Special Lecture
2022 Aircraft Engine Technology Award

ASME INTERNATIONAL GAS TURBINE INSTITUTE

TURBO EXPO 2022

Turbomachinery Technical Conference & Exposition

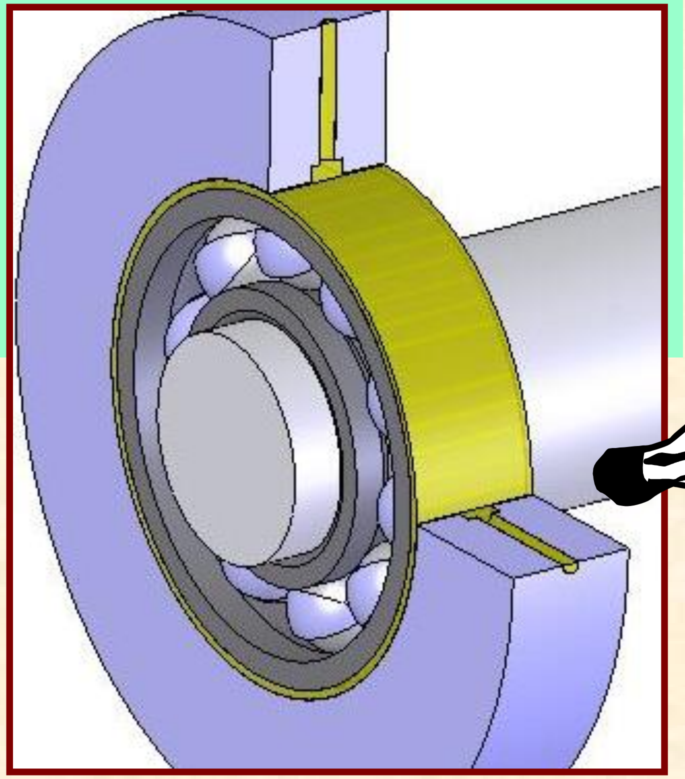
JUNE 13–17, 2022

ROTTERDAM AHOY CONVENTION CENTRE
ROTTERDAM,NL

Proceedings of ASME Turbo Expo 2022: Turbomachinery Technical Conference and Exposition, June 13-17,2022. Rotterdam, The Netherlands

