: tune system critical speeds out of RPM operating range.

SFDs

Introduce damping to limit peak amplitudes at critical speeds that must be traversed.

2. Subharmonic rotor instabilities

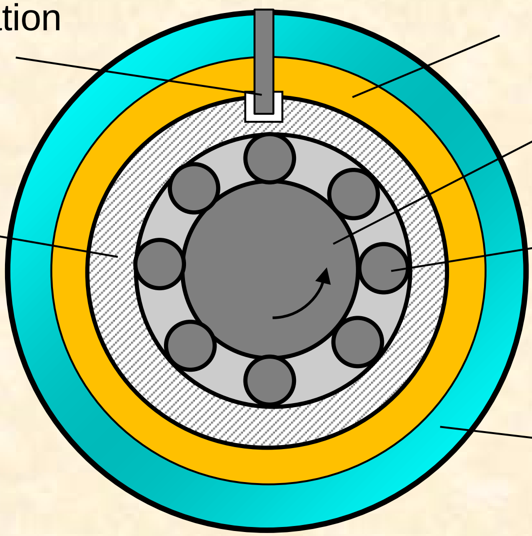
Eliminate instability mechanism, i.e. change bearing design if oil whip is present.

Raise natural frequency of rotor system as much as possible.

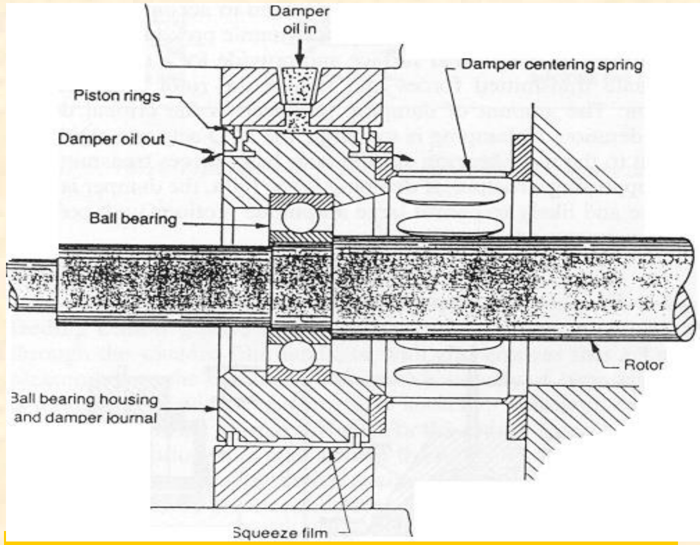
SFDs

Introduce damping to increase onset rotor speed above the operating speed range.

SFD Operation



SFD with dowel pin

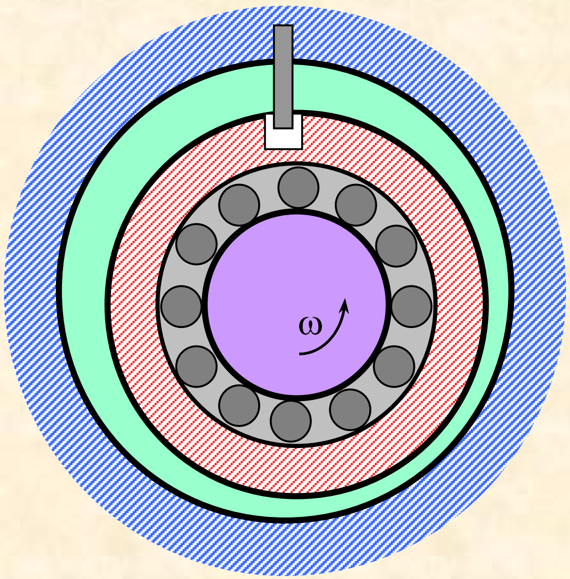


SFD in parallel with squirrel cage

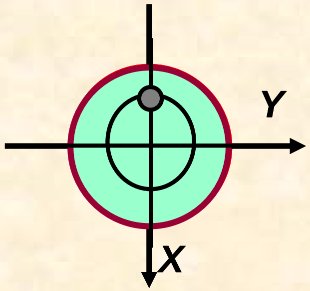
In a SFD, the journal whirls but does not spin. The lubricant film is squeezed due to rotor motions, and fluid film (damping) forces are generated as a function of the journal velocity.

The shaft is mounted on ball bearings, its outer race prevented from rotation with either a squirrel cage (US), or a dowel pin (UK).

SFD operation & design



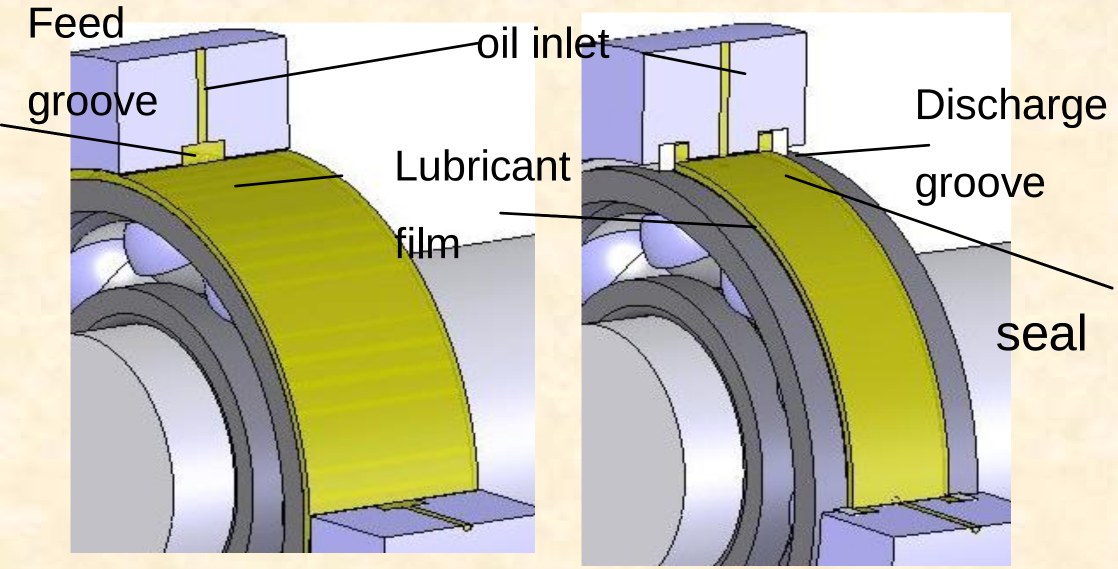
SFD with dowel pin



In aircraft gas turbines and compressors, squeeze film dampers aid to attenuate rotor vibrations and to provide mechanical isolation.

Too little damping may not be enough to reduce vibrations. Too much damping may lock damper & degrades system rotordynamic performance

SFD dynamic force performance



depends on

a) **Geometry** (*L*, *D*, *c*)

b) **Lubricant** (density, viscosity)

c) **Supply pressure and through flow conditions (grooves)**

d) **Sealing devices**

e) **Operating speed** (frequency) & **journal kinematics**

- **Flow regimes:** (laminar, superlaminar, turbulent)
- **Type of lubricant cavitation**:

  gaseous or vapor

  air ingestion & entrapment

Brief literature review

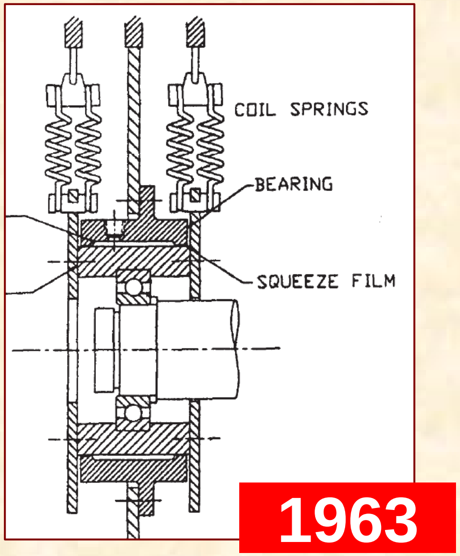
Parsons (1889)

Discloses first use of a SFD as a part of the first modern-day steam turbine.

(by Topori at Polish Wikipedia)

Cooper (1963)

Rolls Royce engineer investigates experimentally the performance of rotating machinery with a SFD.



In 1970s, SFDs become essential components in aircraft engines and multistage high pressure centrifugal compressors.

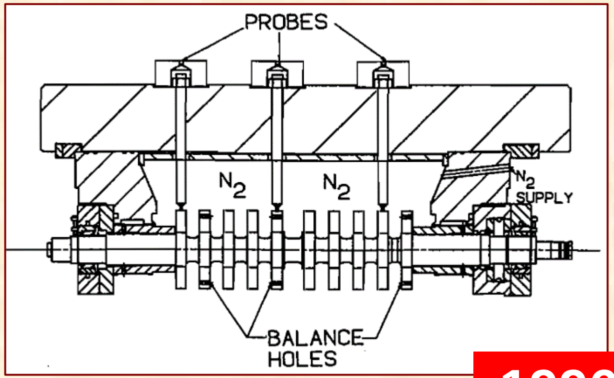
Brief literature review (Turbomachinery Symposium)

Zeidan et al. (1996)

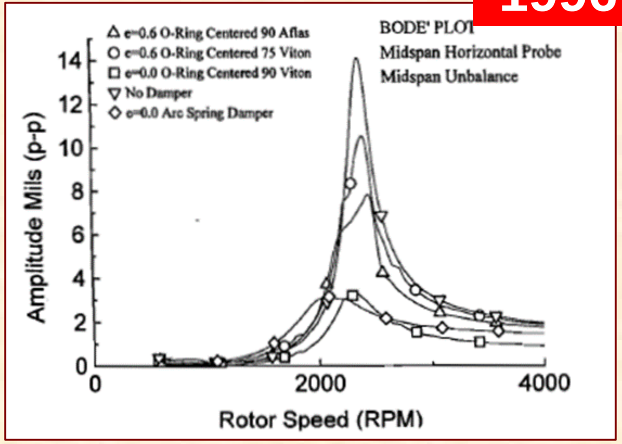
Give historical development of SFDs since 1960's and discuss major technical issues for their integration into turbomachinery, including oil cavitation vs. air ingestion and fluid inertia effects.

Kuzdal and Hustak (1996)

Test various damper configurations (open and sealed ends) → optimized SFD reduces rotor synchronous motions and improves the stability threshold of rotor bearing systems.



1996



Other relevant past work

- **Della Pietra and Adilleta (2002):** Comprehensive review of research conducted on SFDs over last 40 years.

Parameter identification in SFDs:

- **Tiwari et al. (2004):** Comprehensive review of parameter identification in fluid film bearings.

(2006-2010) San Andrés and Delgado (SFD & MECHANICAL SEAL, improved predictive models).

GT 2006-91238, GT 2007-24736, GT 2008-50528, GT 2009-50175

(2012-2018) San Andrés and students (Sealed ends SFD and with feed central groove)

GT 2012-68212 , GT 2013-94273 , GT 2014-26413, GT 20015-43096, GT 2016-43096, GT 2016-56695, 2016 ATPS & TPS, GT2017-63152, GT2018-76224, 18-STLE, 18-IFToMM

SFD applications

Jet engines with rolling element bearings:

a) To reduce synchronous peak amplitudes,
b) Limit peak amplitudes at critical speeds,
c) To isolate structural components (lower transmissibility), and
d) To provide a margin of safety for blade loss.

Light hydrocarbon compressors with instability problems

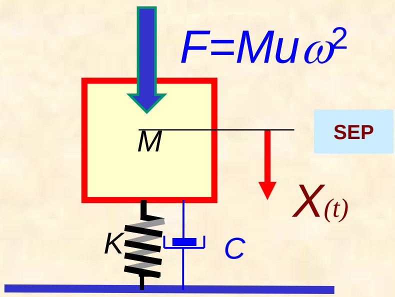
a) To stabilize unit by introducing damping and reducing cross-coupled effect of seals, hydrodynamic bearings, etc.
b) To enhance limited damping available from tilting pad bearings.

Other benefits of SFDs on rotordynamic performance are:

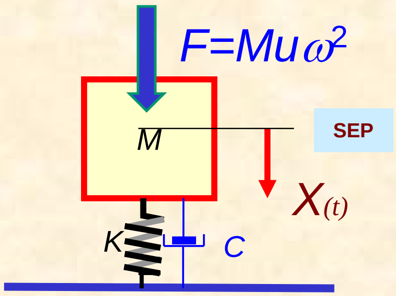
- Tolerance to larger rotor motions
- Reduced balancing requirements

* Simpler alignment
* Less mount fatigue

What is the effect of viscous damping on the dynamic response of a mechanical system?



Simple spring-damper-mass system



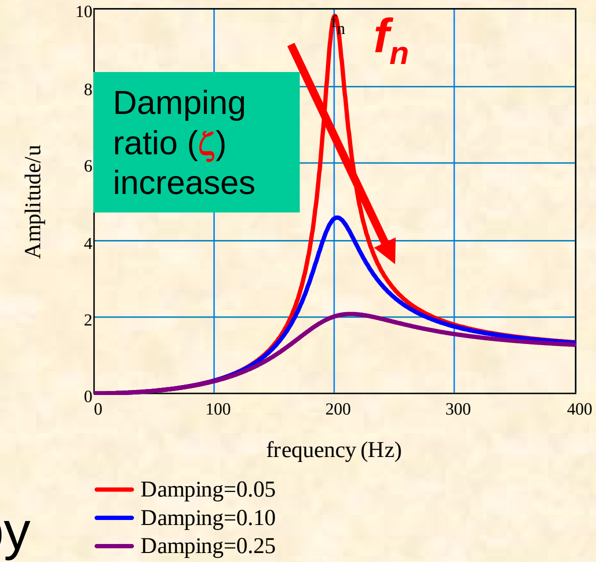
EOM: $M A_x = F - F_{damper} - F_{spring}$

$$M\ddot{X} + C\dot{X} + KX = F_{(t)}$$

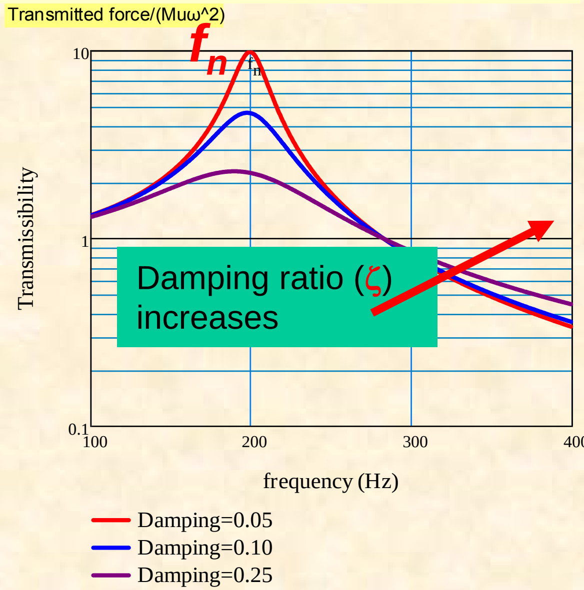
System response defined by natural frequency (f_n) & damping ratio (ζ)

$$f_n = 2\pi\sqrt{K/M};\quad \zeta = C/\left(2\sqrt{KM}\right)$$

Response **amplitude** |*X/u*|

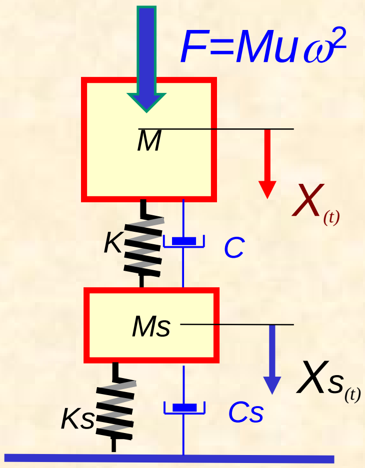


Transmissibility (to ground)



Damping helps only when rotor traverses a critical speed (natural frequency=f_n) but increases force transmissibility for operation above 1.44 f_n

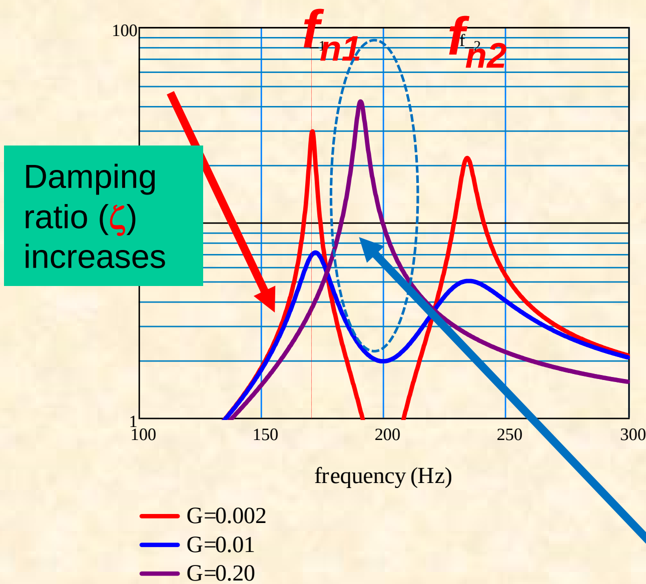
More complex K-C-M system : rotor on flexible supports



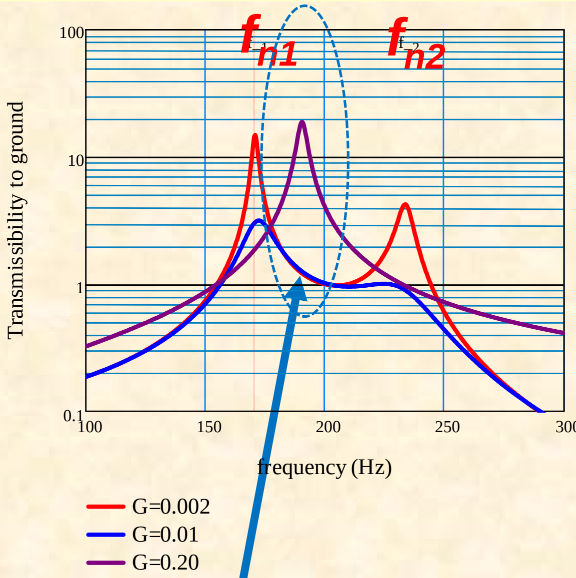
example

$$\frac{M_s}{M} = \frac{K_s}{K} = 0.1;\ C = 0;\ C_s \text{ varies}$$

Response **amplitude** |*X/u*|



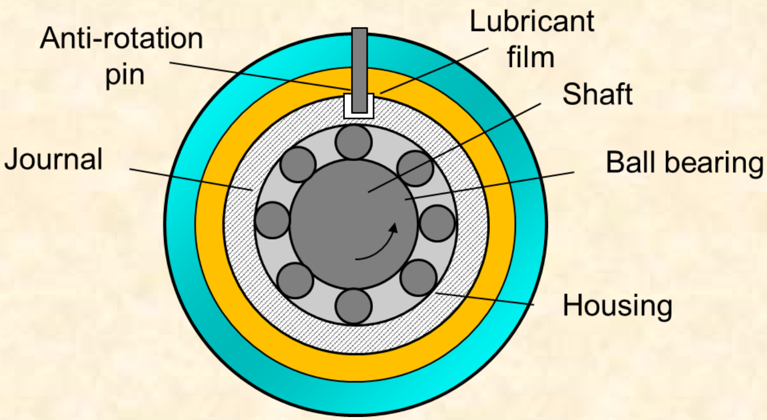
Transmissibility (to ground)



Excessive damping LOCKS supports and increases system response.

More complicated response. Damping helps only when traversing a critical speed (natural frequency=f_{n1} and f_{n2}) but increases force transmissibility.

SFDs – the bottom line



Too little damping may not be enough to reduce vibrations.

Too much damping may lock damper & will degrade system performance.

SFDs must be designed with consideration of the whole rotor-bearing system.

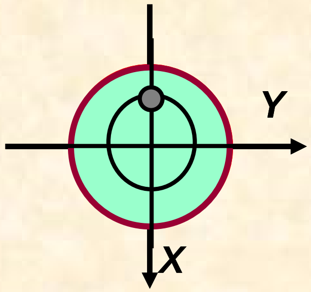
Physical damping is not as important as the system damping ratio!

$$\zeta = C / C_{crit} = C / 2\sqrt{K\,M}$$

SFD models for forced response

Damping is needed for safe passage through critical speeds and to provide or increase system stability.

Thus, models for SFD forced response are:



Imbalance response analysis:

SFD forces for circular centered whirl orbits.