2022 Aircraft Engine Technology Award Special Lecture

MEASUREMENTS AND MODELS OF SQUEEZE FILM DAMPERS' FORCED RESPONSE AND A BIRD'S VIEW TO AIR INGESTION & ENTRAPMENT



Luis San Andrés

Mast-Childs Chair Professor

Fellow ASME

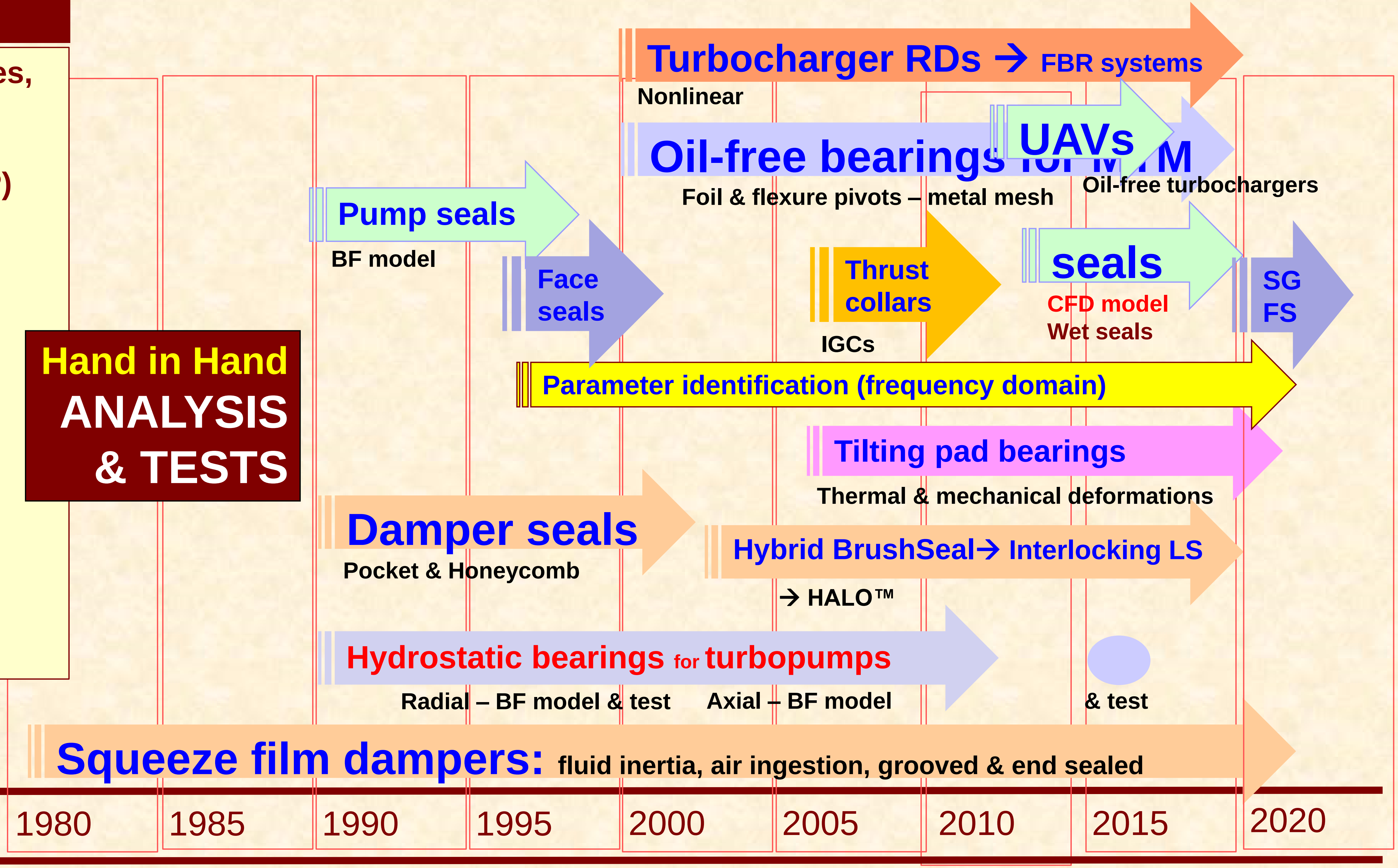
TEXAS A&M UNIVERSITY

J. Mike Walker '66 Department of Mechanical Engineering

Timeline since dinosaur age

Funding Sources

John Crane, Baker-Hughes,
Trane, Elliott Co.
Blue Origin,
Army Research Lab (CUP)
NSF, NASA GRC,
Pratt & Whitney
Northrop Grumman
Rocketdyne
Honeywell TT,
Danfoss TurboCor
Borg-Warner TC,
Torishima Pumps
MHI, Hitachi RL,
Samsung, Key Yang,
Hyundai HI, Capstone MT
Siemens **TRC**



Today a bird's view into SFD forced performance

Squeeze film dampers:

Air ingestion & entrapment

1980	1985	1990	1995	2000	2005	2010	2015	2020
	GE Aviation, Pratt & Whitney		NSF, TRC		Honeywell Aerospace, TRC	Pratt & Whitney, TRC		

TRC: Turbomachinery Research Consortium

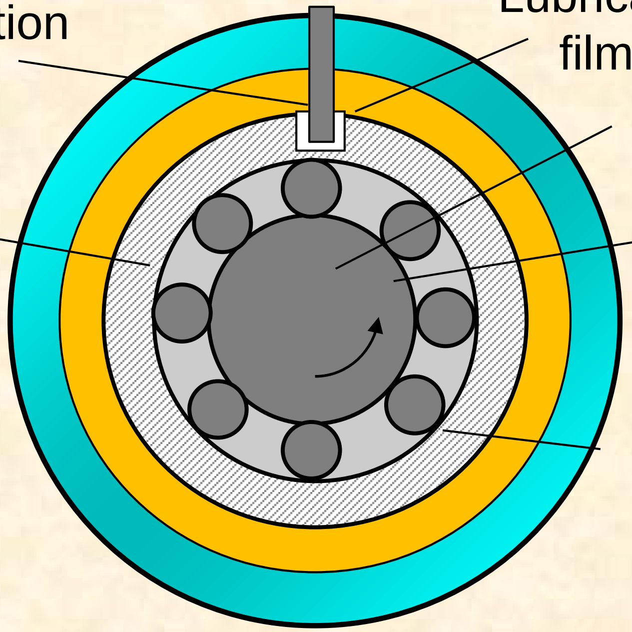
The early quest – last millennium

Whose woods these are I think I know…..

The woods are lovely, dark and deep,
But I have promises to keep,
And miles to go before I sleep,
And miles to go before I sleep.

Robert Frost, “Stopping by Woods on a Snowy Evening”

Squeeze Film Dampers (SFDs)



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

SFDs aid to attenuate rotor vibrations, suppress system instabilities, and provide mechanical isolation.

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

Most common problems in rotordynamics

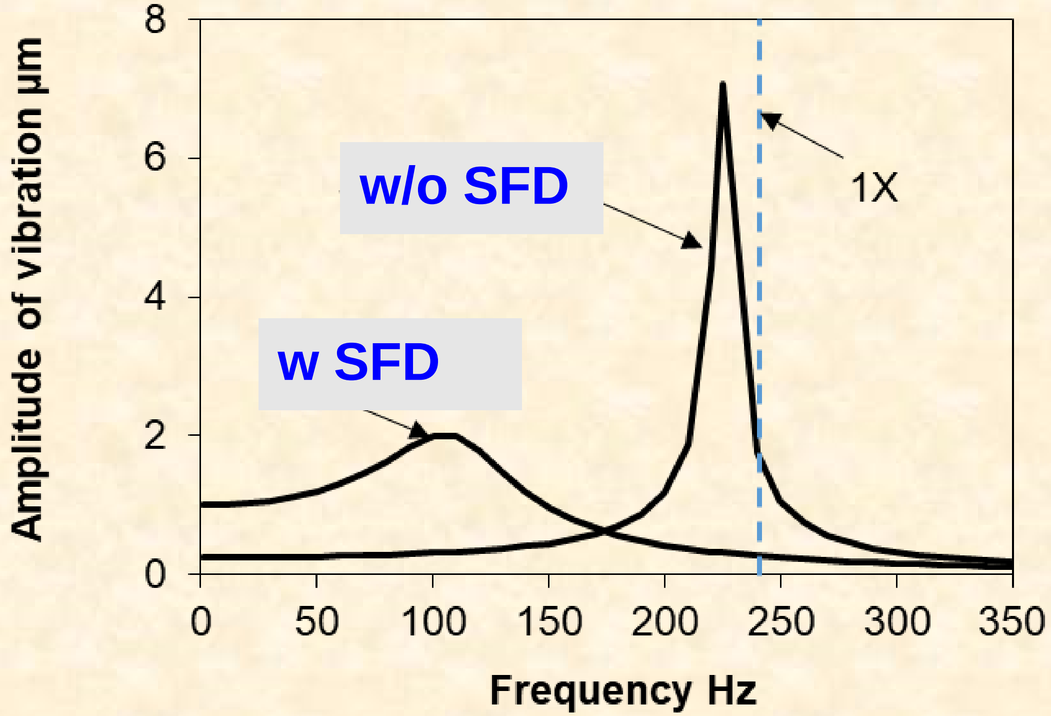
1. Excessive steady state synchronous vibration levels:

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.