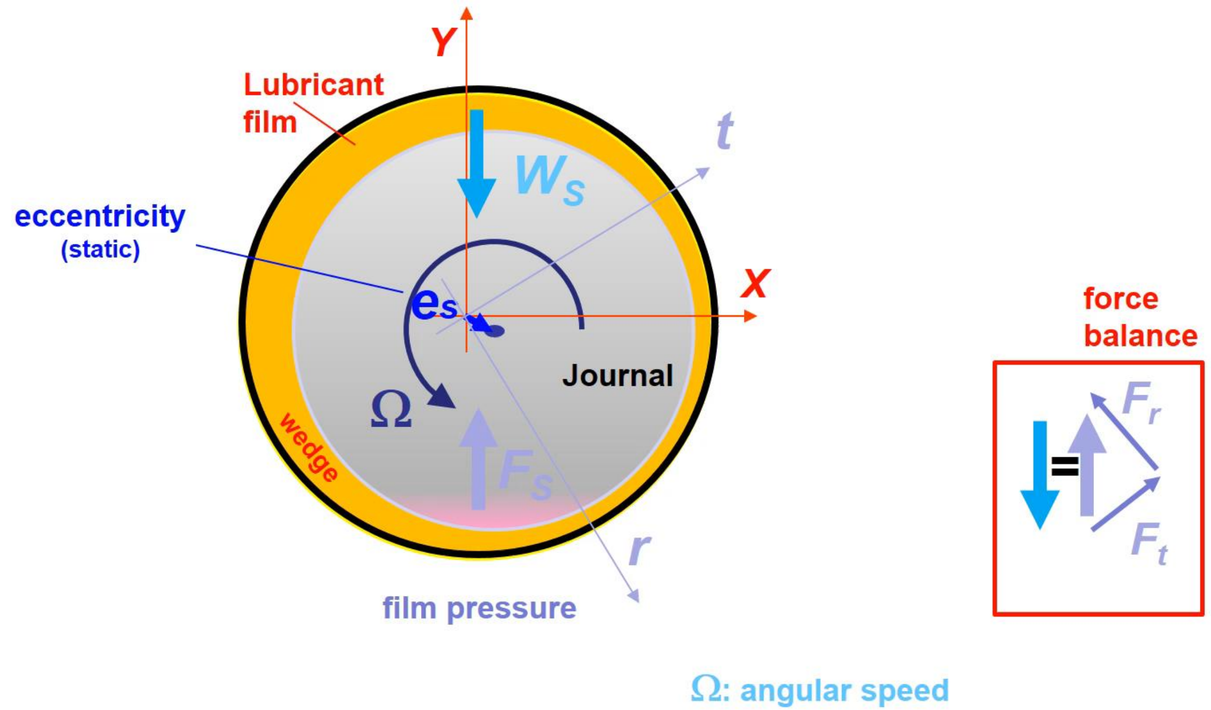
Rotordynamic eigenvalue & stability analysis:

SFD force coefficients for dynamic journal motions about a static (equilibrium) position.

Impedance formulations for transient response analysis of rotor-bearing response.

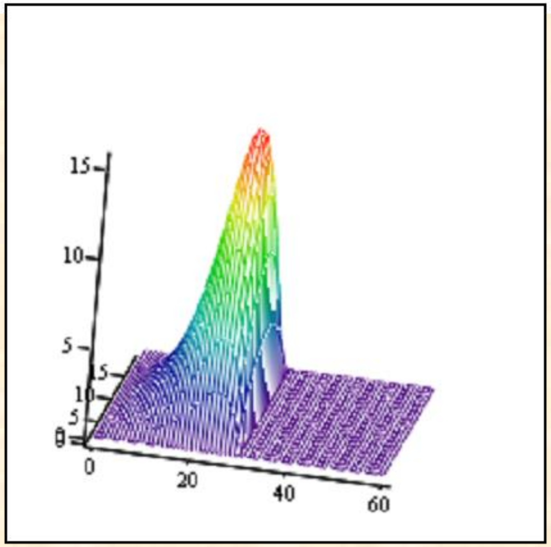
Early (ab)use in academic studies; nowadays common with fast PCs

Journal bearing model: steady state



$v_t = 0$; $v_r = 0$

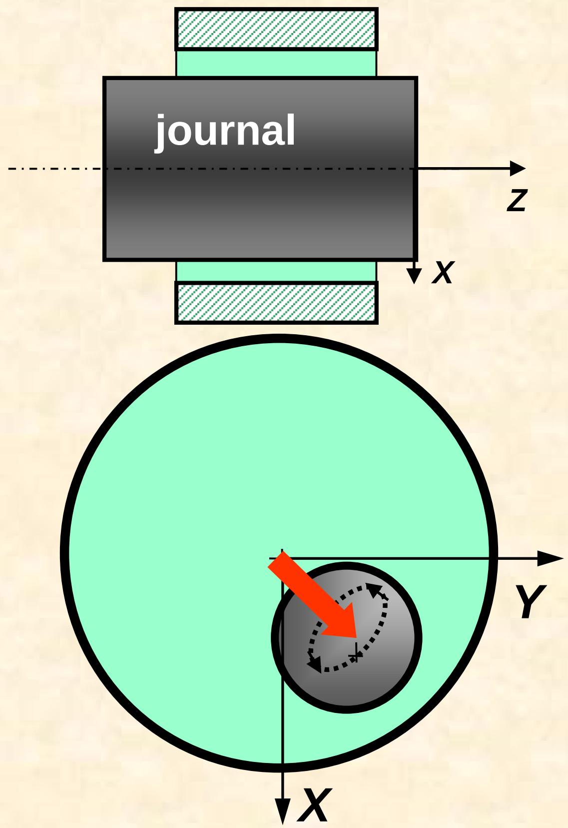
$a_r = 0$; $a_t = 0$



P_S *

Pressure field is invariant with time and increases as film thickness decreases to a minimum.

SFD model: journal motions off-centered



V_X , V_Y : journal center velocity (X,Y)
A_X , A_Y : journal center acceleration (X,Y)

Forces in a SFD describing small amplitude motions about a static off centered journal position

$$-F_X = C_{XX} V_X + C_{XY} V_Y + M_{XX} A_X + M_{XY} A_Y$$

$$-F_Y = C_{YX} V_X + C_{YY} V_Y + M_{YX} A_X + M_{YY} A_Y$$

C: damping, **M:** inertia force coefficients

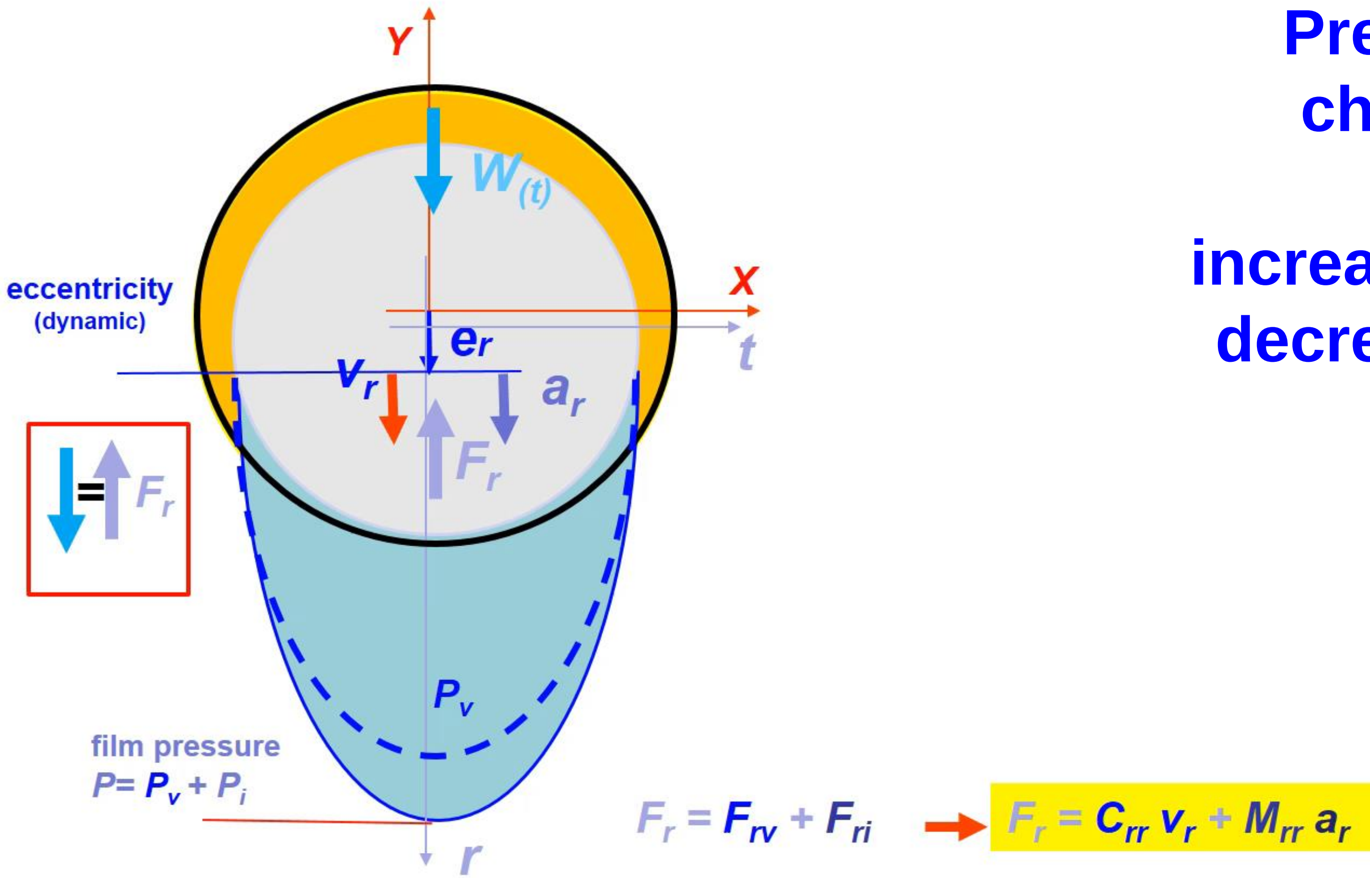
SFD force coefficients

NL functions of static journal eccentricity e_s

$$-\begin{Bmatrix} F_X \\ F_Y \end{Bmatrix} = \begin{bmatrix} C_{XX} & C_{XY} \\ C_{YX} & C_{YY} \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} + \begin{bmatrix} M_{XX} & M_{XY} \\ M_{XY} & M_{YY} \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix}$$

SFD model: pure radial squeeze (plunging motion)

0 ; $v_r = f(t)$
$a_r = 0$; $a_t = 0$



Pressure field changes with time and increases as film decreases. Note pressure reversals.

SFD model: off-center motions

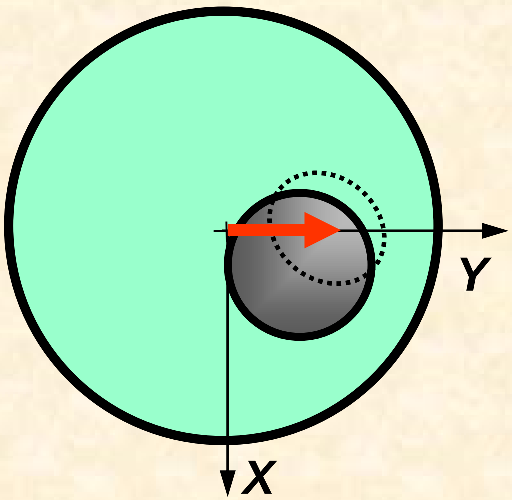
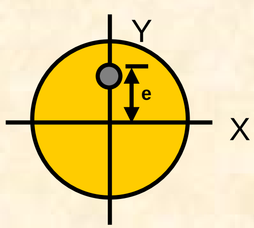


Table 1: **Linearized force coefficients for open ends SFD**

(small amplitude motions about off-center static position $\varepsilon = e/c$)

Full film (No cavitation)	π-Film Model (Cavitated)
$C_{XX} = \mu D \left(\frac{L}{c}\right)^3 \frac{\pi\left(1+2\varepsilon^2\right)}{2\left(1-\varepsilon^2\right)^2}$	$C_{XX} = \mu D \left(\frac{L}{c}\right)^3 \frac{\pi}{2} \frac{\left[3\varepsilon + i\,\frac{\left(1+2\varepsilon^2\right)}{2}\right]}{2\left(1-\varepsilon^2\right)^2}$
$C_{XY} = 0$	$C_{XY} = \mu D \left(\frac{L}{c}\right)^3 \frac{\varepsilon}{\left(1-\varepsilon^2\right)^2}$
$C_{YY} = \mu D \left(\frac{L}{c}\right)^3 \frac{\pi}{2\left(1-\varepsilon^2\right)^{3/2}}$	$C_{YY} = \mu D \left(\frac{L}{c}\right)^3 \frac{\pi}{4\left(1-\varepsilon^2\right)^{3/2}}$
$C_{YX} = 0$	$C_{YX} = 0$
$M_{XX} = \rho D \left(\frac{L^3}{c}\right) \frac{\alpha\pi\left[1-\left(1-\varepsilon^2\right)^{1/2}\right]}{12\,\varepsilon^2\left(1-\varepsilon^2\right)^{1/2}}$	$M_{XX} = \rho D \left(\frac{L^3}{c}\right) \frac{\alpha\,(i - \pi - 2\varepsilon)}{24\,\varepsilon^2}$
$M_{XY} = 0$	$M_{XY} = \rho D \left(\frac{L^3}{c}\right) \frac{\alpha\left[\ln\left\{\frac{(1-\varepsilon)}{(1+\varepsilon)}\right\} - 2\varepsilon\right]}{24\,\varepsilon^2}$
$M_{YY} = \rho D \left(\frac{L^3}{c}\right) \frac{\alpha\;\pi\left[1-\left(1-\varepsilon^2\right)^{1/2}\right]}{12\,\varepsilon^2}$	$M_{YY} = \rho D \left(\frac{L^3}{c}\right) \frac{\alpha\;\pi\left[1-\left(1-\varepsilon^2\right)^{1/2}\right]}{24\,\varepsilon^2}$
$M_{YX} = 0$	$M_{YX} = 0$

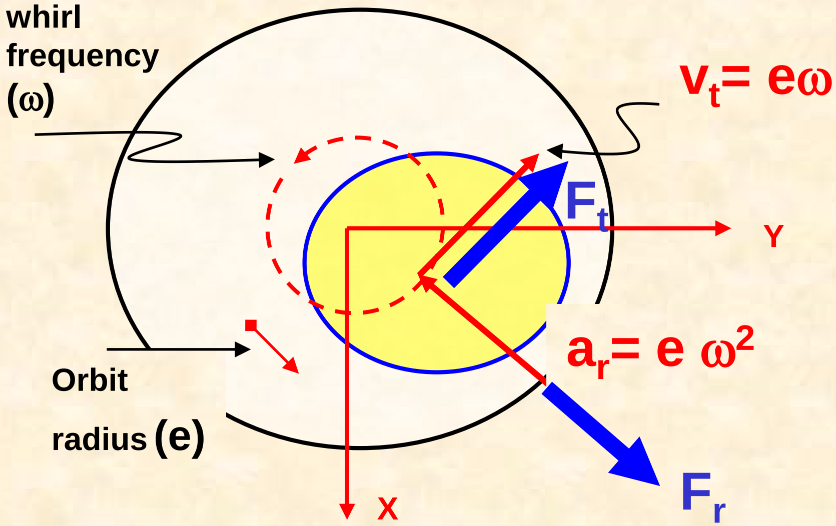
$$-\begin{Bmatrix} F_X \\ F_Y \end{Bmatrix} = \begin{bmatrix} C_{XX} & C_{XY} \\ C_{YX} & C_{YY} \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} + \begin{bmatrix} M_{XX} & M_{XY} \\ M_{XY} & M_{YY} \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix}$$



$$i = \frac{2\cos(-\varepsilon)}{\left(1-\varepsilon^2\right)^{1/2}}$$

SFD force coefficients
NL functions of static journal eccentricity e_s

Kinetics of whirl (circular) orbits



Journal center velocity with radial & tangential (v_r, v_t) components, and also acceleration (a_r, a_t)

For circular centered orbits, amplitude *e* is constant and whirl frequency = ω.

Circular centered orbit

$v_t = e\,\omega$; $v_r = 0$

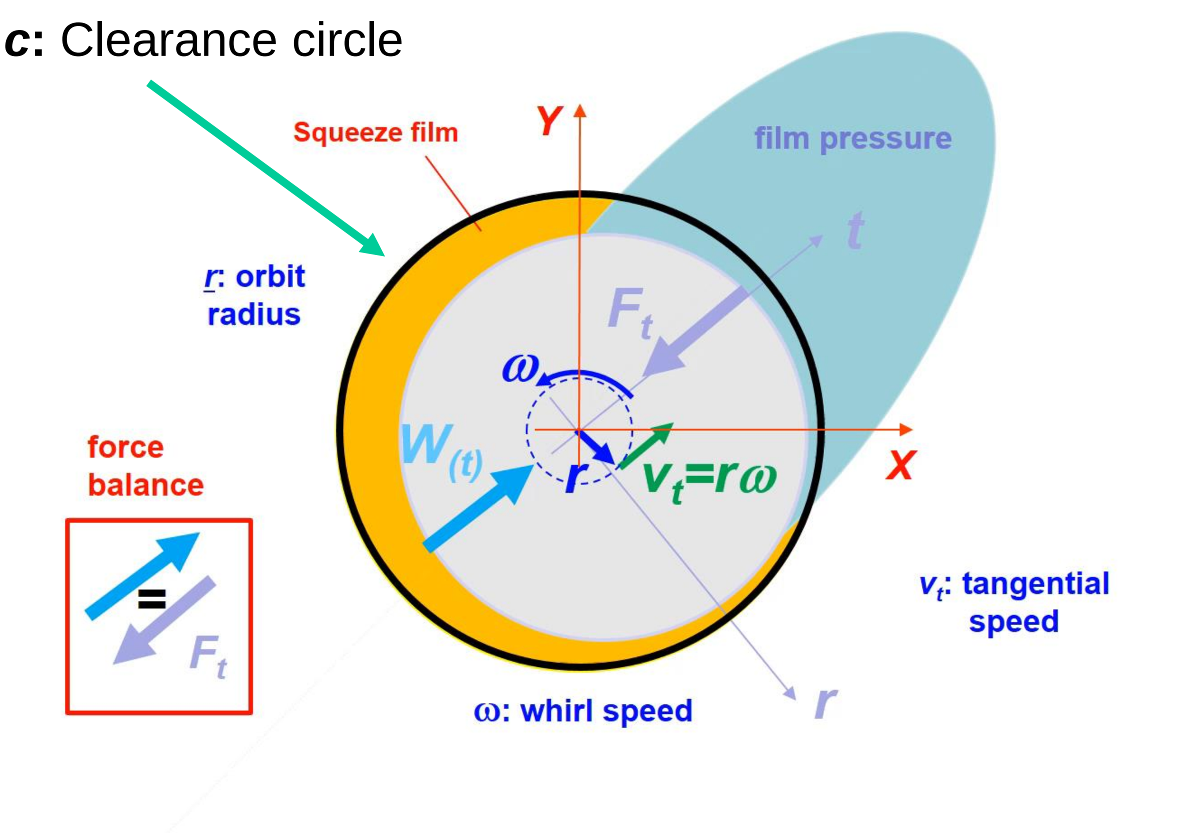
$a_r = -e\,\omega^2$; $a_t = 0$

SFD reaction forces:

$$F_r = -(C_{rt}\, v_t + M_{rr}\, a_r)$$

$$F_t = -(C_{tt}\, v_t + M_{tr}\, a_r)$$

SFD model: circular centered orbits



Circular centered orbit

$v_t = e\,\omega\,;\quad v_r = 0$

$a_r = -e\,\omega^2;\quad a_t = 0$

SFD reaction forces:

$$F_r = -\,(C_{rt}\,v_t + M_{rr}\,a_r)$$

$$F_t = -\,(C_{tt}\,v_t + M_{tr}\,a_r)$$

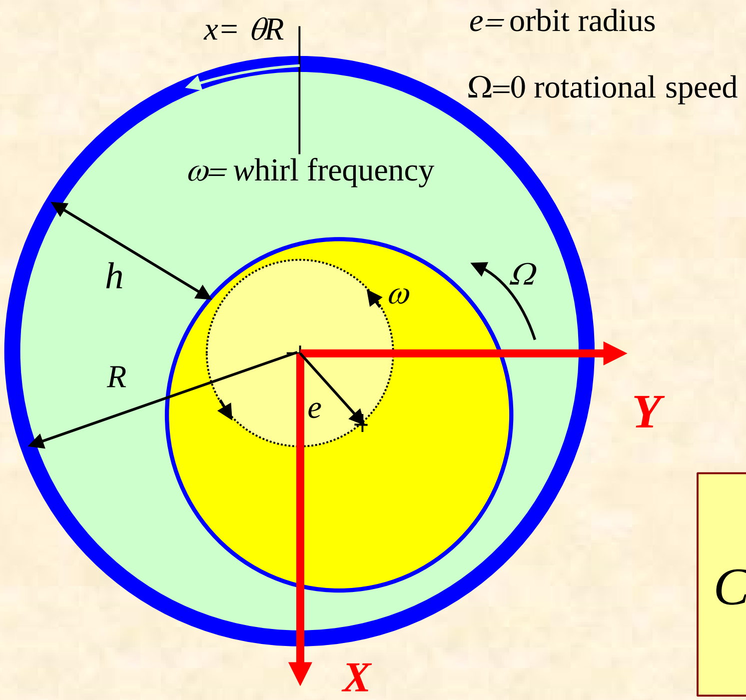
C: damping,
M: inertia force coefficients

Pressure is invariant in rotating frame. *P* follows -d*h*/d*t* rather than *h* (film)

SFDs DO NOT have a stiffness

Misnomer: $K_{rr} = \omega C_{rt}$

SFD model: small amplitude centered orbit

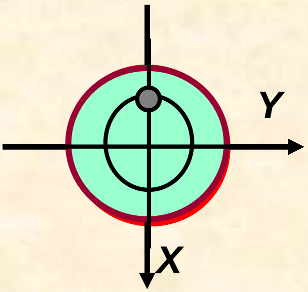


FULL FILM MODEL

Damping (C) & inertia (M) force coefficients by Reinhart & Lund (1975)

$$C_{XX} = C_{YY} = C_{tt} = 12\pi \frac{\mu R^3 L}{c^3}\left[1 - \frac{\tanh\left(L/D\right)}{\left(L/D\right)}\right]$$

$$M_{XX} = M_{YY} = M_{rr} = \pi \frac{\rho R^3 L}{c}\left[1 - \frac{\tanh\left(L/D\right)}{\left(L/D\right)}\right]$$



Damping ~ $(R/c)^3$, Inertia ~ R^3/c

SFD sealed vs open

FULL FILM MODEL

Open ends

$$C_{XX} = C_{YY} = C_{tt} = \tfrac{1}{2}\pi \frac{\mu D L^3}{c^3}$$

$$M_{XX} = M_{YY} = M_{rr} = \frac{\pi \rho D}{24}\left(\frac{L^3}{c}\right)$$

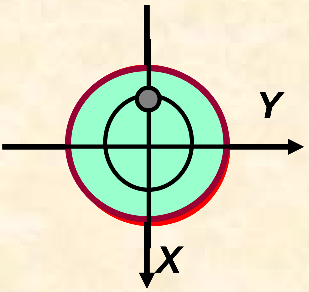
(fully) Sealed ends

$$C_{XX} = C_{YY} = C_{tt} = \pi \frac{12}{8} \frac{\mu D^3 L}{c^3}$$

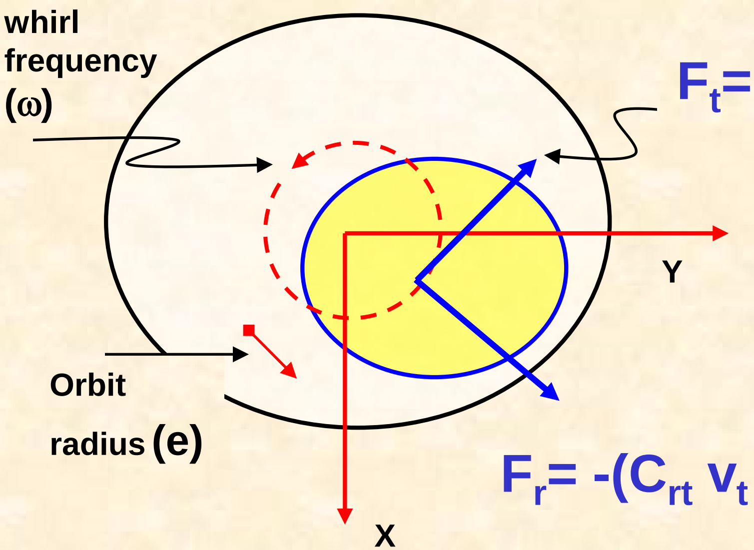
$$M_{XX} = M_{YY} = M_{rr} = \pi \frac{\rho D^3 L}{8c}$$

$$C_{tt} \frac{sealed}{open} = M_{rr} \frac{sealed}{open} = \left(\frac{D}{L}\right)^2$$

Increase in damping (and inertia) is large!
For (*L/D*)=0.2=1/5, increase is **25** fold



SFD model: circular centered orbits



$F_t = -(C_{tt}\, v_t + M_{tr}\, a_r)$

$F_r = -(C_{rt}\, v_t + M_{rr}\, a_r)$

$v_t = e\,\omega$; $v_r = 0$

$a_r = -e\,\omega^2$; $a_t = 0$

π FILM MODEL

Damping & inertia force coefficients

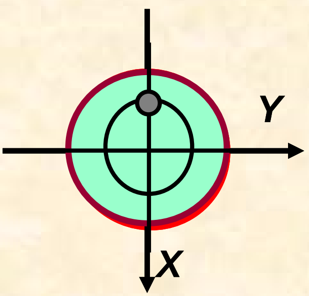
Short length open ends SFD

(PI film model)

$$C_{tt} = \frac{\pi\,\mu\,D}{4\left(1-\varepsilon^2\right)^{3/2}}\left(\frac{L}{c}\right)^3 ;\; C_{rt} = \frac{\mu\,\varepsilon\,D}{\left(1-\varepsilon^2\right)^2}\left(\frac{L}{c}\right)^3$$

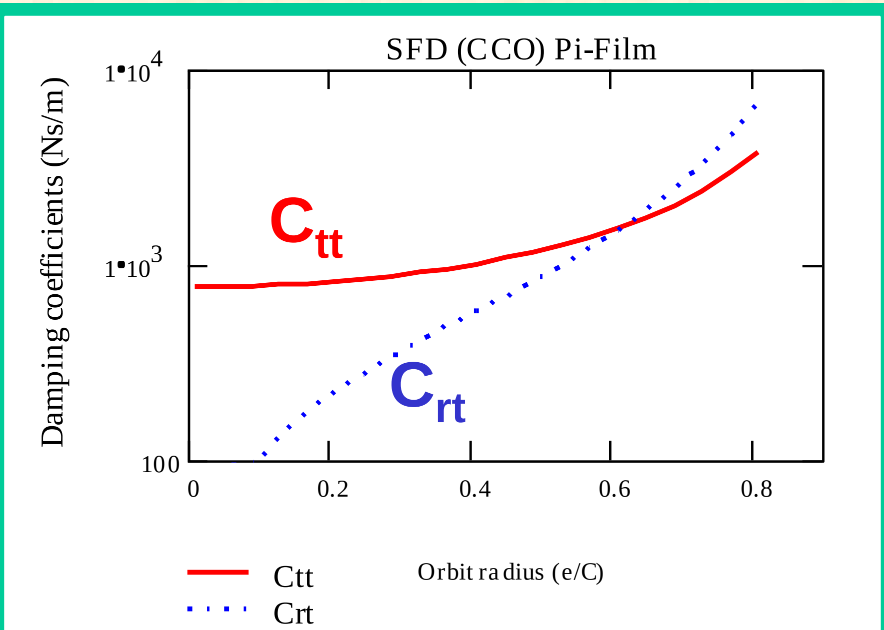
$$M_{rr} = \frac{\pi\,\rho\,D}{24}\left(\frac{L^3}{c}\right)\left[1-2\left(1-\varepsilon^2\right)^{1/2}\right]\left\{\frac{\left(1-\varepsilon^2\right)^{1/2}-1}{\varepsilon^2\left(1-\varepsilon^2\right)^{1/2}}\right\};$$

$$M_{tr} = -\frac{27}{140\,\varepsilon}\,\rho\,D\left(\frac{L^3}{c}\right)\left[2+\frac{1}{\varepsilon}\ln\left(\frac{1-\varepsilon}{1+\varepsilon}\right)\right]$$



Damping ~ $(L/c)^3$, Inertia ~ L^3/c

SFD model: circular centered orbits



$L = 0.05$ m

$D = 0.2$ m

$C = 8 \cdot 10^{-4}$

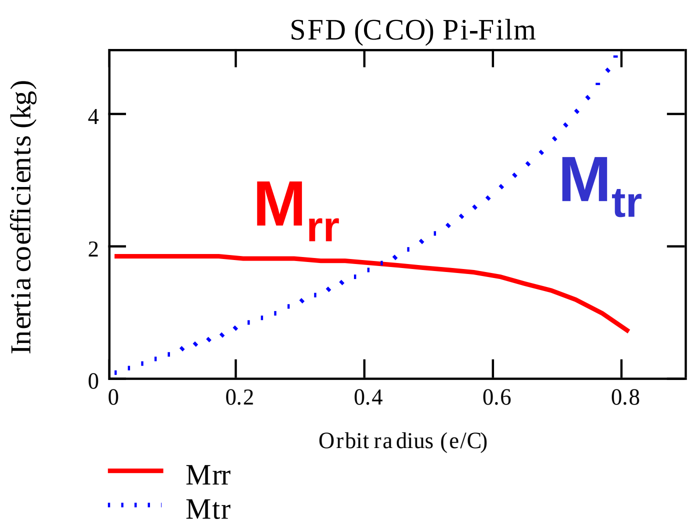
$\mu = 0.02$ Pa.s

$\frac{L}{D} = 0.25$ $\frac{D}{C} = 250$

Short length open ends SFD –π film model

$$F_r = -(C_{rt}\, v_t + M_{rr}\, a_r)$$

$$F_t = -(C_{tt}\, v_t + M_{tr}\, a_r)$$



$M_j = 12.252$ kg - steel journal

$M_o = 0.022$ kg - lubricant

Nonlinear force coefficients,

Large damping,

Large inertia for

$Re_s = \rho \omega^2 c/\mu > 10$

SFD model & test data: circular centered orbit

Damping coefficient

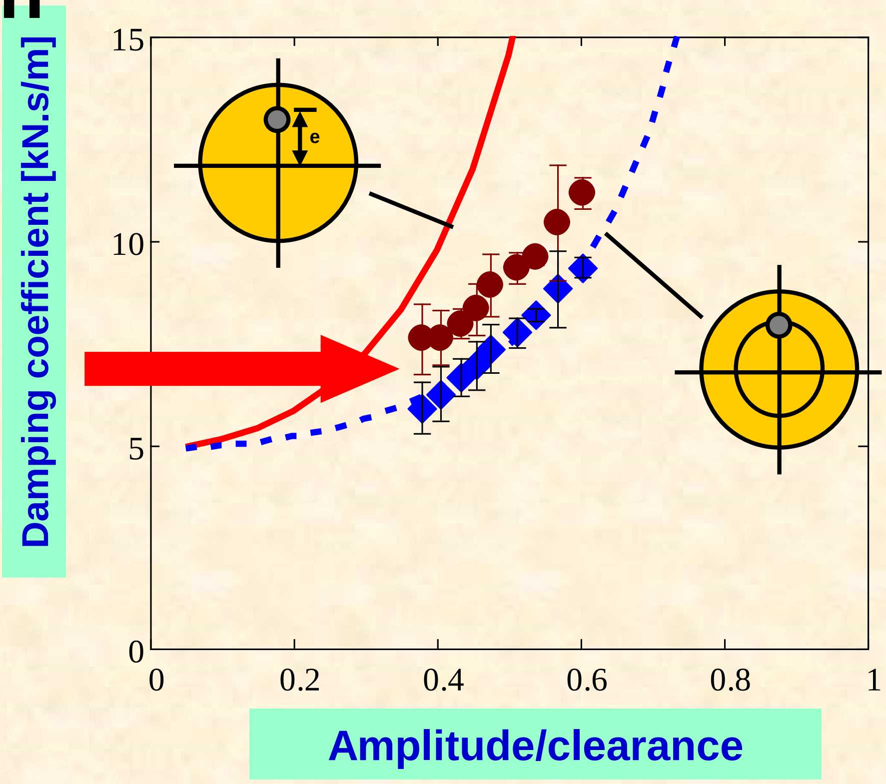
GT 2009-50175

Circular centered orbits,

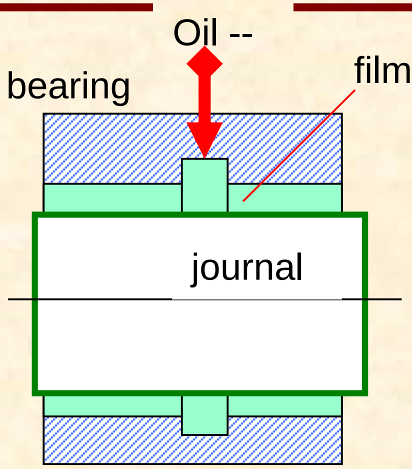
NO lubricant cavitation

L=1 inch
D=5 inch
C=5 mil
ISO VG2 oil

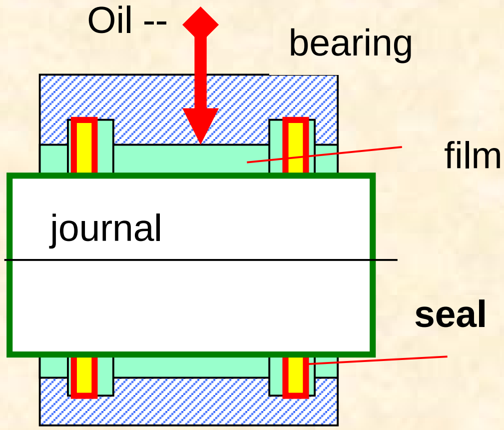
Nonlinear force coefficient:
depends on amplitude of motion (*e/c*)



SFD feed groove and exit grooves



SFD with feed groove



SFD with end grooves and seals

Feed holes with small diameter (high flow) resistance or with check valves used to prevent back flow and distortions in dynamic film pressures

Feed & discharge grooves

Interact with film flow, develop large dynamic film pressures, Induce inertia force coefficients even in small clearance (*c*) SFDs

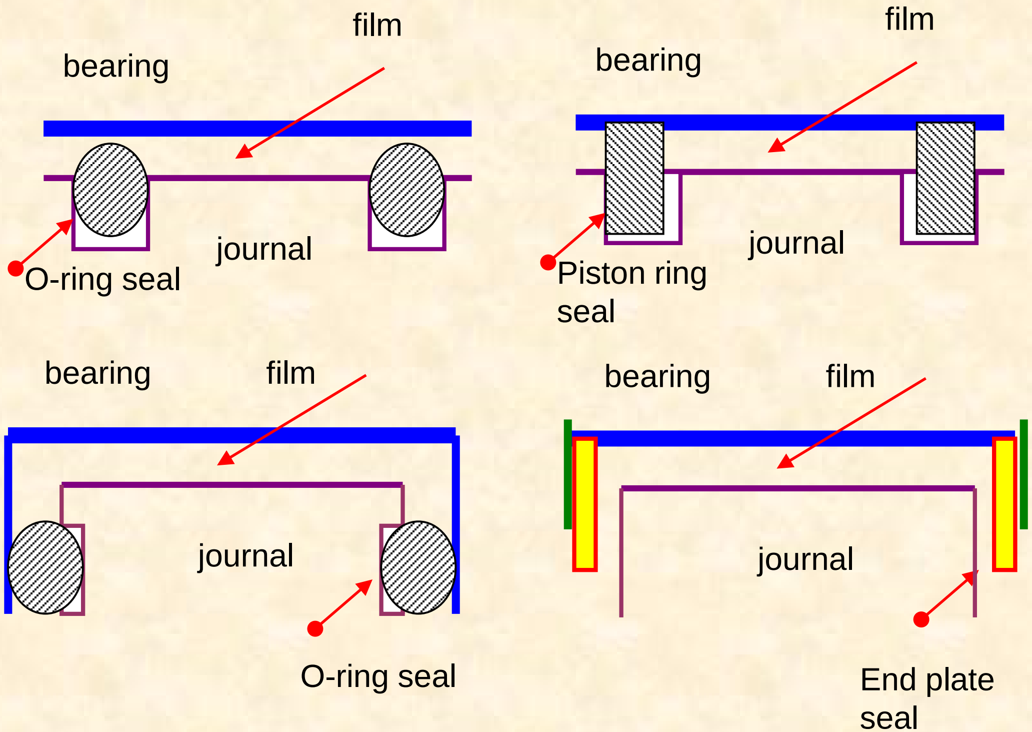
Too shallow grooves: increase damping (d_g/c)<10

Too deep grooves: increase added mass (d_g/c)>10

Types of end seals for SFDs

Reduce thru flow and increase damping. Most seal types cannot prevent air ingestion

Industrial applications use O-rings, while jet engines implement piston rings.



O-ring issues:

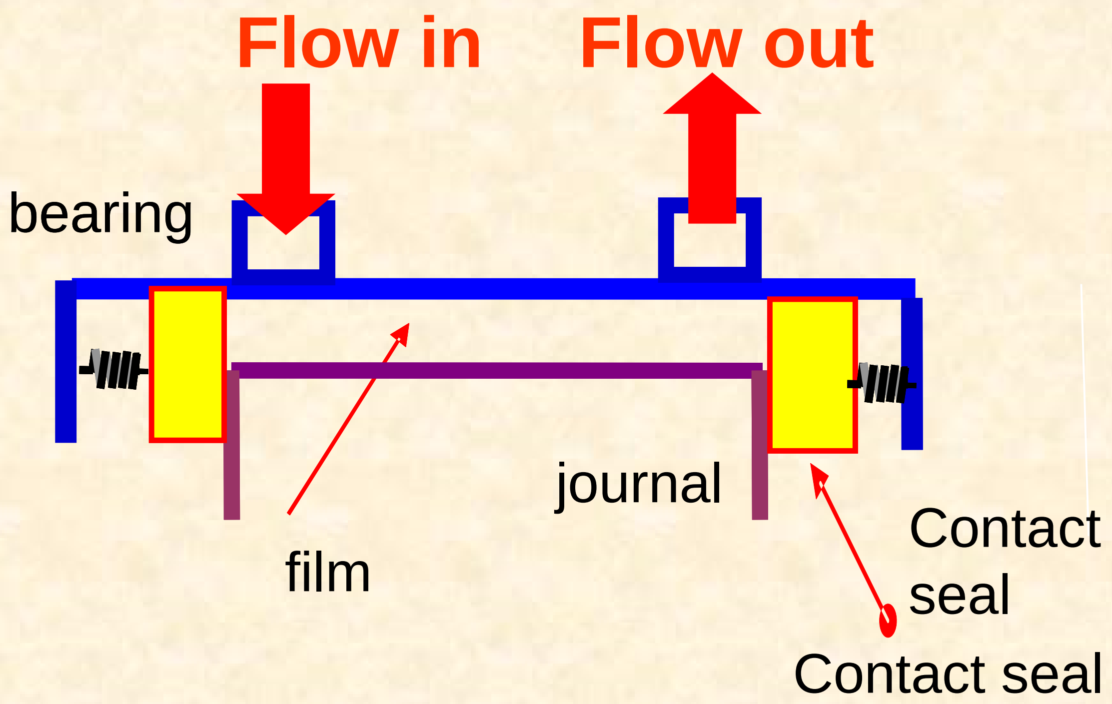
Low weight (replace squirrel cage),
Special groove machining,
Material compatibility

Piston ring issues:

Cocking and locking
Splits – leak too much

Design is highly empirical, except for end plate seals

SFD + mechanical seal



SFD with mechanical seal

Spring loaded contacting face seal (metal-metal). Seal keeps lubricant for long periods of time.

No side leakage allowed.

Forced performance complicated by dry-friction at contact area.

SFD air ingestion: CONCLUSIONS

1. Irregular air fingering occurs for most test conditions

2. Air entrainment is most persistent at high squeeze velocity: whirl amplitude x whirl frequency. A modest increase in feed pressure does alleviate issue.

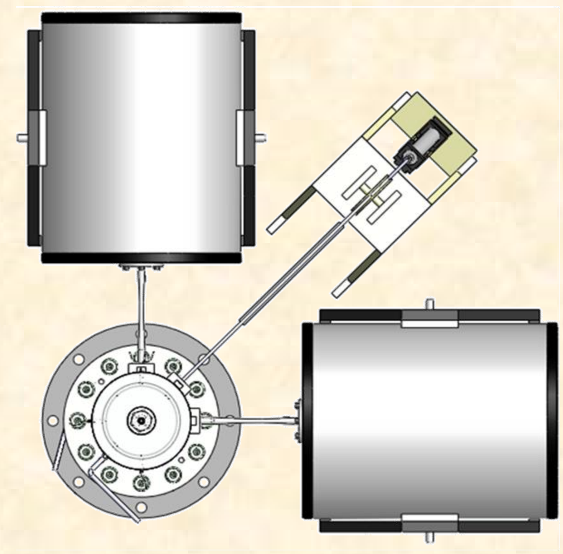
3. Tangential (damping) forces remain uniform or decrease as whirl frequency increases, even for a large supply pressure.