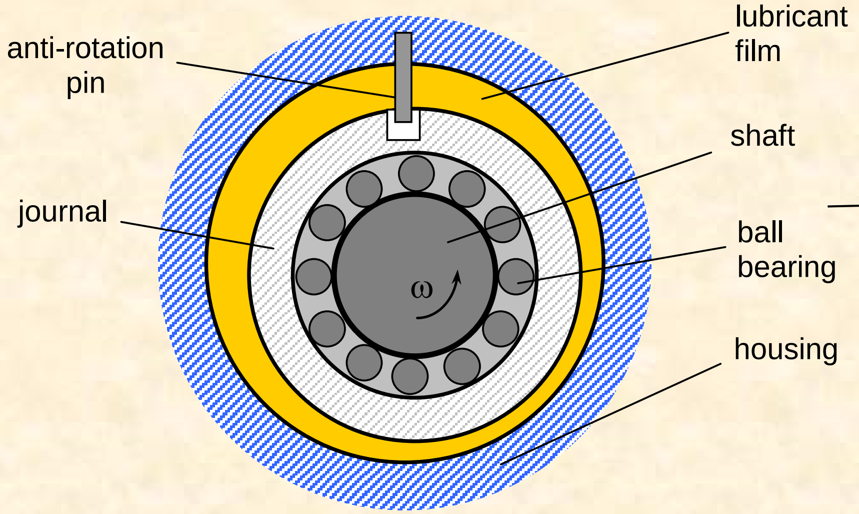
Rise natural frequency of rotor system as much as possible.

SFDs

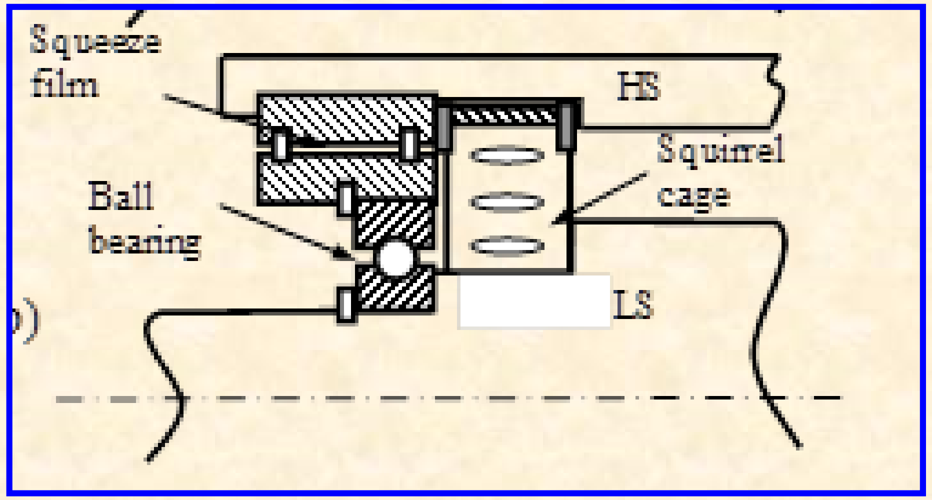
Introduce damping to increase onset rotor speed above the operating speed range.

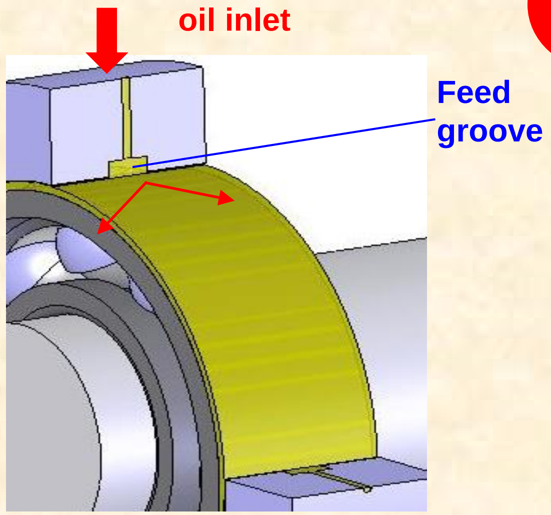
SFD configurations

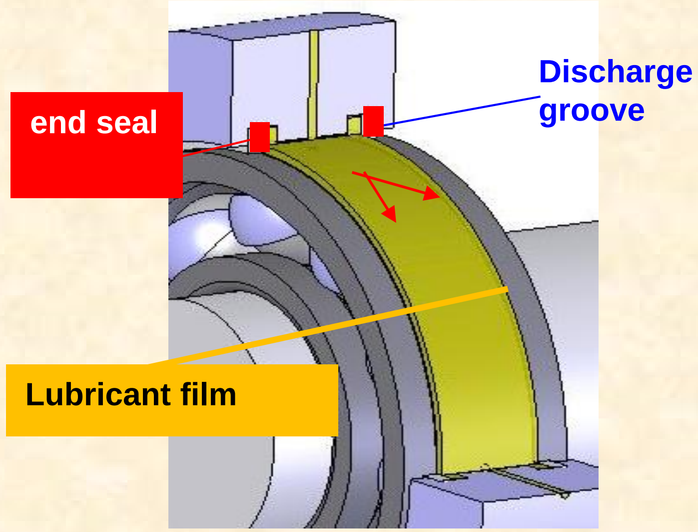
a **SFD with anti-rotation pin**



b **SFD with elastic cage**







(c) with a supply groove - open ends

(d) with a supply hole - end seals

Brief history of SFDs

Parsons (1889)