Watch digital videos of flow field and dynamic pressure at

http://rotorlab.tamu.edu

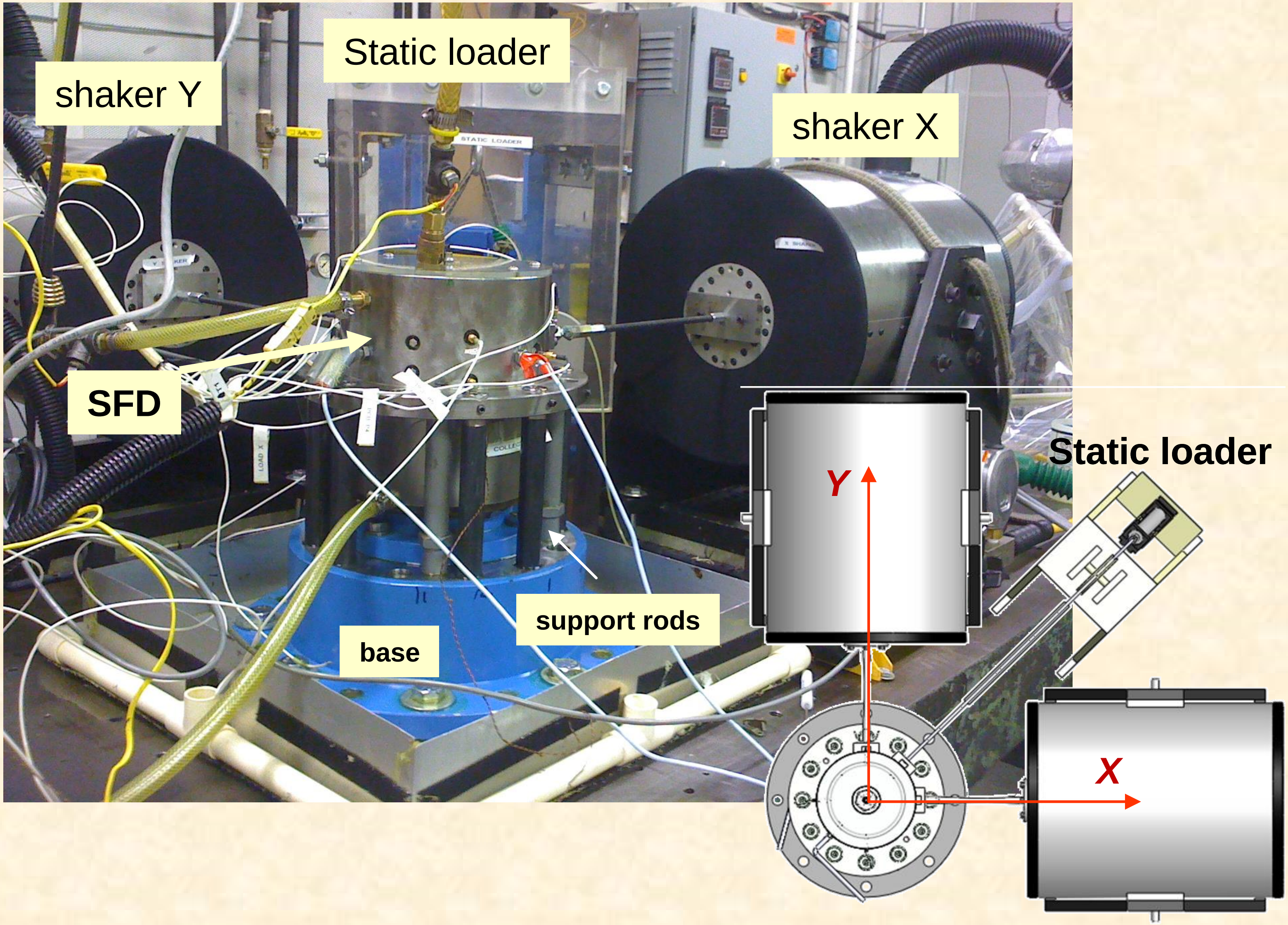
NSF funded research (1996-99)

Identification of SFD force coefficients for two SFDs:
open ends & sealed ends



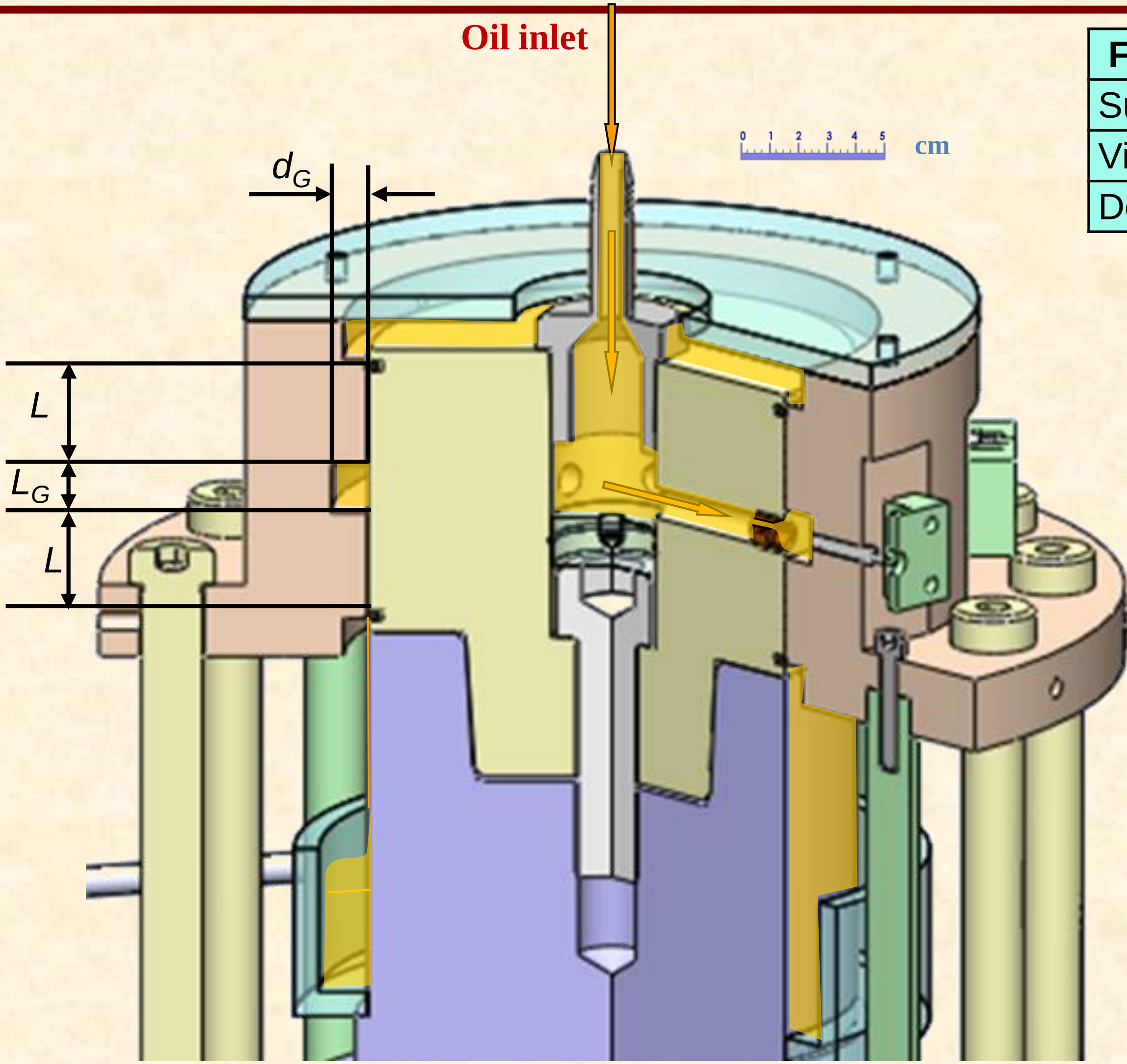
Sponsor: Pratt & Whitney Engines 2008-2018

Test rig photograph



Lubricant flow path

ISO VG 2 oil



Fluid properties ISO VG2	
Supply temperature (T_{in})	25 °C
Viscosity	2.96 c-Poise
Density	785 kg/m^3

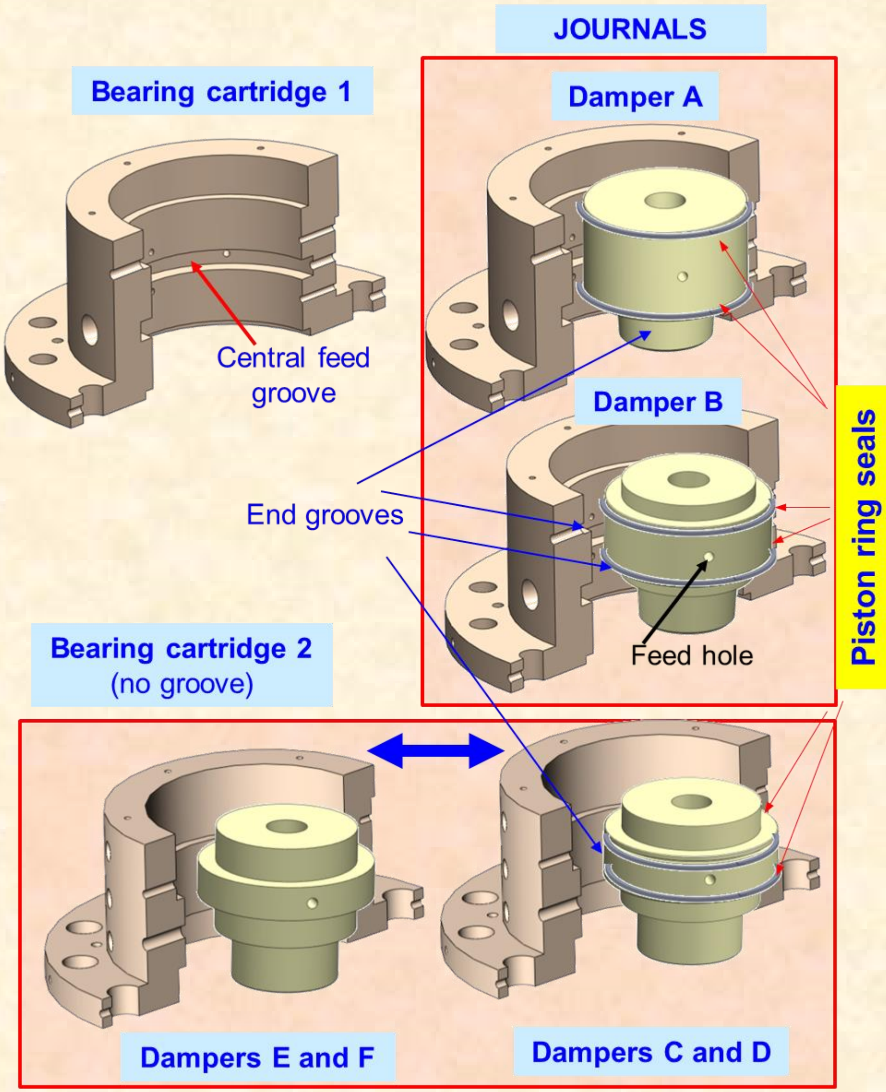
	Long journal (A)	**Short journal (B)**
Land length (L)	25.4mm	12.7mm
Land clearance (c)	0.14 mm	0.13 mm

Journal diameter (D)	12.7cm
Groove length (L_G)	12.7 mm
Groove depth (d_G)	9.52 mm

SFD test configurations

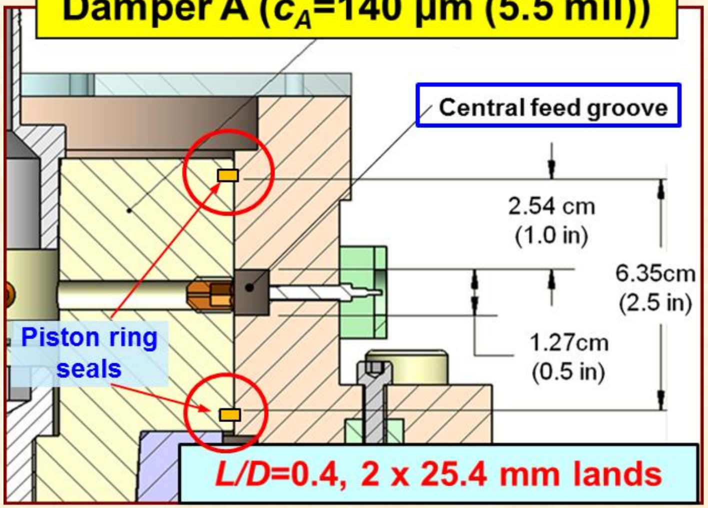
Configuration	Land Length, Central Groove	End Grooves	Radial Clearance
A	2 X L=25.4 mm film lands - **Central groove**	Yes	c_{A-1}=141 µm c_{A-2}=251 µm
* B	2 X L=12.7 mm film lands - **Central groove**		c_B=138 µm
* C	L=25.4 mm film land - **No** feed groove	Yes	c_C=130 µm
* D *			c_D=254 µm
E	L=25.4 mm film land - **No** feed groove	No	c_E=122 µm
F			c_F=267 µm

*** SFD: open ends and sealed with piston ring seals**

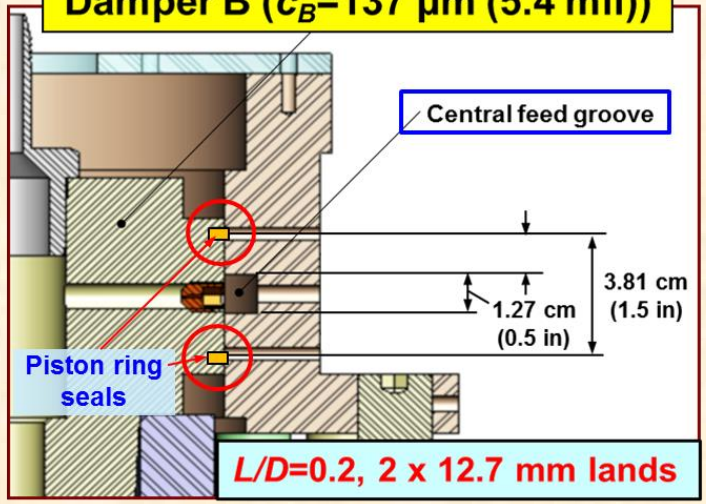


Multiple-year test program (2008- 2018)

Damper A (c_A=140 μm (5.5 mil))



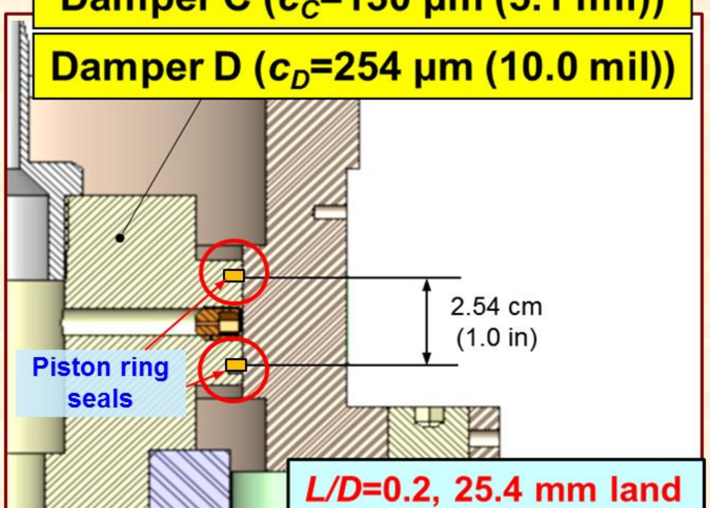
Damper B (c_B=137 μm (5.4 mil))



Objective: Optimize SFD influence on rotor dynamics.

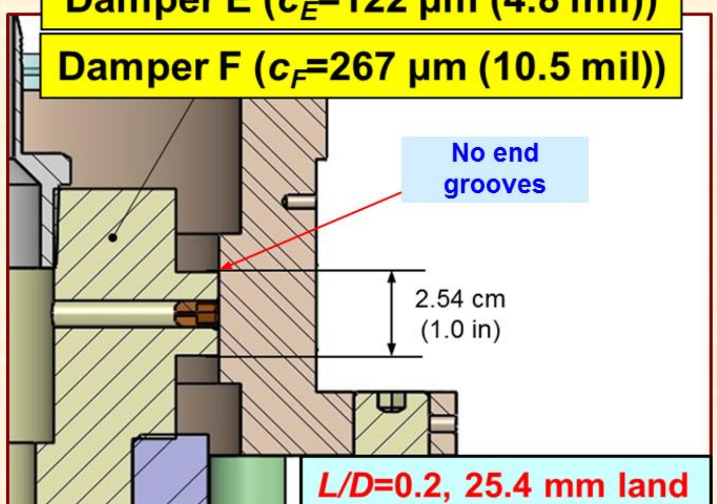
Damper C (c_C=130 μm (5.1 mil))

Damper D (c_D=254 μm (10.0 mil))

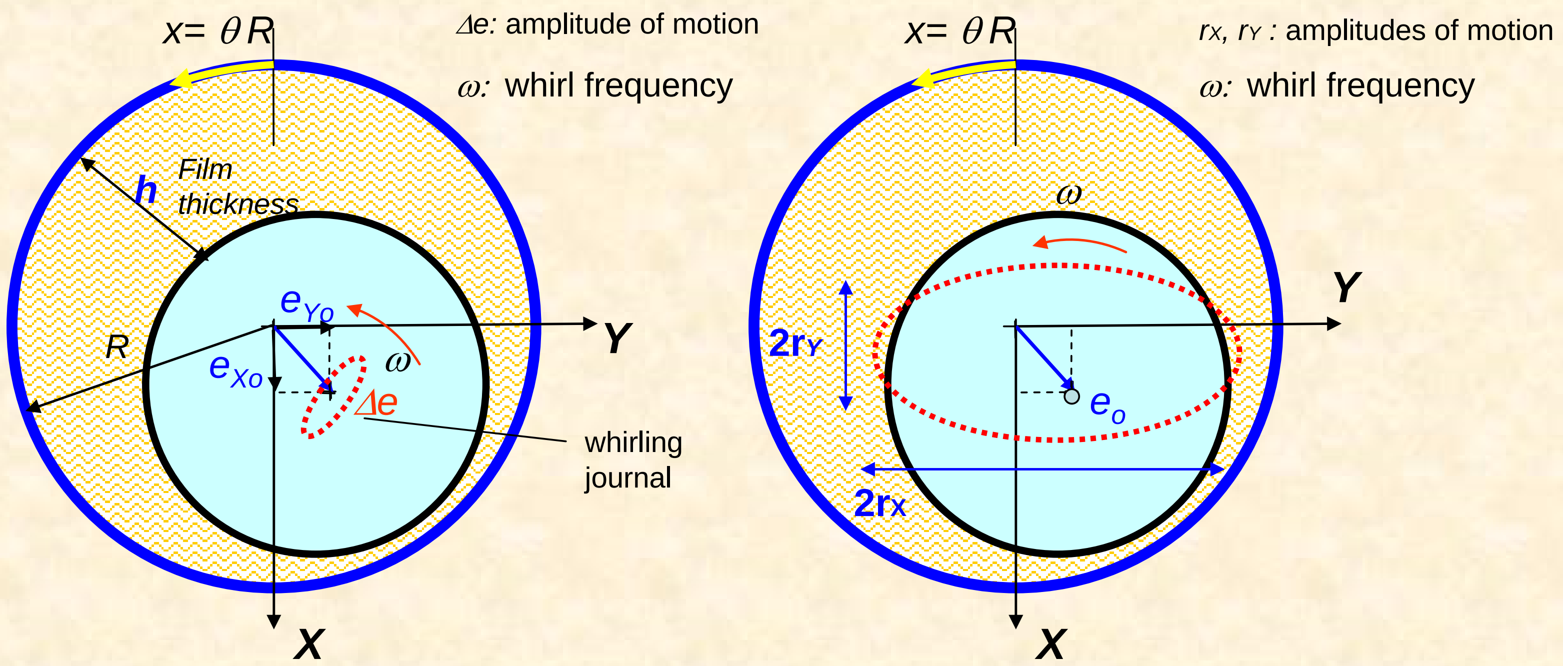


Damper E (c_E=122 μm (4.8 mil))

Damper F (c_F=267 μm (10.5 mil))



Types of induced motions



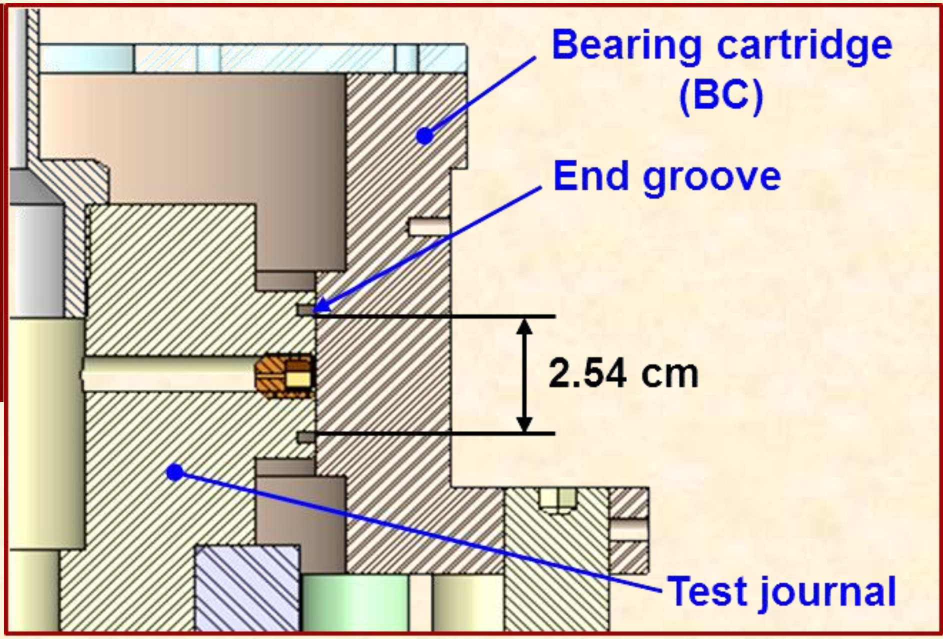
(a) small amplitude journal motions (b) large amplitude journal motions

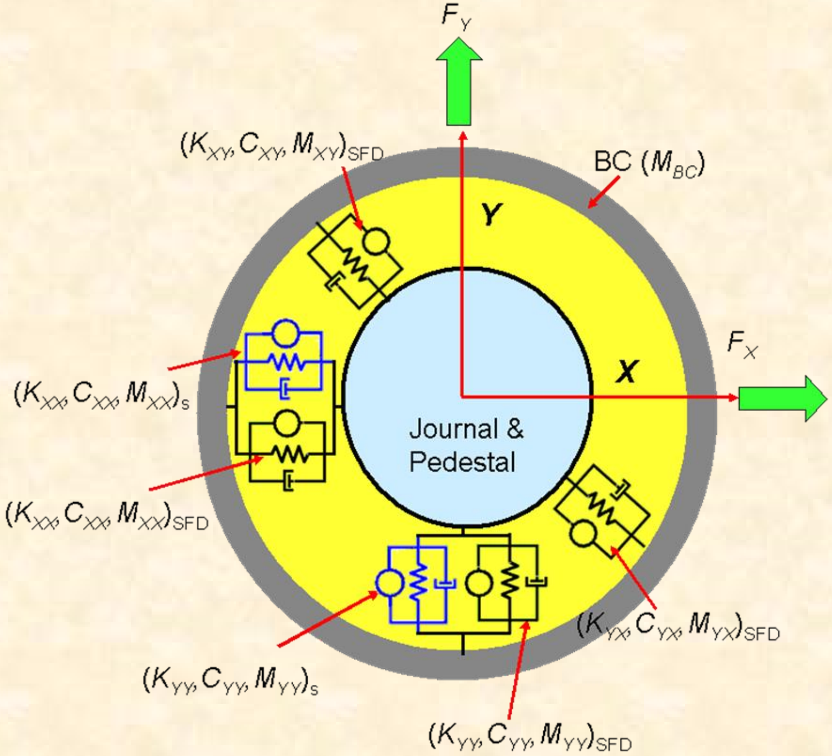
Applications:

K,C, M (force coefficients)
RBS stability analysis

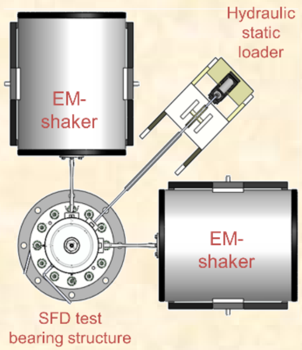
Fx, Fy (reaction forces)
RBS imbalance response

Identification of SFD force coefficients

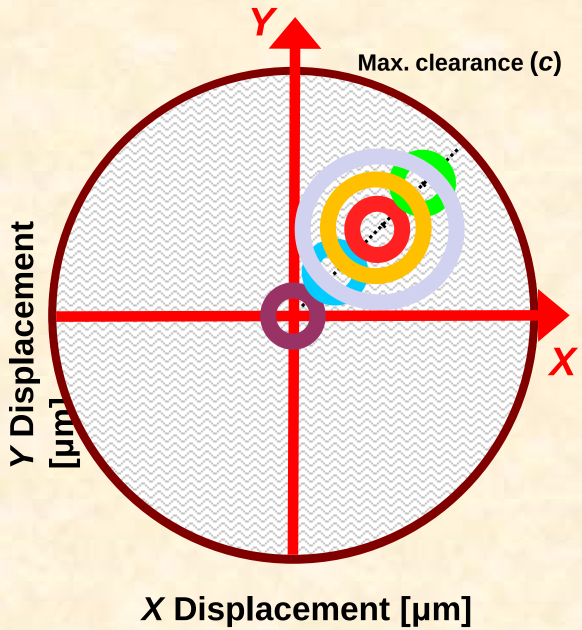


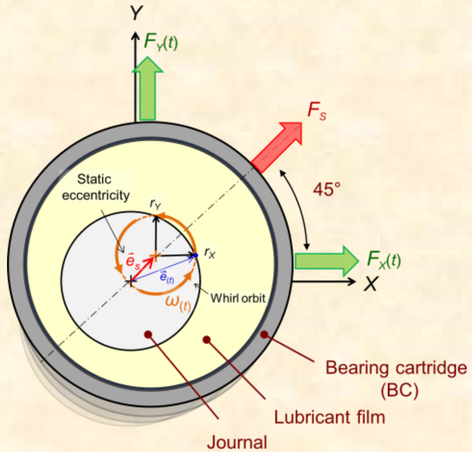


Test procedure



Evaluate SFD force coefficients from whirl orbits: amplitude (r) grows with offset or static eccentricity (e_s) – 45° away.