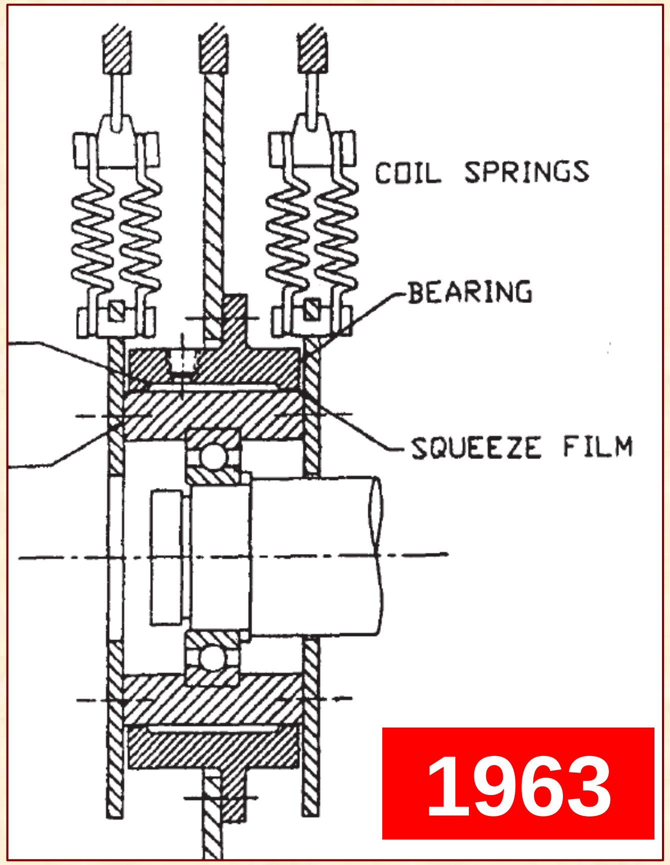
Discloses first use of a SFD in first modern-day steam turbine.

(by Topori at Polish Wikipedia)

Cooper (1963)

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

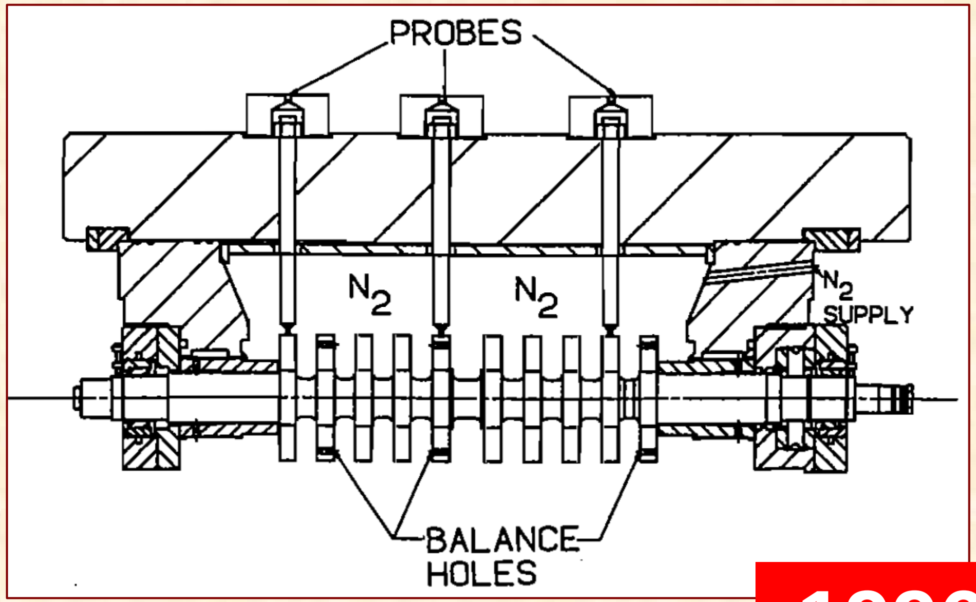
Since the 1970s, SFDs are essential components in aircraft engines and high pressure centrifugal compressors.



Brief history of SFD (Turbomachinery Symposium)

Zeidan et al. (1996)

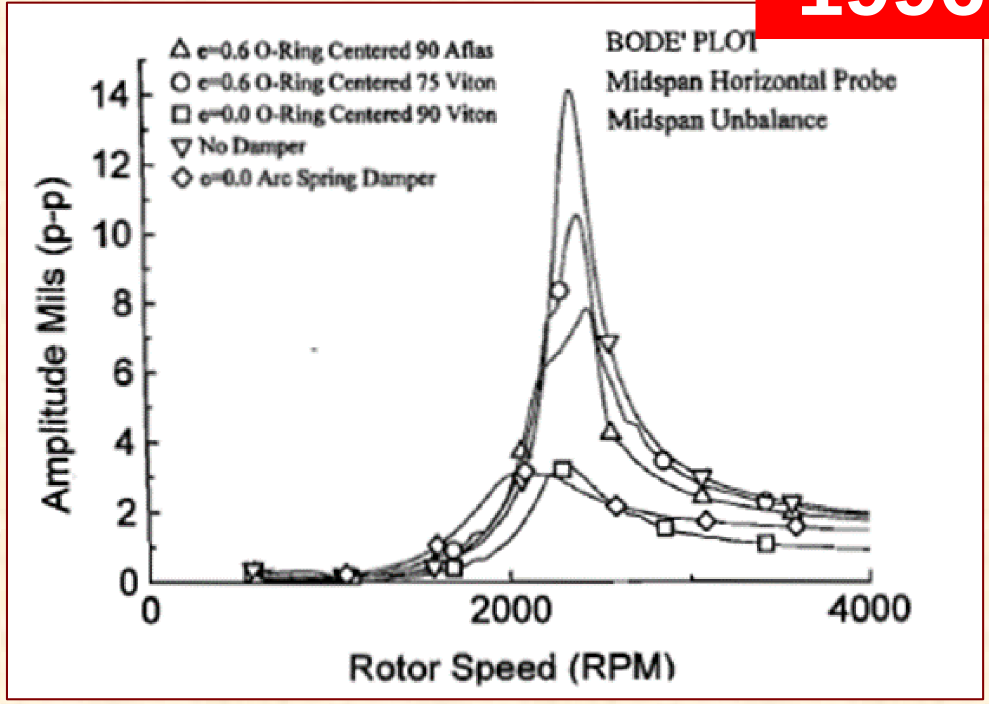
Discuss major technical issues for SFD integration into turbomachinery: oil cavitation vs. air ingestion and fluid inertia effects.