(1) Apply loads → record BC motions

Shakers apply forces

$$\mathbf{F}^1 = \mathrm{Re}\left(\begin{bmatrix} F_X{}^1 \\ iF_Y{}^1 \end{bmatrix} e^{i\omega t}\right)$$

$$\mathbf{F}^2 = \mathrm{Re}\left(\begin{bmatrix} F_X{}^2 \\ -iF_Y{}^2 \end{bmatrix} e^{i\omega t}\right)$$

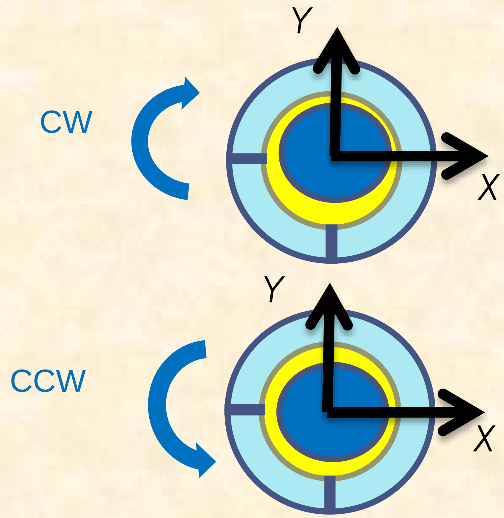
Record BC displacements and accelerations

$$\mathbf{z}^1 = \begin{bmatrix} x^1_{(t)} \\ y^1_{(t)} \end{bmatrix} = \begin{bmatrix} X^1 \\ Y^1 \end{bmatrix} e^{i\omega t}$$

$$\mathbf{a}^1$$

$$\mathbf{z}^2 = \begin{bmatrix} x^2_{(t)} \\ y^2_{(t)} \end{bmatrix} = \begin{bmatrix} X^2 \\ Y^2 \end{bmatrix} e^{i\omega t}$$

$$\mathbf{a}^2$$



Load $\mathbf{F}_{(t)}$, displacement $\mathbf{z}_{(t)}$ and acceleration $\mathbf{a}_{(t)}$ recorded at each frequency

EOM: Frequency Domain

$$[\mathbf{K_L} + i\omega\mathbf{C_L} - \omega^2\mathbf{M_L}]\bar{\mathbf{z}} = \bar{\mathbf{F}} - M_{BC}\bar{\mathbf{a}}$$

$$\rightarrow \mathbf{H_L z}$$

Unknown Parameters:

$\mathbf{K_L}$, $\mathbf{C_L}$, $\mathbf{M_L}$

Identification of parameters

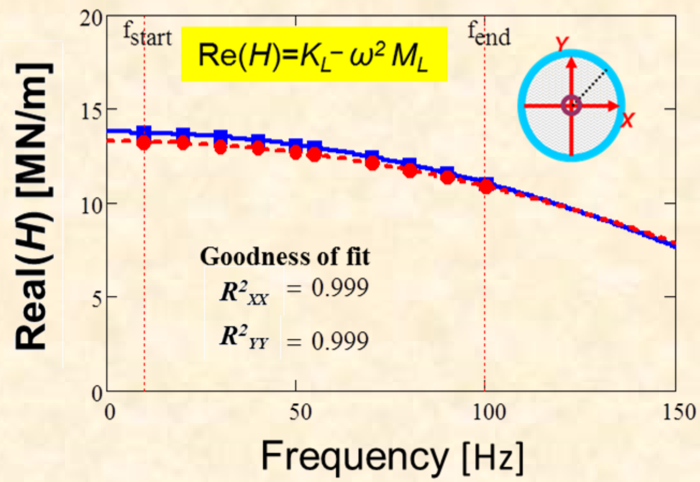
Step 2 : Transform to frequency domain and curve fit H_L's

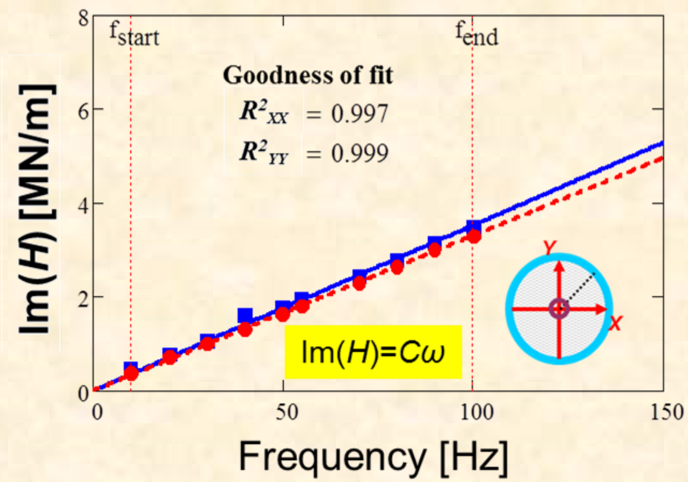
Complex dynamic stiffness

$\mathrm{Re}\left([\bar{\mathbf{F}} - M_{BC}\bar{\mathbf{a}}]\bar{\mathbf{z}}^{-1}\right) \rightarrow \mathbf{K}_L - \omega^2 \mathbf{M}_L$

$\mathrm{Im}\left([\bar{\mathbf{F}} - M_{BC}\bar{\mathbf{a}}]\bar{\mathbf{z}}^{-1}\right) \rightarrow \mathbf{C}_L \omega$

r/c_A=0.2





Physical model Re(H_{XX}) → K-ω^2M and Im(H_{XX})→$C\omega$ agree well with experimental data.
Damping C is constant over frequency range

Test data Test data
Model Model

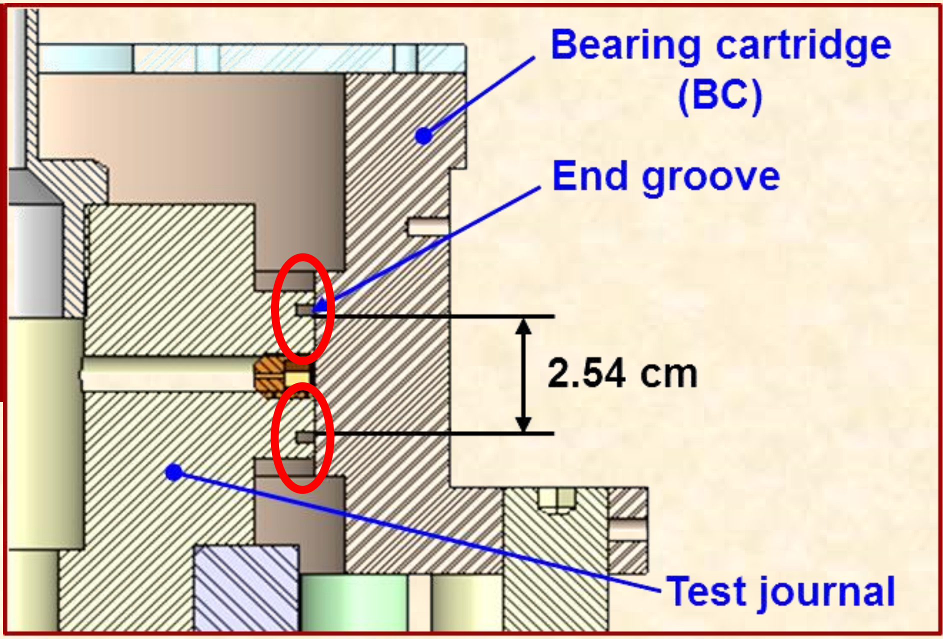
SFD coefficients

$$(K, C, M)_{SFD} = (K, C, M)_L - (K, C, M)_S$$

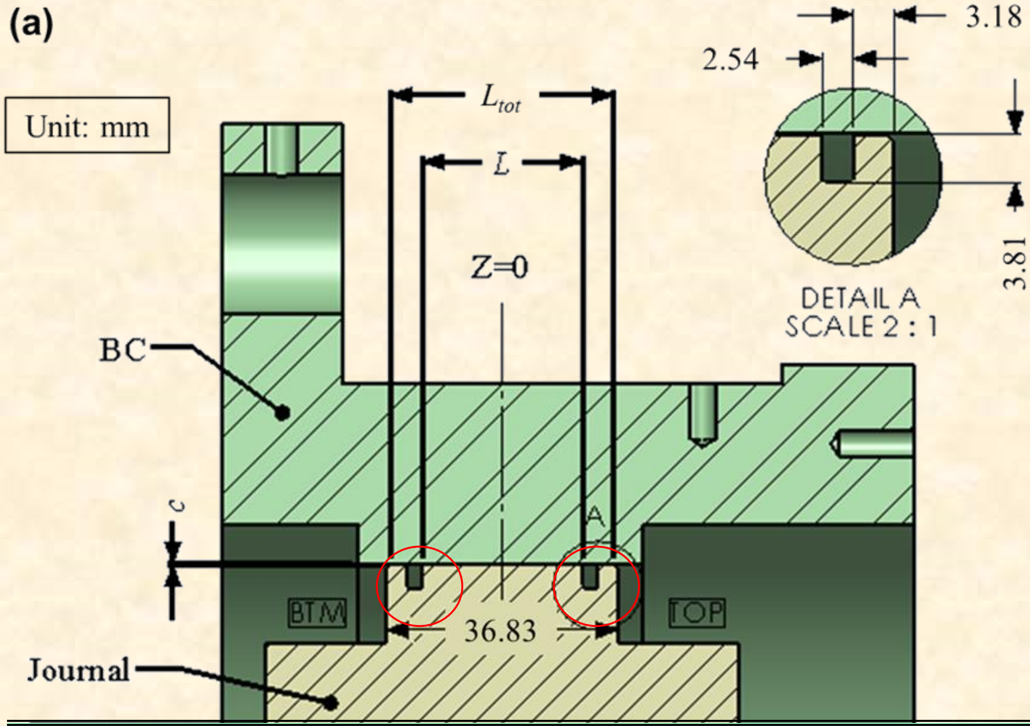
SFD — Test system (lubricated) — Dry structure

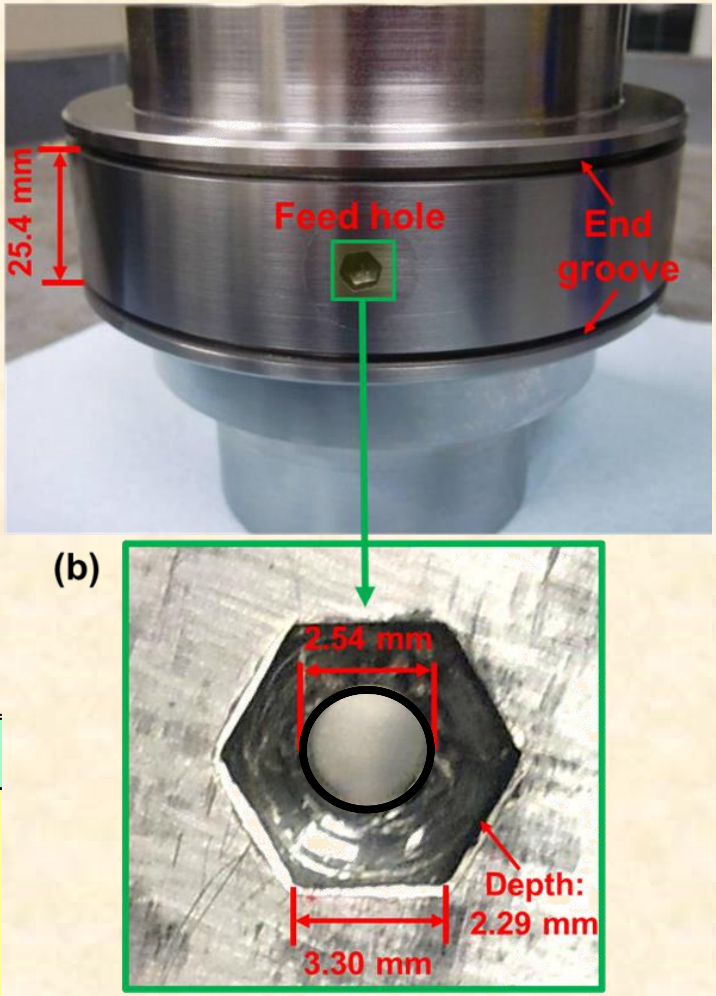
Force coefficients for two open ends SFDs

1 inch land (*L/D*=0.20)
***c*=small (5 mil) & large (10 mil)**



SFD test bearing and film geometry



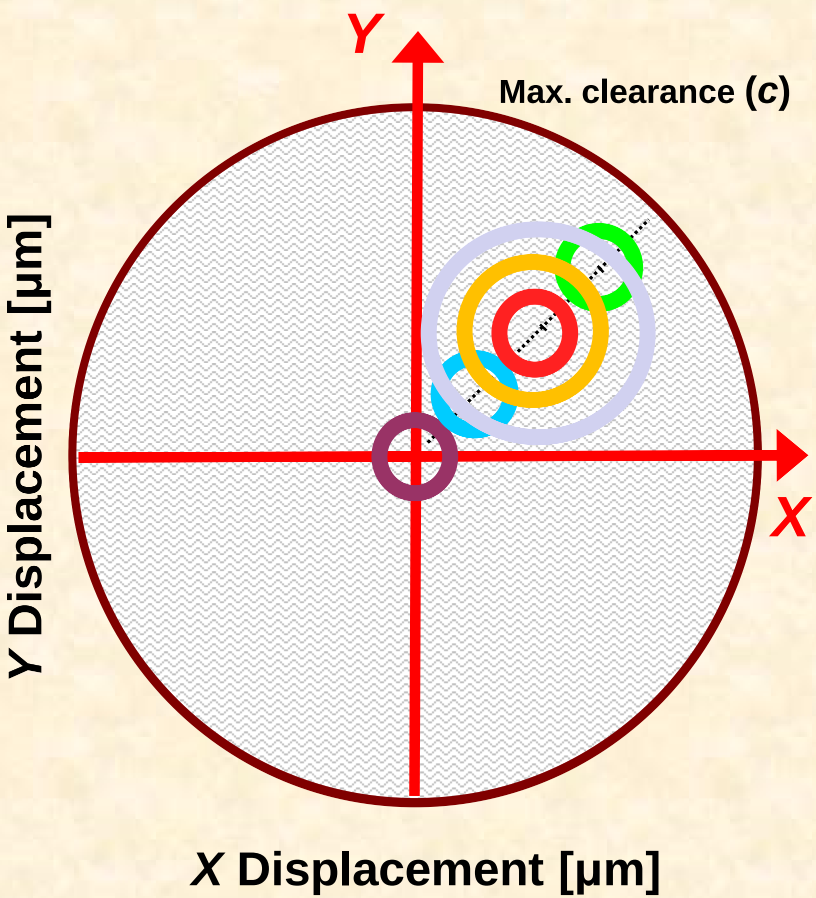


Geometry of SFD	A	B
Film land length, L (mm)	25.4	
Radial clearance, c_A , c_B (mm)	0.129	0.254
End grooves: depth × width (mm)	3.8 × 2.5	
Total wetted length, L_{tot} (mm)	36.8	

Tests conducted

Excitation frequency 10 – 100 Hz

Evaluate SFD force coefficients from circular orbits: amplitude (*r*) grows. with offset or static eccentricity (e_s) – 45° away from *X*-*Y* axes.



Operating condition	Damper A 0.129 mm	Damper B 0.254 mm
Whirl amplitude *r* (µm)	6.4 - 76	38 - 190
Static eccentricity e_s (µm)	0 - 63	0 - 190
Max. squeeze film Reynolds No. (Re_s)	8.4	11.9

Squeeze film Reynolds #

$$Re_S = \frac{\rho \omega c^2}{\mu}$$

Normalization of force coefficients

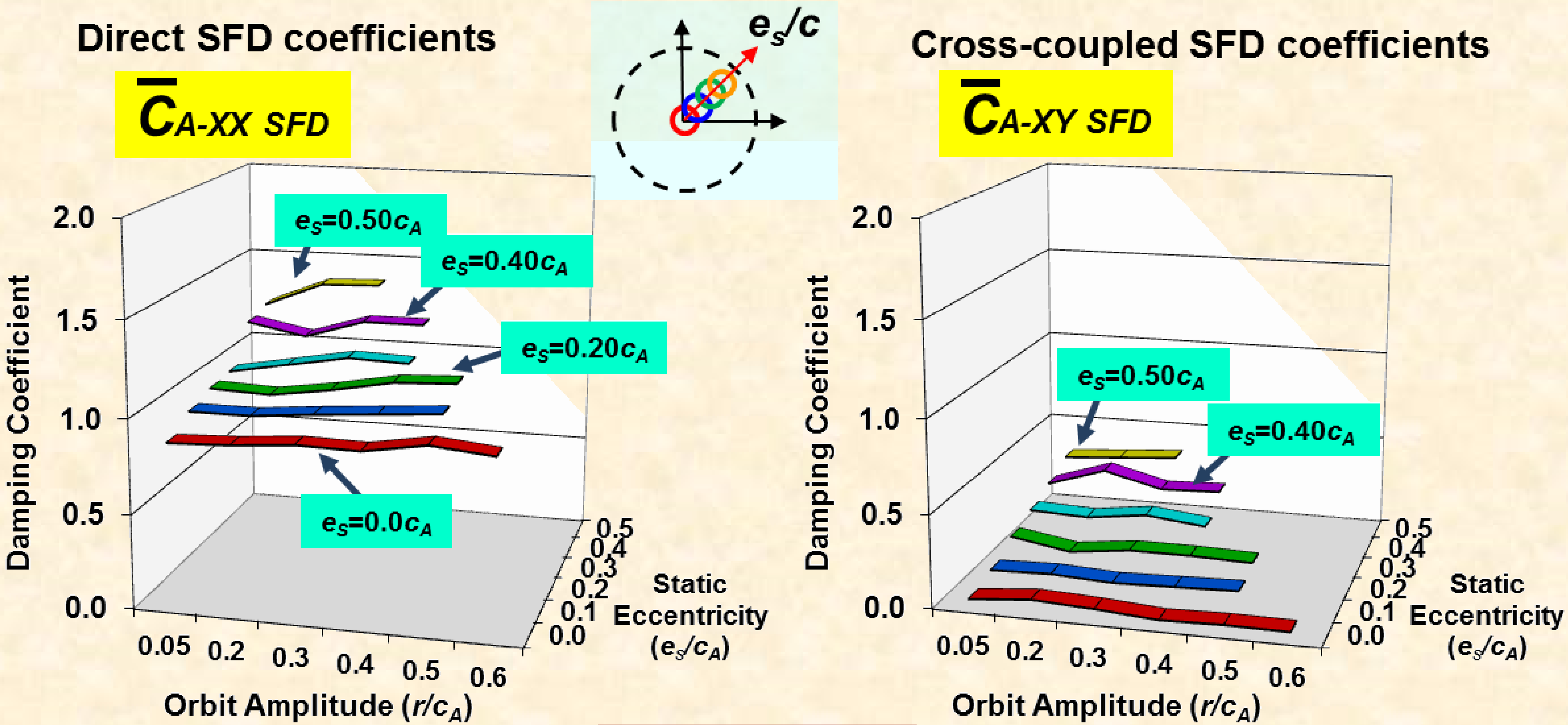
Force coefficients normalized to magnitudes from classical formulas (prior slide):

$$\bar{C} = C / C^{*} \qquad \bar{M} = M / M^{*}$$

Damper A c_A = 0.129 mm	C^*_A = 6.01 kN.s/m, M^*_A = 2.69 kg
Damper B c_B = 0.254 mm	C^*_B = 0.95 kN.s/m, M^*_B = 1.37 kg

$\bar{C}$ **~1** & $\bar{M}$ **~1** **denote one to one agreement with predictive formulas.**

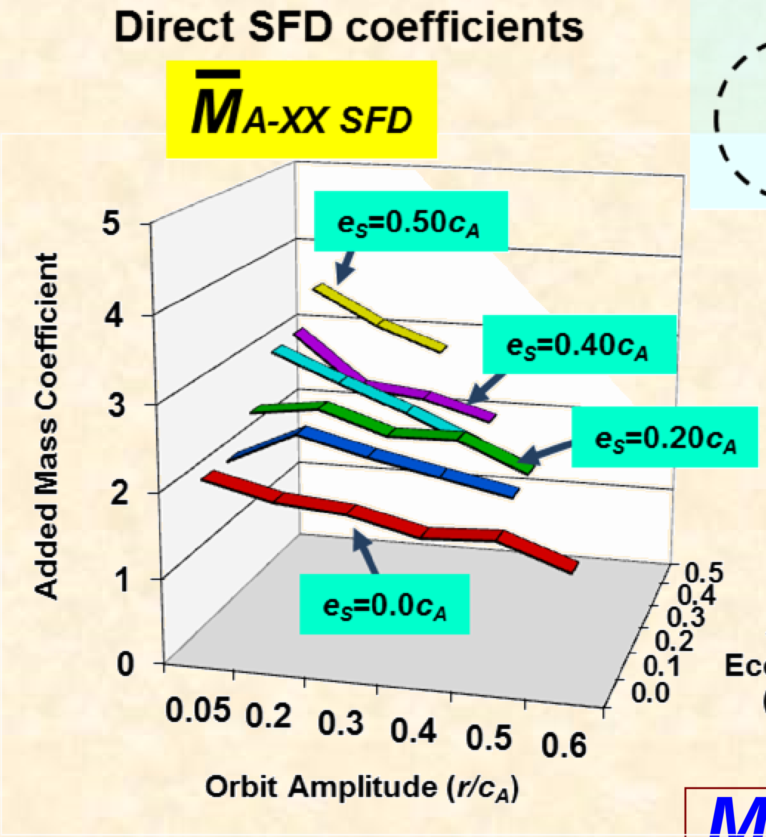
Damper A (c_A = 129 μm) – damping coeffs.

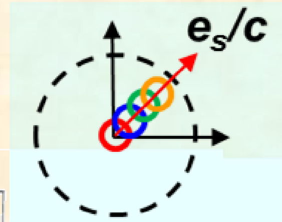


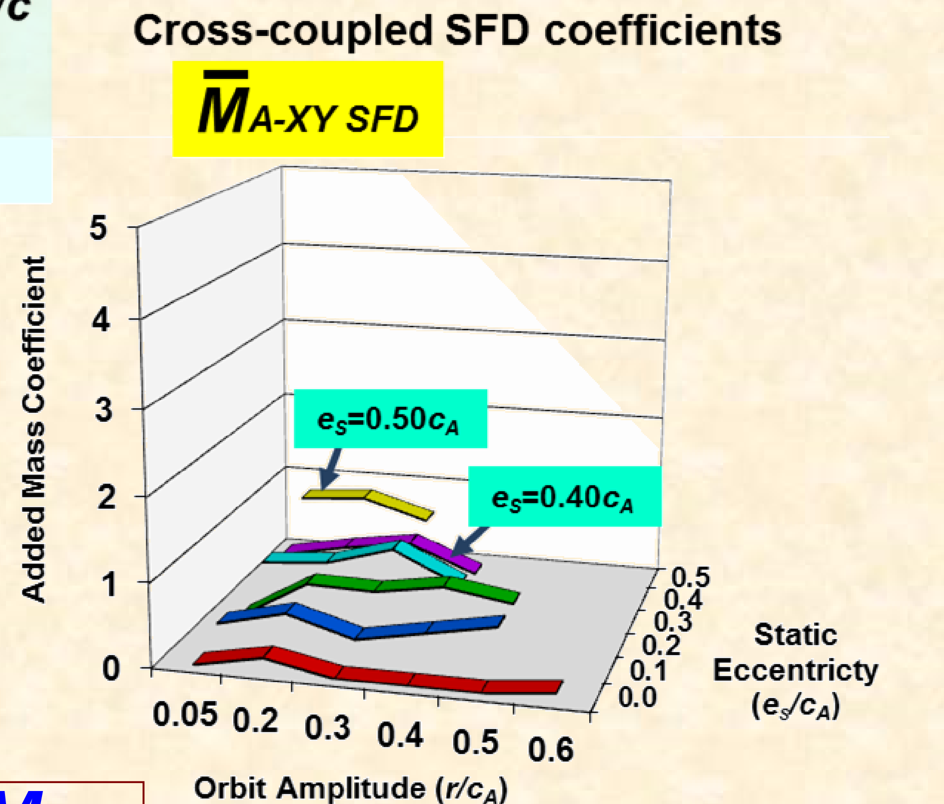
$C_{XX} \sim C_{YY}$

Findings: Damping coefficients **increase** with increasing **orbit amplitude** and **static eccentricity**. At r/c_A ≤0.2, $\overline{C}_{A-xx}$ ~ **0.85** denotes L_{eff} is too long.

Damper A (c_A = 129 μm) added mass coeffs.







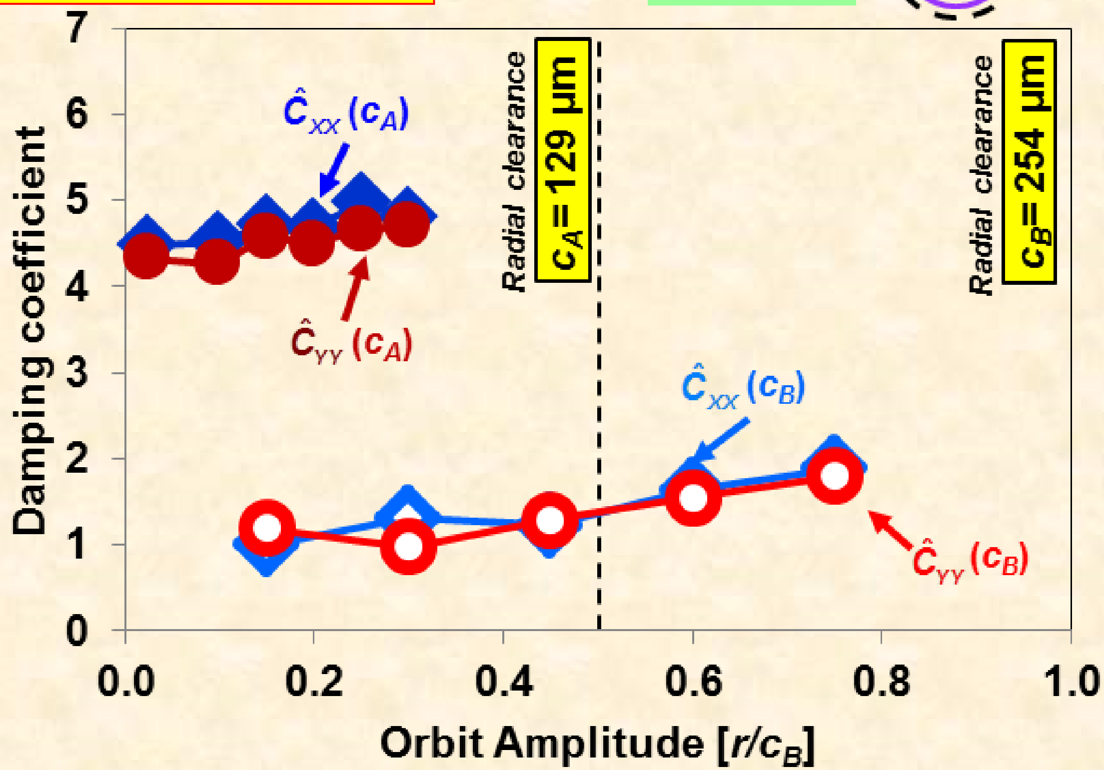
$M_{XX} \sim M_{YY}$

Findings: Added mass coefficients **increase** with increasing **static eccentricity**; but **decrease** with increasing **orbit amplitude**. **Theory under predicts inertia coefficient, even for small amplitude motions.**

Compare damping coeffs. of two dampers

$$\hat{C} = C \Big/ C_{B(r/c_B=0.15)}$$

$e_S/c=0$



Damper A	Damper B
c_A=0.129 mm	c_B=0.254 mm

Recall → $C \sim \mu \left(\frac{1}{c}\right)^3$

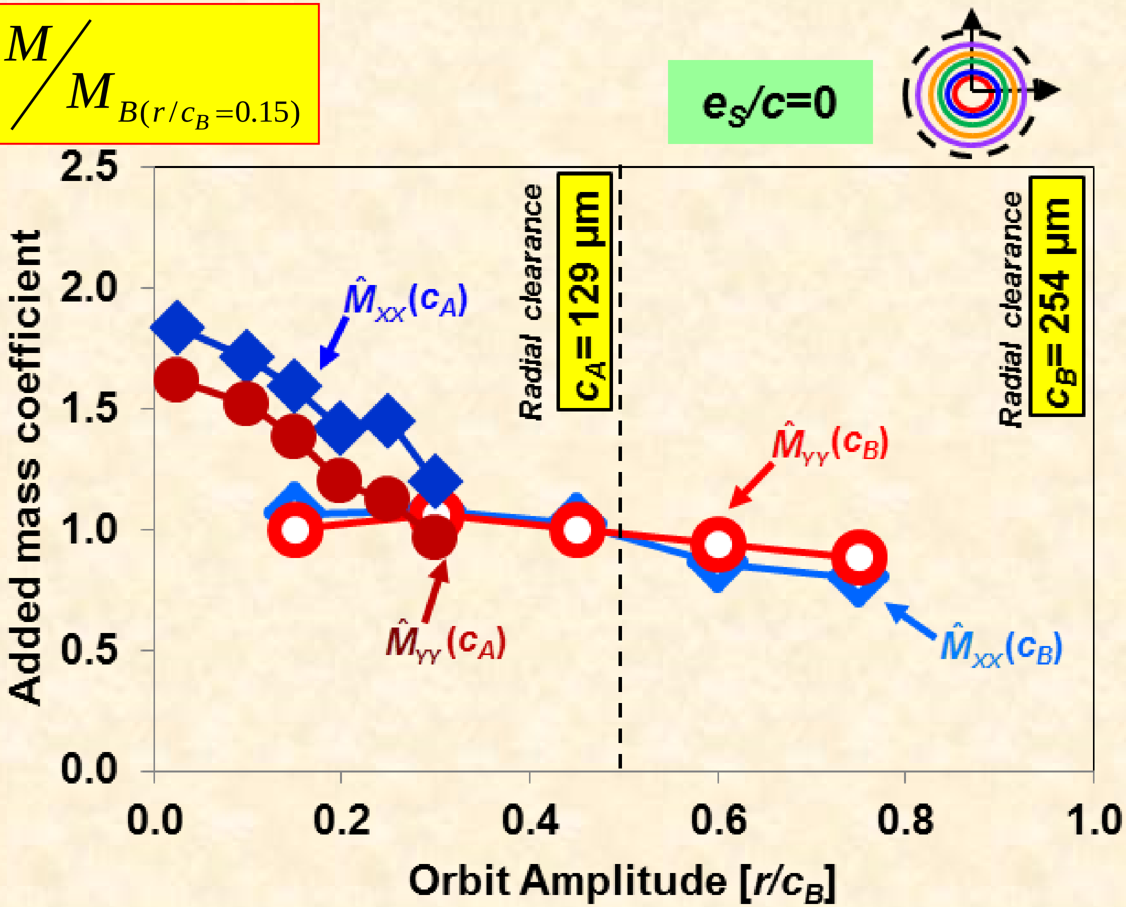
$$\left(\frac{c_B}{c_A}\right)^3 \left(\frac{\mu_A}{\mu_B}\right) = \left(\frac{0.25}{0.13}\right)^3 \left(\frac{2.5}{2.7}\right) = 6.9$$

Damping coefficients for small film clearance (c_A) damper are **~4 times larger** than the coefficients obtained with larger clearance (c_B) SFD.

Compare inertia coeffs. of two dampers

$$\hat{M} = M \Big/ M_{B(r/c_B=0.15)}$$

$e_S/c=0$



Damper A	Damper B
c_A=0.129 mm	c_B=0.254 mm

Recall → $M \sim \left(\frac{1}{c}\right)$

$$\left(\frac{c_B}{c_A}\right) = \left(\frac{0.25}{0.13}\right) = 1.96$$

Added mass coefficients for the small film clearance (c_A) damper are **~1.8 times higher** than the coefficients obtained with larger clearance (c_B) SFD.

Closure I: open ends SFD with end grooves

From circular orbit tests

Damper A	Damper B
c_A=0.129 mm	c_B=0.254 mm

(a)Dynamic pressures in the end **grooves** are not nill.

(b) For both dampers, **direct damping** coefficients do not show great sensitivity to the size of the orbit radius (*r*).

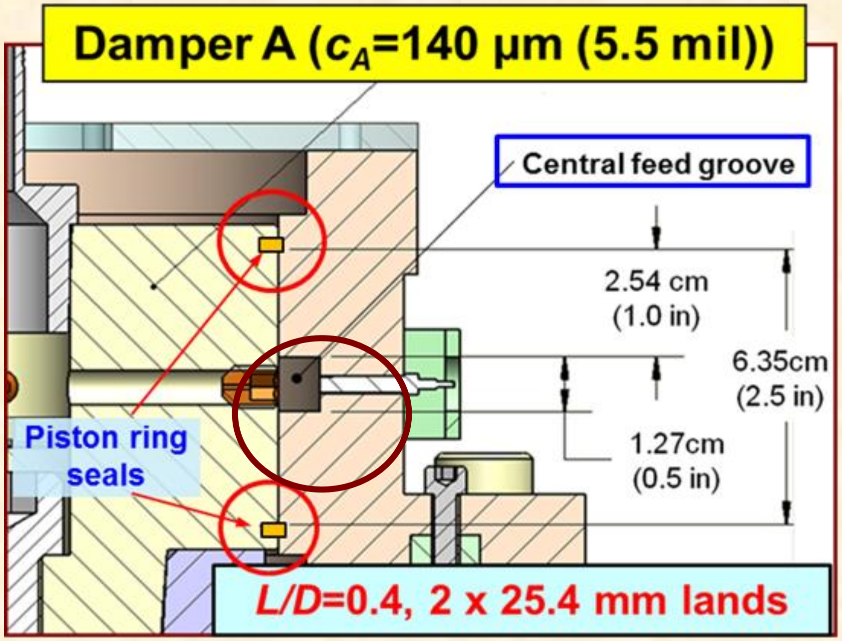
(c) **Inertia** coefficients for the large clearance damper B are **insensitive** to orbit amplitude (*r*), while small clearance damper A shows added masses **decreasing** with an orbit size (*r*).

End grooves lead to a significant squeeze film action that must be carefully characterized.

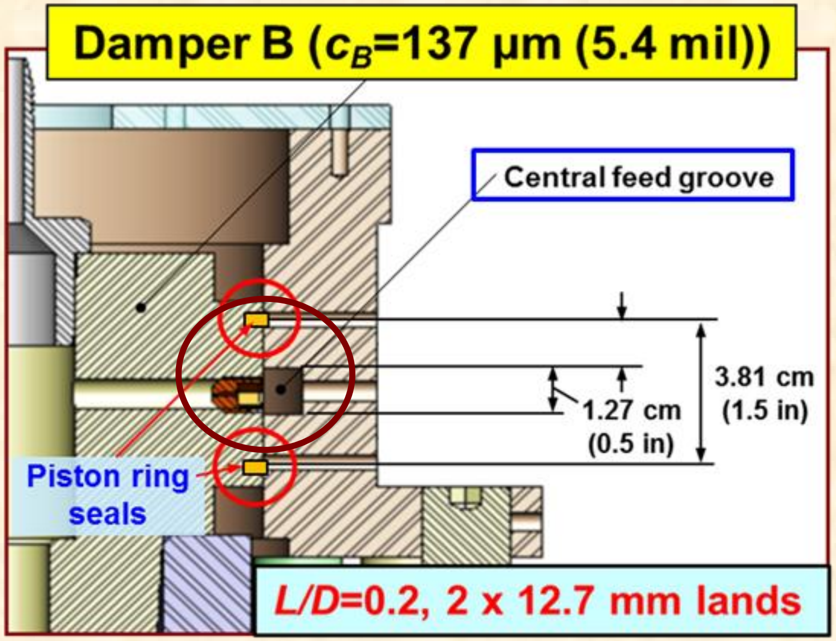
→ $L_{tot} > L_{eff} > L$

SFD force coefficients
Comparison between short and long open ends dampers with a central groove

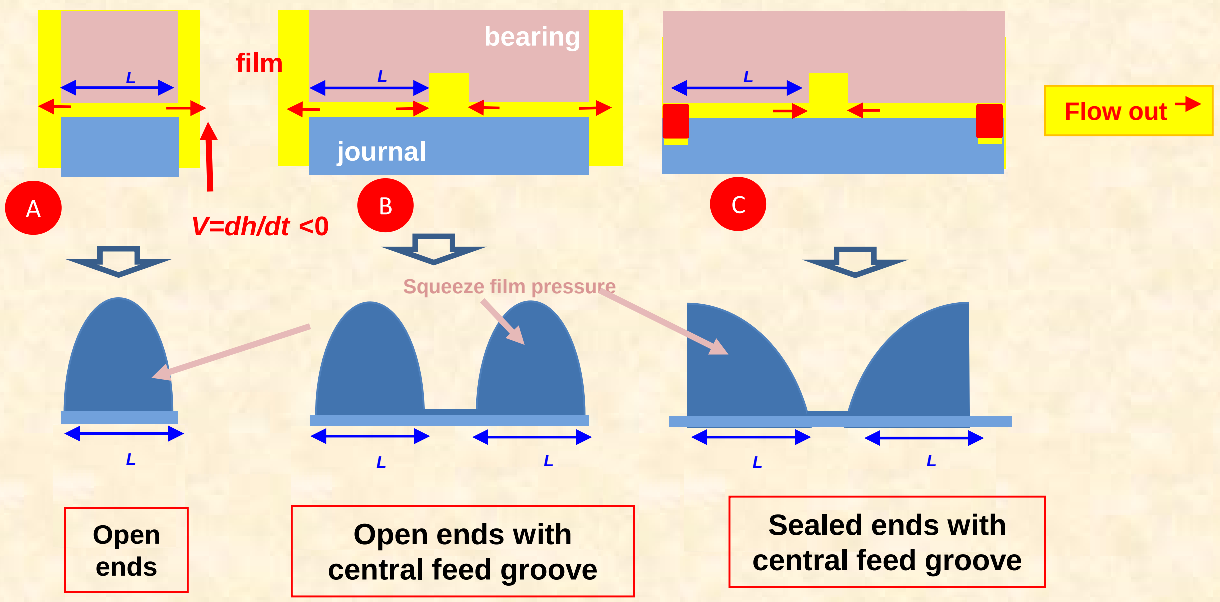
Long: 2 x 25.4 mm lands



Short: 2 x12.7 mm lands



Generation of dynamic pressure in film and groove



Does a central groove isolate a damper into two independent halves?

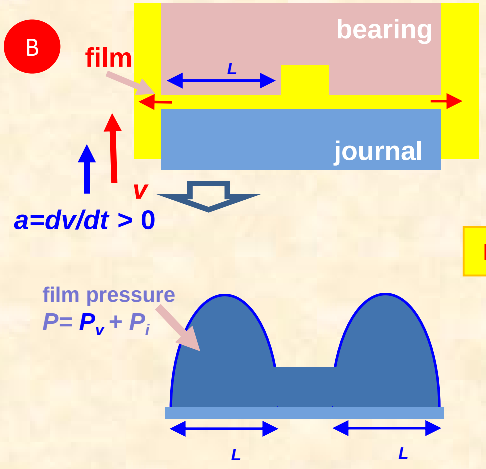
Conventional knowledge:

A groove has constant pressure

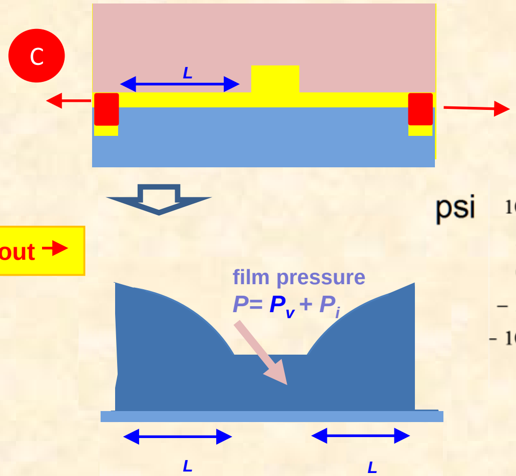
Generation of dynamic pressure

Measured:

Does a central groove isolate a damper into two independent halves?