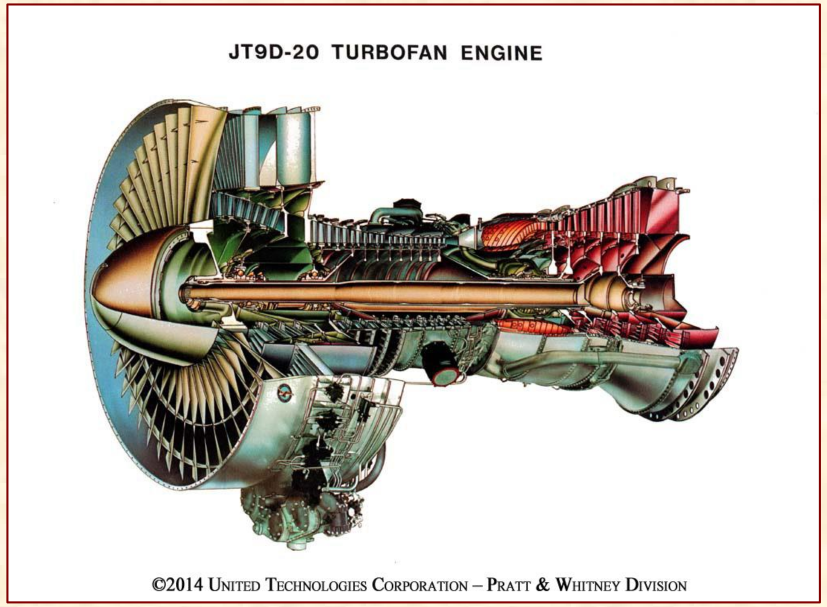
1996

Kuzdal and Hustak (1996)

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



SFD applications



Jet engines with rolling element bearings:

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

Hydrocarbon compressors & steam turbines

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:

- Tolerance to larger rotor motions * Simpler alignment
- Reduced balancing requirements * Less mount fatigue

More recent literature on SFDs

Della Pietra and Adilleta (2002): Review of research conducted on SFDs over last 40 years (since 1960s')

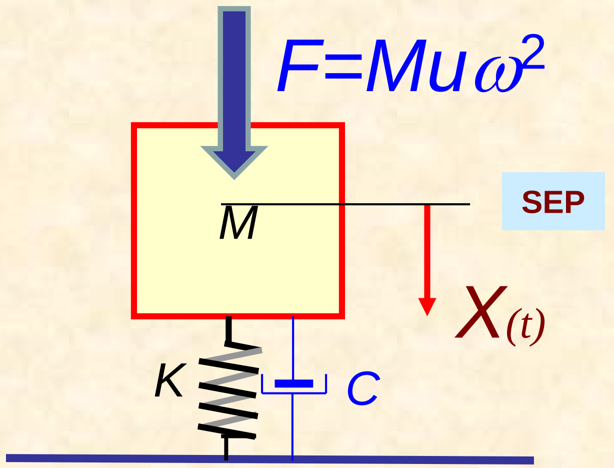
(2006-2010) San Andrés and Delgado (SFD & MECHANICAL SEAL, improved predictive models).

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

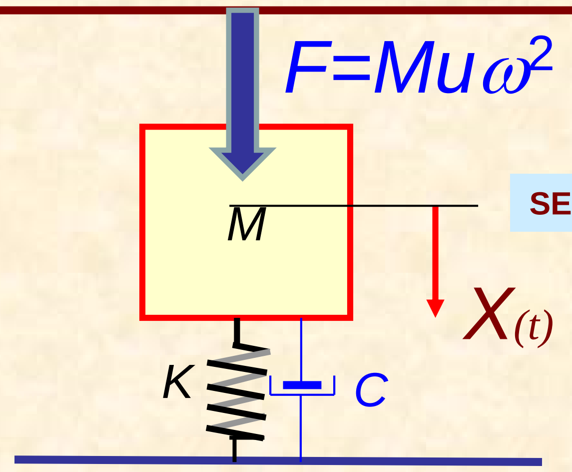
(2012-2020) San Andrés and students (SFDs for aircraft)

GT 2012-68212 , GT 2013-94273 , GT 2014-26413, GT 20015-43096,
GT 2016-43096, GT 2016-56695, 2016 A/TPS, GT2018-76224,
GT 2019-90330, GT2020-14182, GT2021-58627, GT2022-81990

How does viscous damping affect the response of a mechanical system?