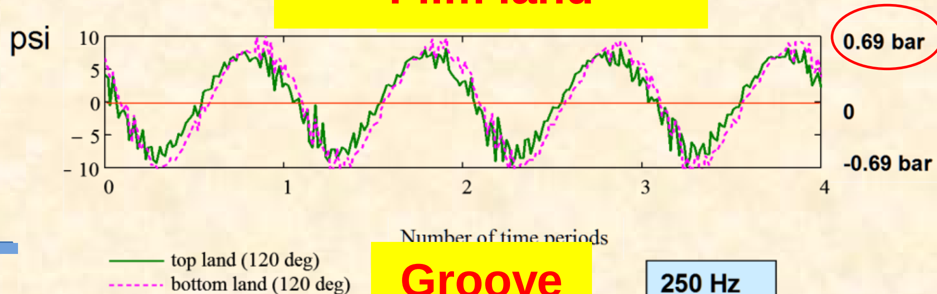


Open ends with central feed groove



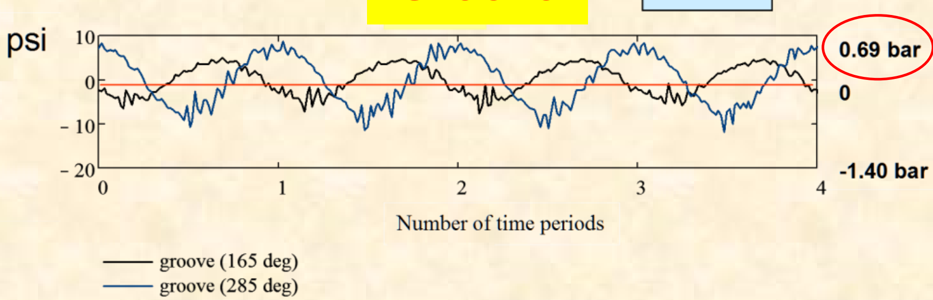
Sealed ends with central feed groove

Film land



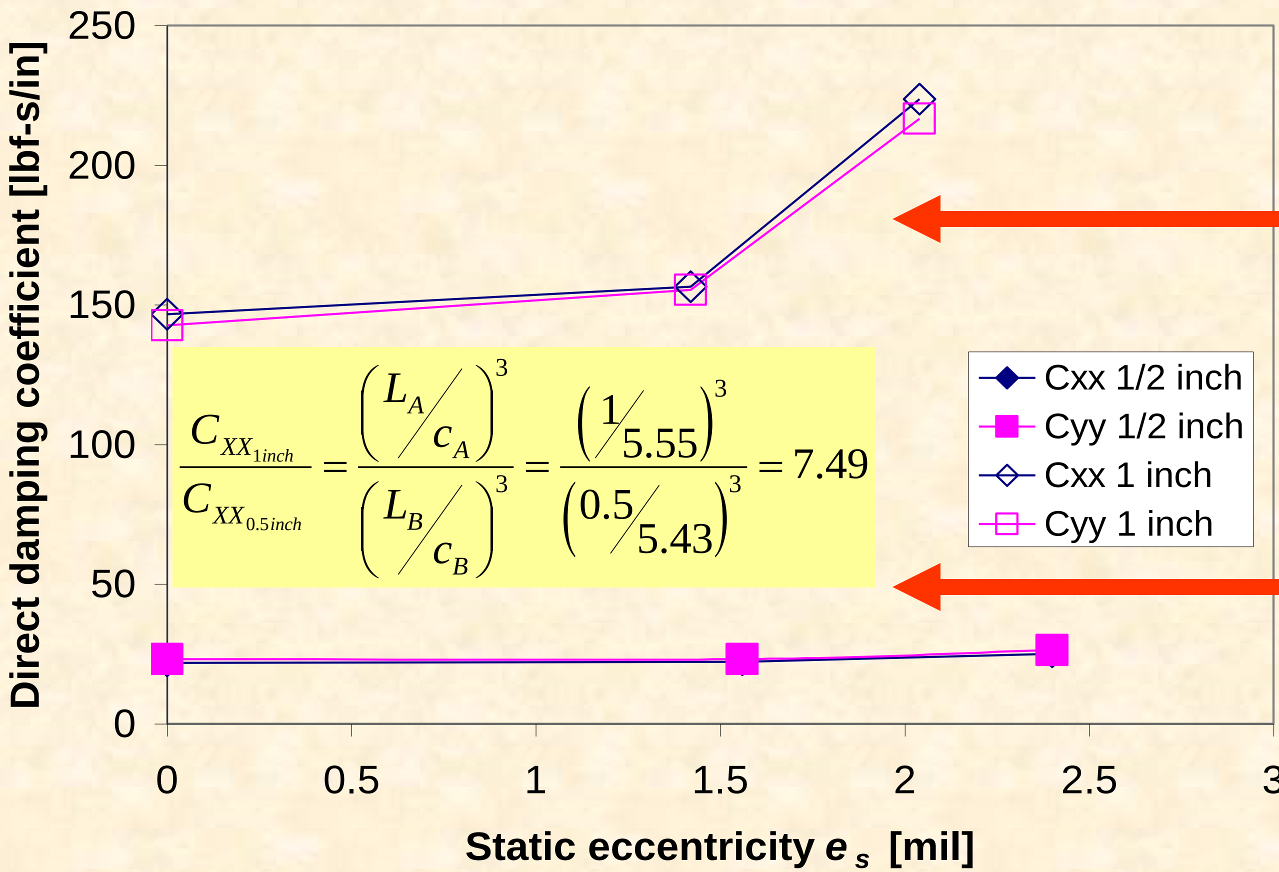
Groove

250 Hz



No! grooves lead to a significant squeeze film (pressure) action

compare SFD damping



$$\frac{C_{XX_{1inch}}}{C_{XX_{0.5inch}}} = \frac{\left(L_A / c_A\right)^3}{\left(L_B / c_B\right)^3} = \frac{\left(1 / 5.55\right)^3}{\left(0.5 / 5.43\right)^3} = 7.49$$

$C_{XX} \sim C_{YY}$

Long (*L*=1 inch)

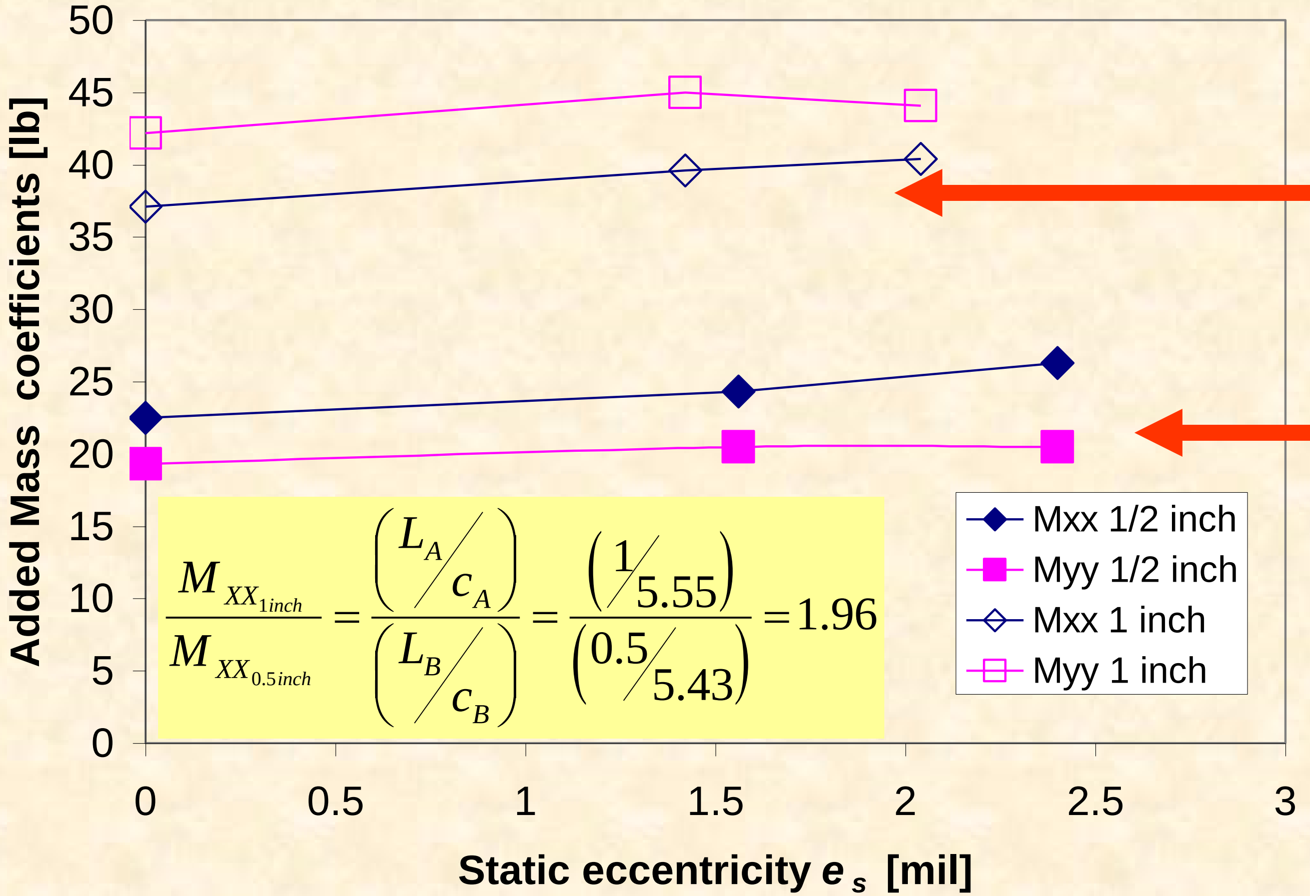
$C_{XX} \sim C_{YY}$

Short (*L*=0.5 inch)

Ratio of coefficients ~ (L/c^3)

Long and short SFDs (circular orbits)

compare SFD inertia



M_{XX} , M_{YY}

Long (L=1 inch)

M_{XX} , M_{YY}

Short (L=0.5 inch)

Ratio of coefficients ~ (L/c)

Long and short SFDs (circular orbits)

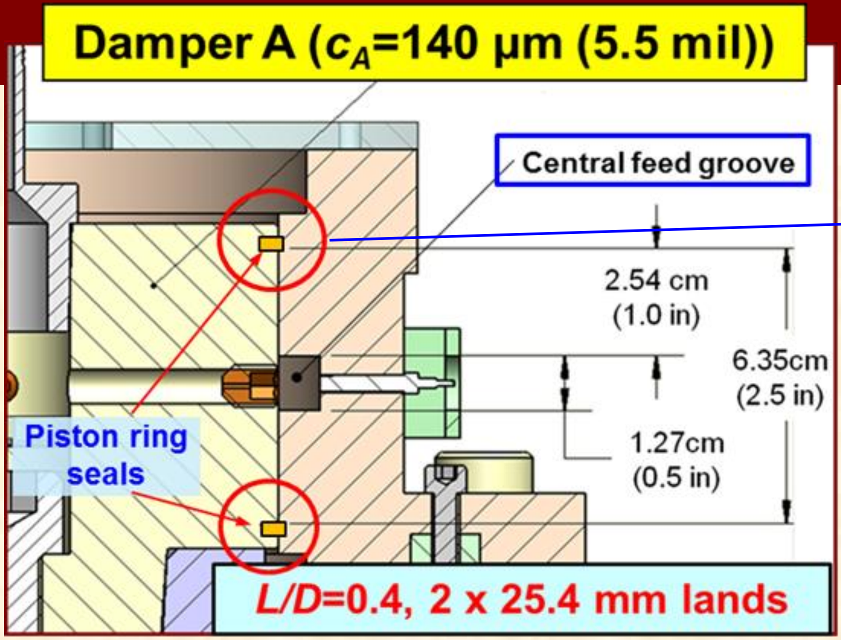
Closure II: Long vs short SFDs

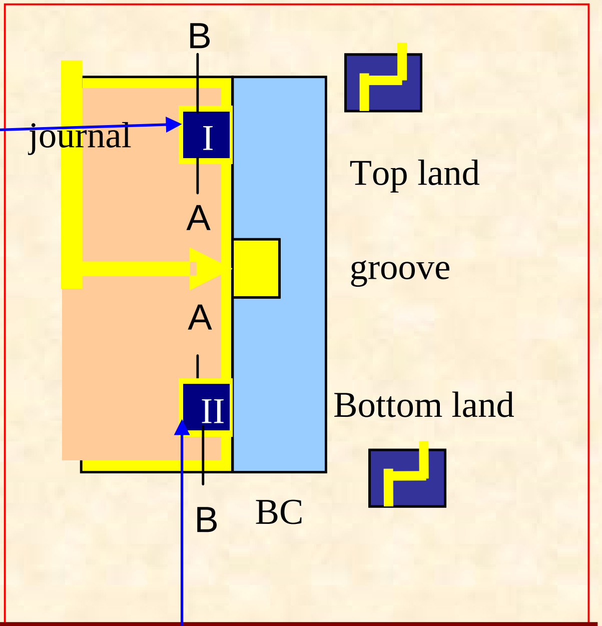
Open ends long damper shows ~ 7 times more damping than short length damper. Inertia coefficients are two times larger.

SFD force coefficients are more a function of static eccentricity (max. 2 mil) than amplitude of whirl & changing little with ellipticity of orbit.

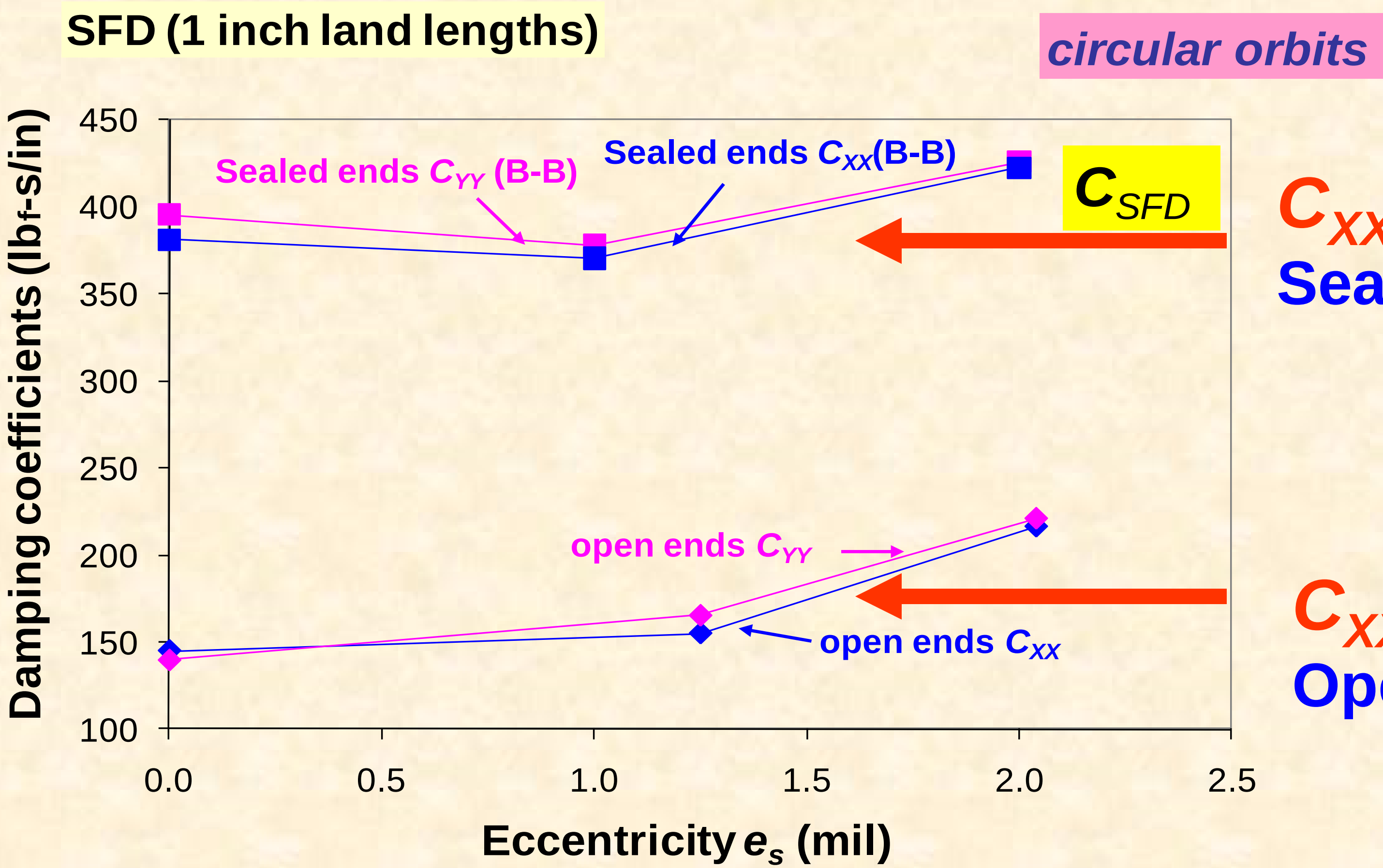
For all damper configurations and most test conditions: cross-coupled damping and inertia force coefficients are a small fraction of the direct force coefficients.

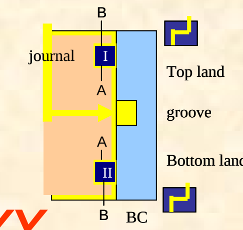
Experimental SFD force coefficients Comparison open ends & sealed ends long (1”) SFD





compare SFD damping





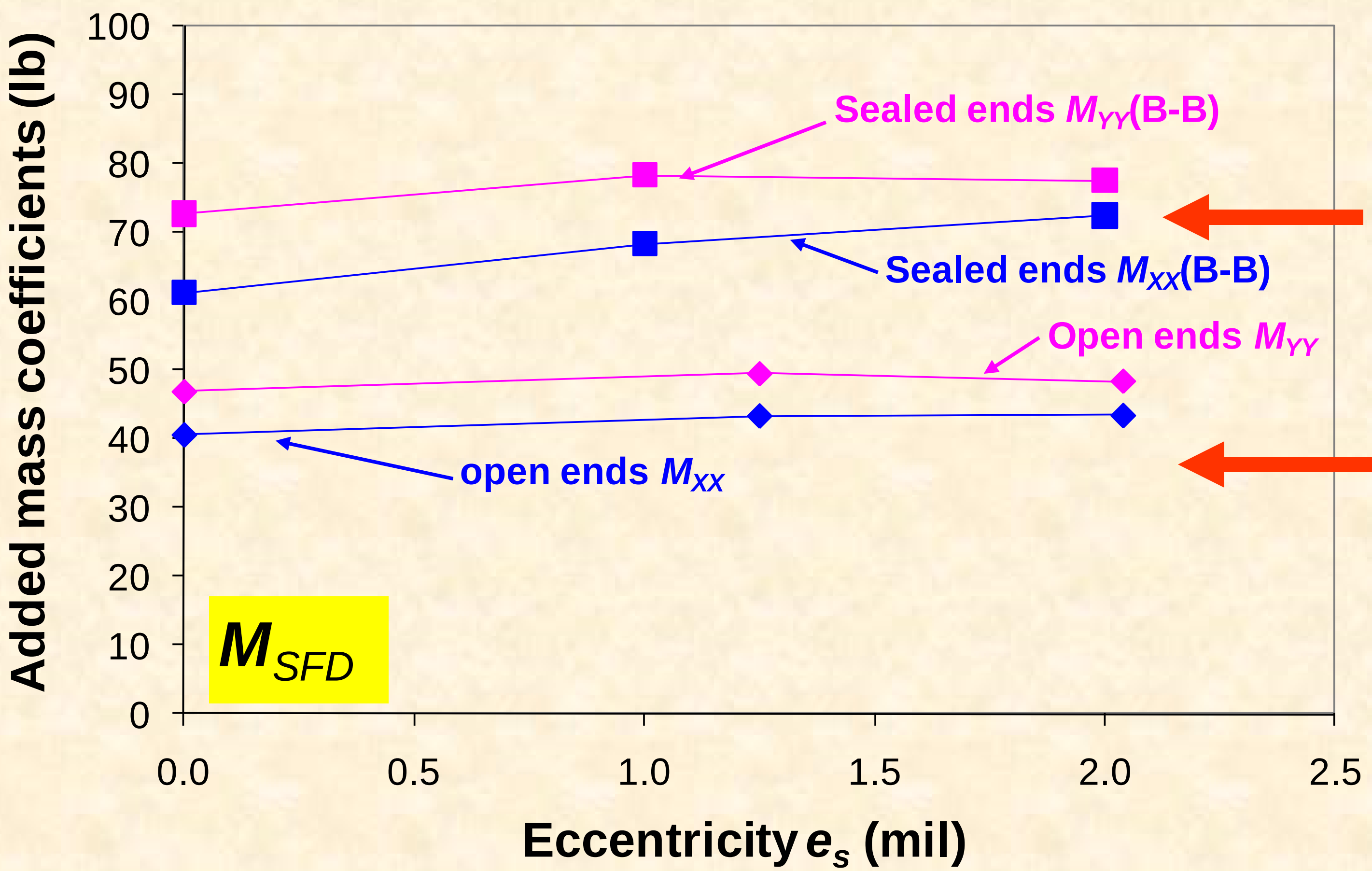
B-B sealed SFD

$C_{XX} \sim C_{YY}$
Sealed ends

$C_{XX} \sim C_{YY}$
Open ends

Open ends vs sealed ends (circular orbits)

compare SFD inertia



M_{XX} , M_{YY}
Sealed ends

M_{XX} , M_{YY}
Open ends

Open ends vs sealed ends (circular orbits)

Closure III: Open vs Sealed SFDS

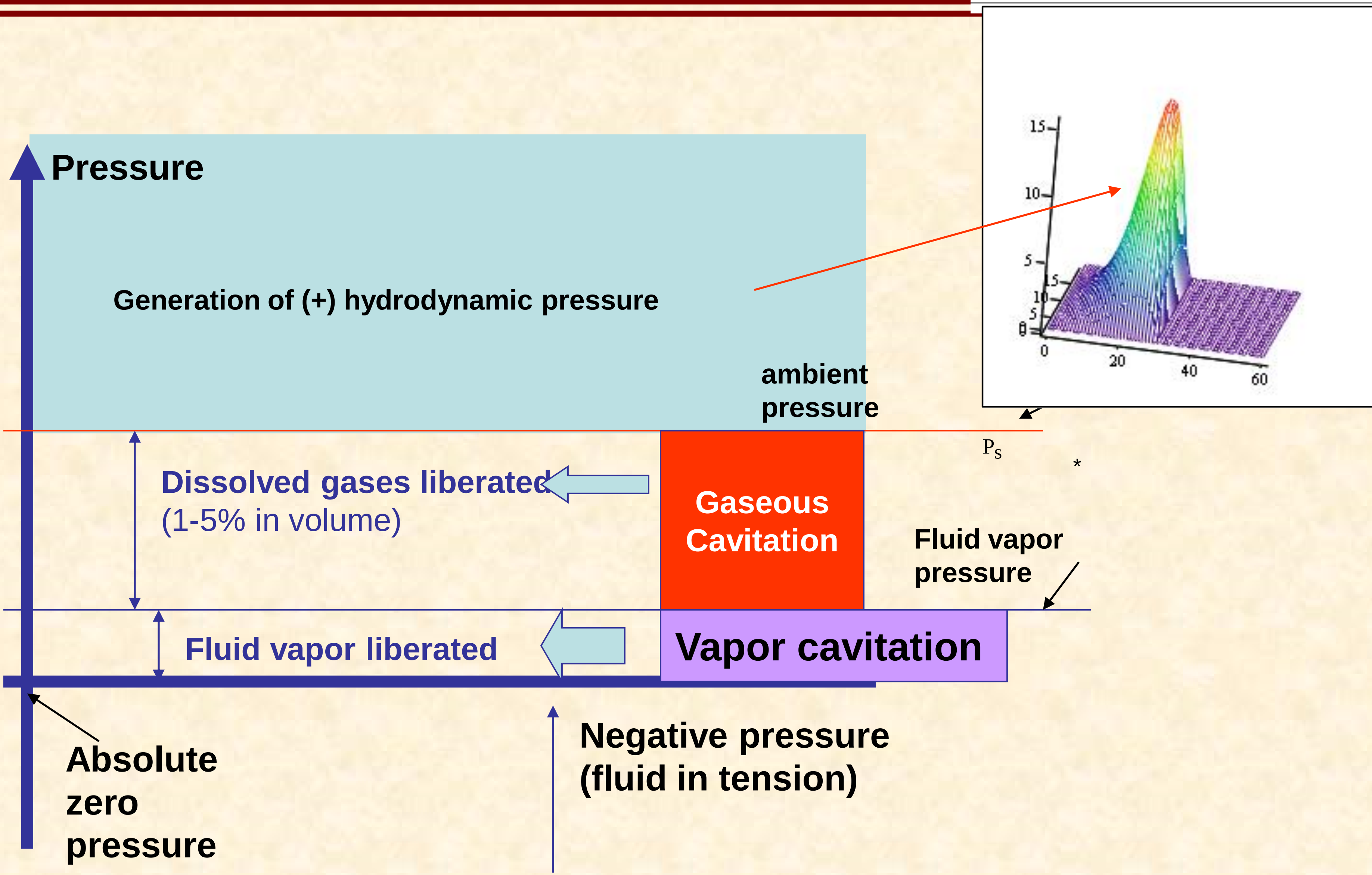
Sealed ends long damper has ~ 3 times more damping than open ends damper. Inertia coefficients are twice as large.