1DOF spring-damper-mass system



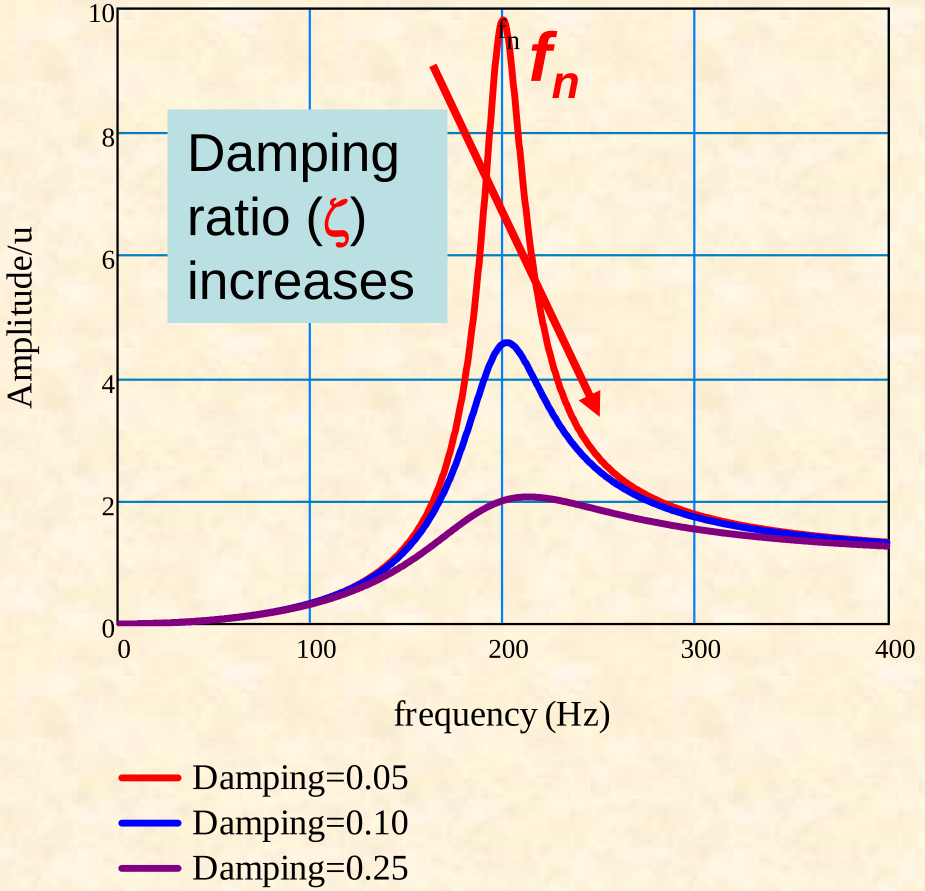
EOM: $M\,Ax = F - F_{damper} - F_{spring}$

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

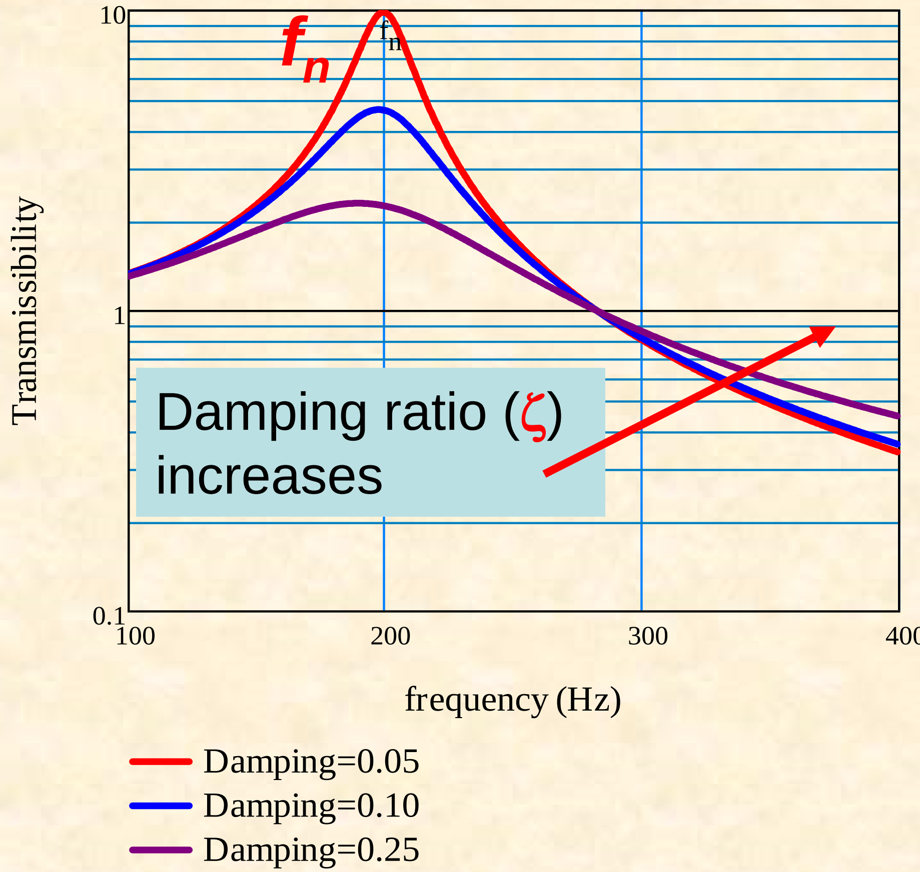
System response defined by natural frequency (f_n) & damping ratio (ζ)

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

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

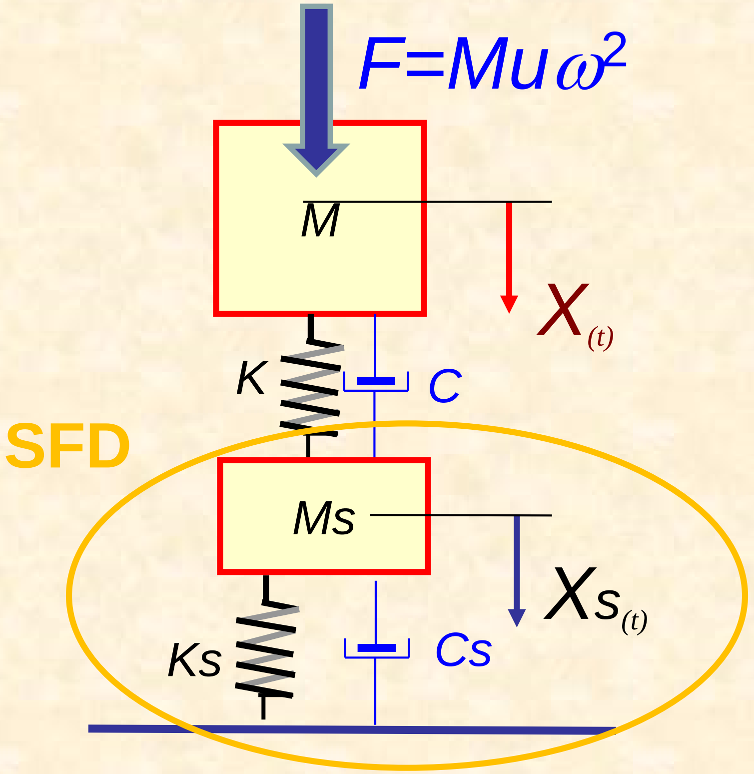


Transmissibility (to ground)



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

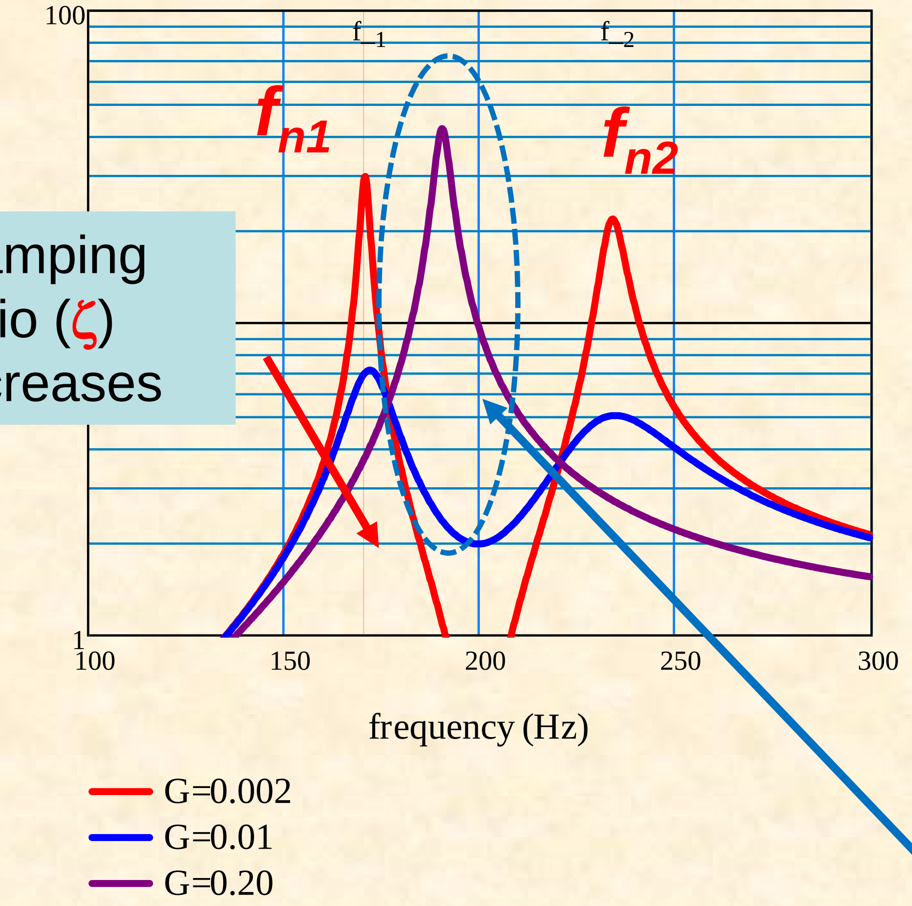
2 DOF K-C-M system : rotor on flexible supports