SFD force coefficients are more a function of static eccentricity (max. 2 mil) than amplitude of whirl and changing little with ellipticity of orbit

Proper installation of piston rings is crucial for adequate sealing.

Oil cavitation **OR** air ingestion in SFDs?

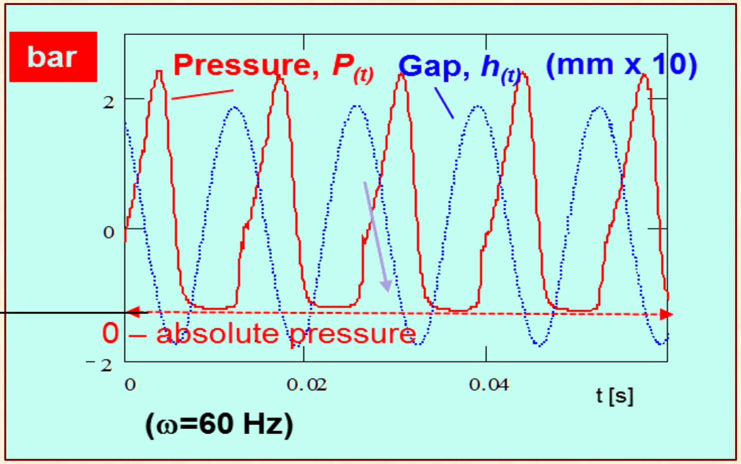
Cavitation in liquid bearings



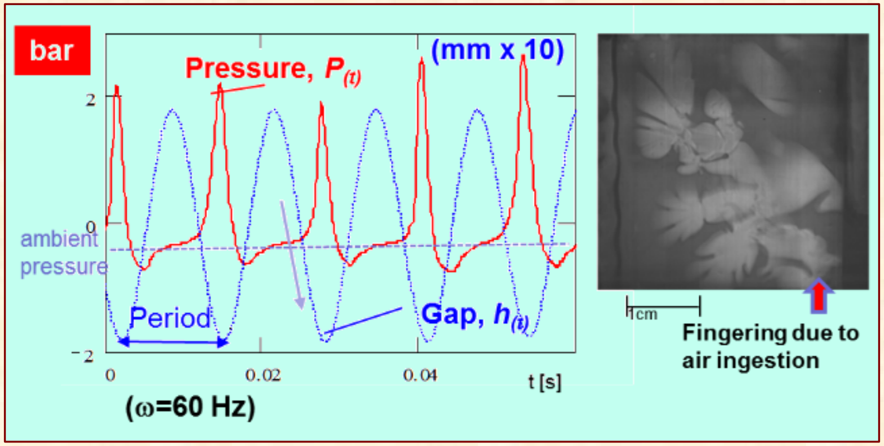
Lubricant cavitation vs. air ingestion in SFDs

Gas cavitation: Cavitation bubble, containing released **dissolved gas** in lubricant, appears steady in a rotating frame.

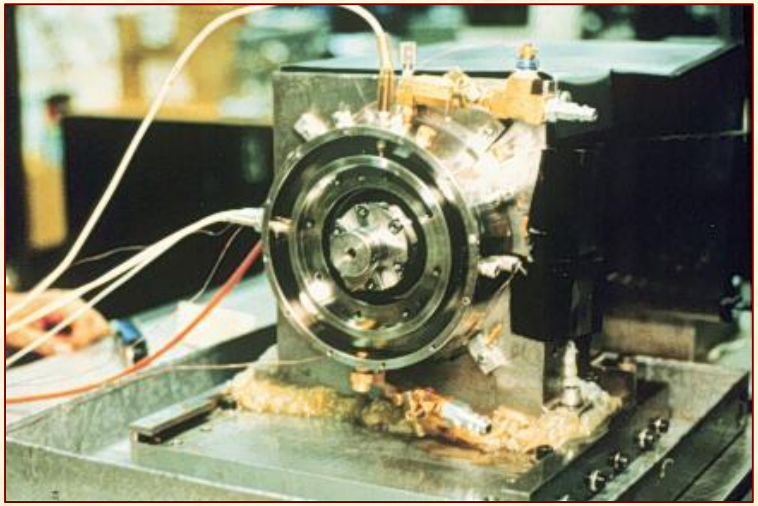
Lubricant vapor cavitation: A **constant pressure zone** at nearly zero absolute pressure.



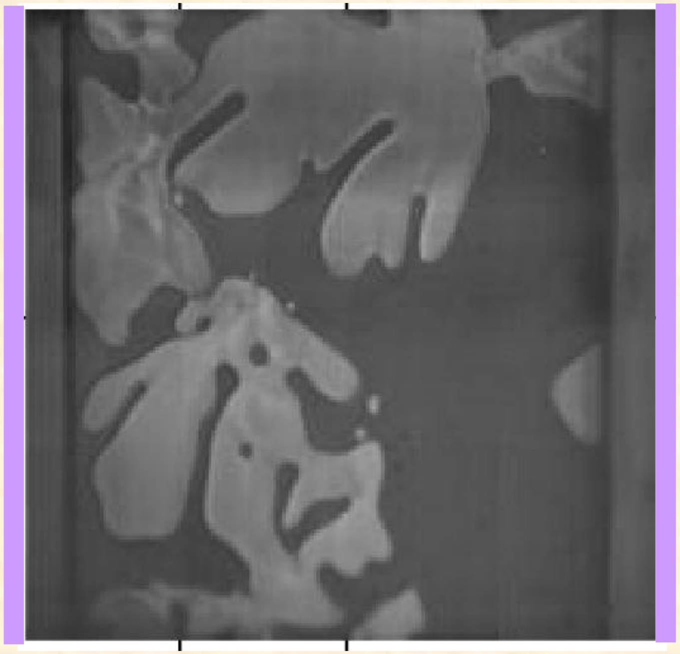
Air ingestion & entrapment: When the gap opens, **air is drawn** to fill an empty volume.



SFD flow visualization of air ingestion



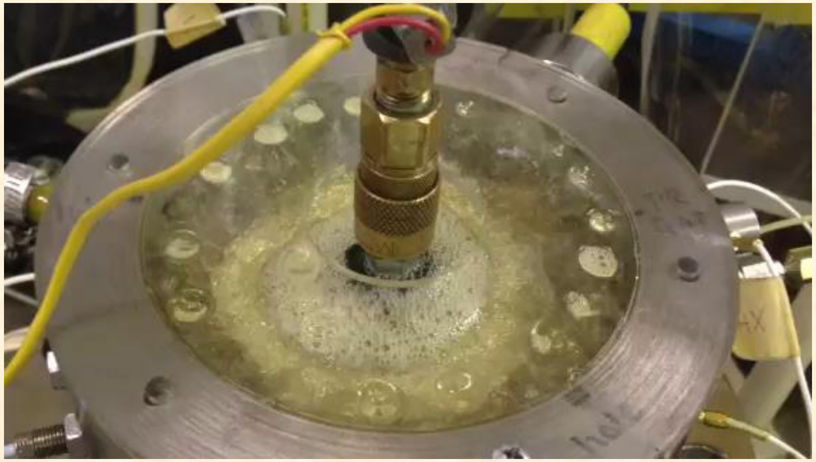
Instances of air entrainment (fingering) and foam formation at damper exit (open end)



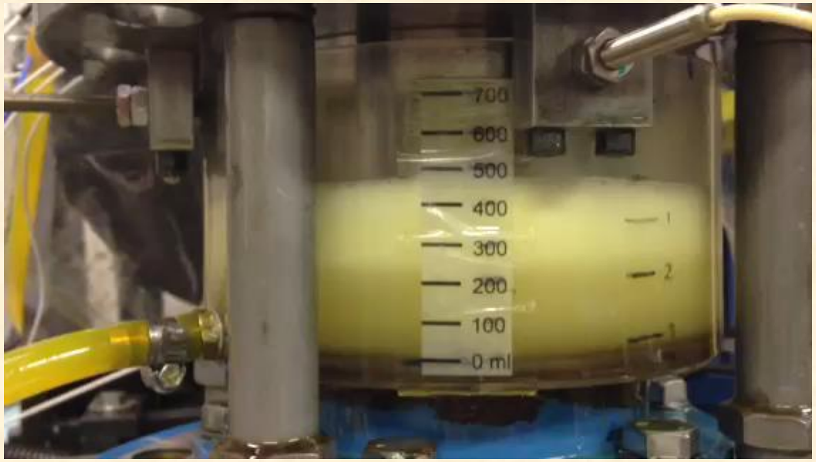
Click above figure to watch video

SFD Operation Issue

Air ingestion and entrapment

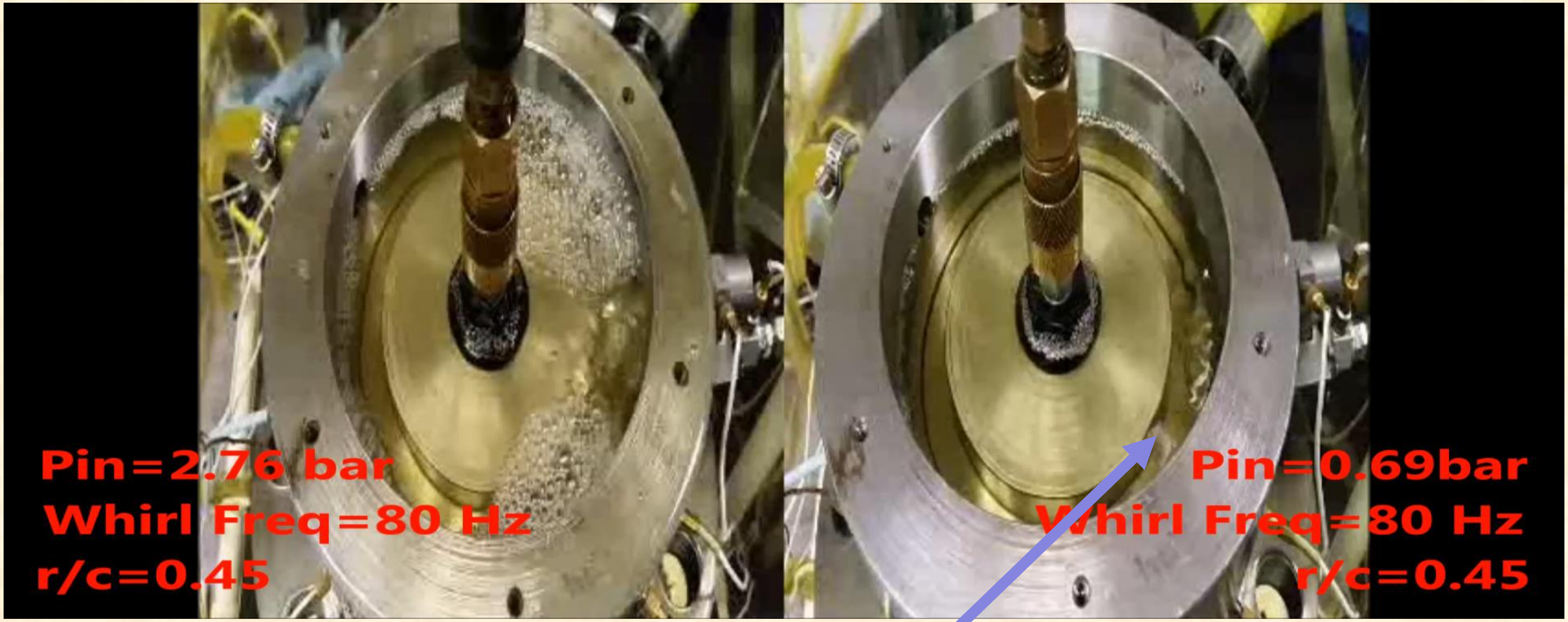


Bubbly lubricant at top and bottom of test rig



Onset of air ingestion

Sealed ends SFD



Oil foamy mixture evolves from the piston ring slit.

Closure IV

From large amplitude whirl orbit tests

(a) **SFD damping** coefficients **increase** with increasing **orbit amplitude** and **static eccentricity**.

(b) **SFD added mass** coefficients **increase** with increasing **static eccentricity** and **decrease** with increasing **orbit amplitude**.

(c) **Deep grooves** contribute to generate significant **added mass** coefficients. Grooves contribute little to magnify damping coefficients.

Predictions correlate well with test results for static eccentricity e_s<0.5c *and* **deviate** with increasing orbit amplitude and static eccentricity.

After 10 years of continued work,

what have we learned?

Conclusion (1):

(a) Damping (*C*) and inertia (*M*) coefficients are ~ isotropic, i.e., C_{XX}~C_{YY} and M_{XX}~M_{YY}. Cross-coupled coefficients are negligible for most whirl type motions.

(b) Simple theory does a modest job in producing physically accurate results for test SFDs with feed groove.

(c) SFDs generate large added mass coefficients, in particular for configurations with (deep) feed and discharge grooves. Fluid inertia must be accounted for in and SFD model.

Grooves lead to a significant squeeze film action (with generation of dynamic pressures)

Conclusion (2):

(d) A sealed SFD produces significantly (4X) more damping and more added mass than an open ends SFD.

(e) The amplitude and shape of whirl motion have small effect on the SFD force coefficients.

(f) Air ingestion impairs the growth of film pressures for increasing orbit amplitudes and frequency → damping coefficients decrease.

The experimental results demonstrate SFDs are mostly **linear** mechanical elements. Simple theory is highly unreliable.

Fundamental learning

The amount of damping (needed) is a critical design consideration

If damping is too large the SFD acts as a rigid constraint to the rotor-bearing system with large forces transmitted to the supporting structure.

If damping is too low, the damper is ineffective and likely to permit large amplitude vibratory motion at synchronous and sub harmonic frequencies.

SFDs must be designed with consideration of the entire rotor-bearing system. SFDs are NOT off-the-shelf elements.

Acknowledgments

Thanks to

Pratt & Whitney Engines (2008-2018)

TAMU Turbomachinery Research Consortium (TRC)

Learn more:

http://rotorlab.tamu.edu

References

Relevant past work

- Della Pietra and Adilleta, 2002, The Squeeze Film Damper over Four Decades of Investigations. Part I: Characteristics and Operating Features, Shock Vib. Dig, (2002), 34(1), pp. 3-26, Part II: Rotordynamic Analyses with Rigid and Flexible Rotors, Shock Vib. Dig., (2002), 34(2), pp. 97-126.

- Zeidan, F., L. San Andrés, and J. Vance, 1996, "Design and Application of Squeeze Film Dampers in Rotating Machinery," Proceedings of the 25th Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, September, pp. 169-188.

- Zeidan, F., 1995, "Application of Squeeze Film Dampers", Turbomachinery International, Vol. 11, September/October, pp. 50-53.

- Vance, J., 1988, "Rotordynamics of Turbomachinery," John Wiley and Sons, New York

Parameter identification:

- Tiwari, R., Lees, A.W., Friswell, M.I. 2004. "Identification of Dynamic Bearing Parameters: A Review," *The Shock and Vibration Digest*, 36, pp. 99-124.

References **SFDs**

2017	San Andrés, L., **Den, S.,** and **Jeung, S-H**., 2018, "On the Force Coefficients of a Flooded, Open Ends Squeeze Film Damper: from Theory to Practice (and Back)," ASME J. Eng. Gas Turbines Power, Vol. **140**(1)**,** (ASME GT2017-63152)
2016	San Andrés, L., and Jeung, S.-H, 2016, Response of a Squeeze Film Damper-Elastic Structure System to Multiple Consecutive Impact Loads," ASME J. Eng. Gas Turbines Power. Vol. **138** (12), (ASME GT2016-56695)
2016	San Andrés, L., S-H Jeung, S. Den, and G. Savela, "Squeeze Film Dampers: A Further Experimental Appraisal of Their Dynamic Performance," Proceedings of the 45th Turbomachinery Symposium, Houston, TX, Sept 12-15. (Tutorial).
2015	San Andrés, L., Jeung, S-H, and Bradley, G., "Force Coefficients for a Short Length Open-Ends Squeeze Film Damper with End Grooves: Experiments and Predictions," ASME GT 2015-43096
2014	San Andrés, L., Jeung, S-H, and Bradley, G., "Experimental performance of a Open Ends Centrally Grooved Squeeze film Damper Operating with Large Amplitude Orbital Motions," ASME GT 2014-25413.
2014	San Andrés, L., "Force Coefficients for a Large Clearance Open Ends Squeeze Film Damper with a Central Groove: Experiments and Predictions," Tribol. Int., **71**, pp. 17-25
2013	San Andrés, L., and Seshagiri, S., "Damping and Inertia Coefficients for Two End Sealed Squeeze Film Dampers with a Central Groove: Measurements and Predictions," ASME J. Eng. Gas Turbines Power, **135**(12), p. 112503.
2012	San Andrés, L., 2012, "Damping and Inertia Coefficients for Two Open Ends Squeeze Film Dampers with a Central Groove: Measurements and Predictions," ASME J. Eng. Gas Turbines Power, 134(10), p. 102506.
2012	San Andrés, L., and Delgado, A., "A Novel Bulk-Flow Model for Improved Predictions of Force Coefficients in Grooved Oil Seals Operating Eccentrically," ASME J. Eng. Gas Turbines Power, 134(5), p. 052509.

References SFDs

2010	Delgado, A., and San Andrés, L., 2010, "A Model for Improved Prediction of Force Coefficients in Grooved Squeeze Film Dampers and Grooved Oil Seal Rings", ASME J. Tribol., 132(3), p. 032202.
2012	San Andrés, L., 2012, "Squeeze Film Damper: Operation, Models and Technical Issues," Modern Lubrication Theory, Notes 13, Texas A&M University Digital Libraries, https://repository.tamu.edu/handle/1969.1/93197
2010	Delgado, D., and San Andrés, L., 2010, "Identification of Squeeze Film Damper Force Coefficients from Multiple-Frequency, Non-Circular Journal Motions," ASME J. Eng. Gas Turbines Power, Vol. 132 (April), p. 042501 (ASME Paper No. GT2009-59175)
2009	Delgado, A., and San Andrés, L., 2009, "Nonlinear Identification of Mechanical Parameters on a Squeeze Film Damper with Integral Mechanical Seal," ASME Journal of Engineering for Gas Turbines and Power, Vol. 131 (4), pp. 042504 (ASME Paper GT2008-50528)
2003	San Andrés, L., and S. Diaz, 2003, "Flow Visualization and Forces from a Squeeze Film Damper with Natural Air Entrainment," ASME Journal of Tribology, Vol. 125, 2, pp. 325-333
2001	Diaz, S., and L. San Andrés, 2001, "Air Entrainment Versus Lubricant Vaporization in Squeeze Film Dampers: An Experimental Assessment of their Fundamental Differences," ASME Journal of Gas Turbines and Power, Vol. 123 (4), pp. 871-877
2000	Tao, L., S. Diaz, L. San Andrés, and K.R. Rajagopal, 2000, "Analysis of Squeeze Film Dampers Operating with Bubbly Lubricants" ASME Journal of Tribology, Vol. 122, 1, pp. 205-210
1997	Arauz, G., and L. San Andrés, 1997 "Experimental Force Response of a Grooved Squeeze Film Damper," Tribology International, Vol. 30, 1, pp. 77-86
1996	San Andrés, L., 1996, "Theoretical and Experimental Comparisons for Damping Coefficients of a Short Length Open-End Squeeze Film Damper," ASME Journal of Engineering for Gas Turbines and Power, Vol. 118, 4, pp. 810-815