example

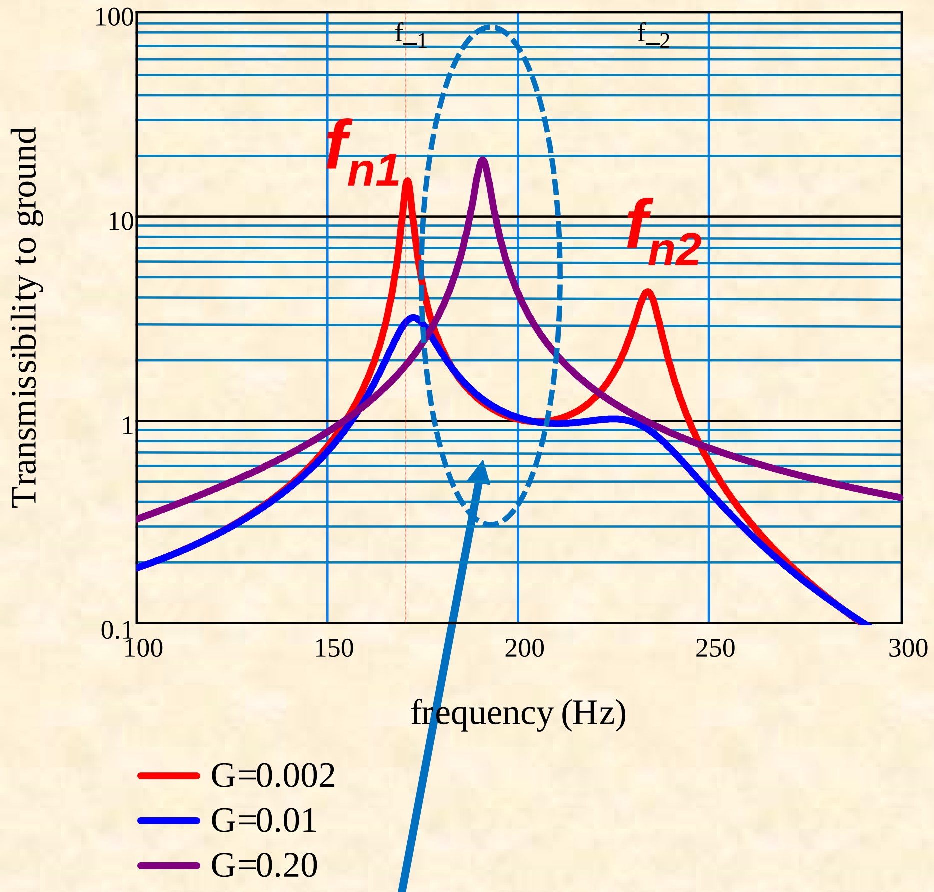
$$\frac{M_s}{M} = \frac{K_s}{K} = 0.1;\ C = 0;\ C_s \text{ varies}$$

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



Damping ratio (ζ) increases

Transmissibility (to ground)



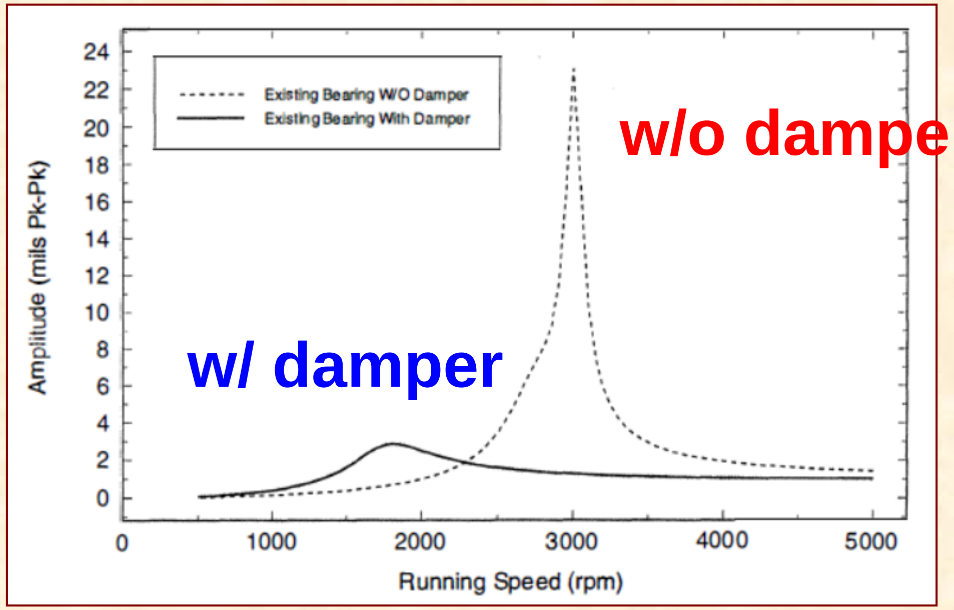
More complicated response. Damping helps only when rotor traverses a critical speed (natural frequency=f_{n1} and f_{n2}) but increases force transmissibility.

Excessive damping LOCKS supports and increases system response.

A typical SFD application

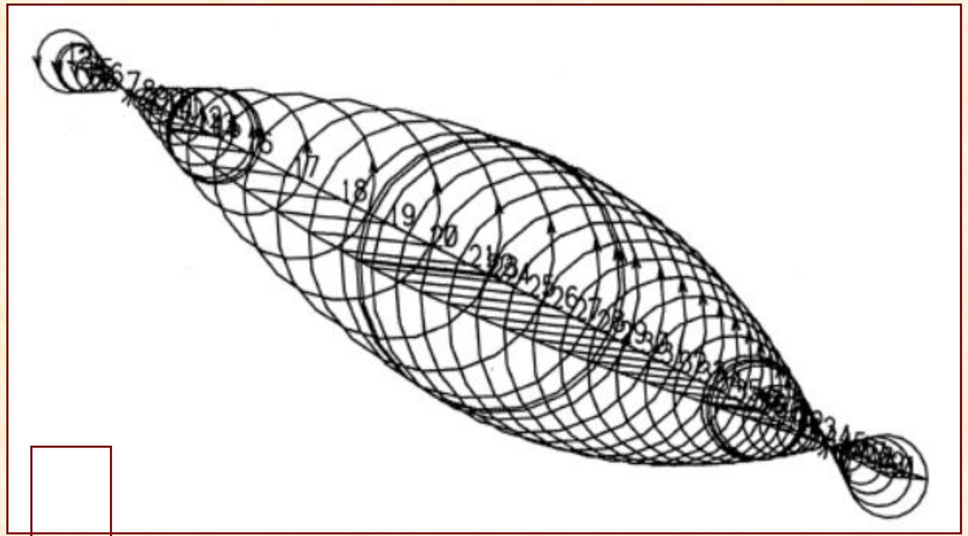
2015 Ertas et al., J. Eng. for Gas Turb. Pwr

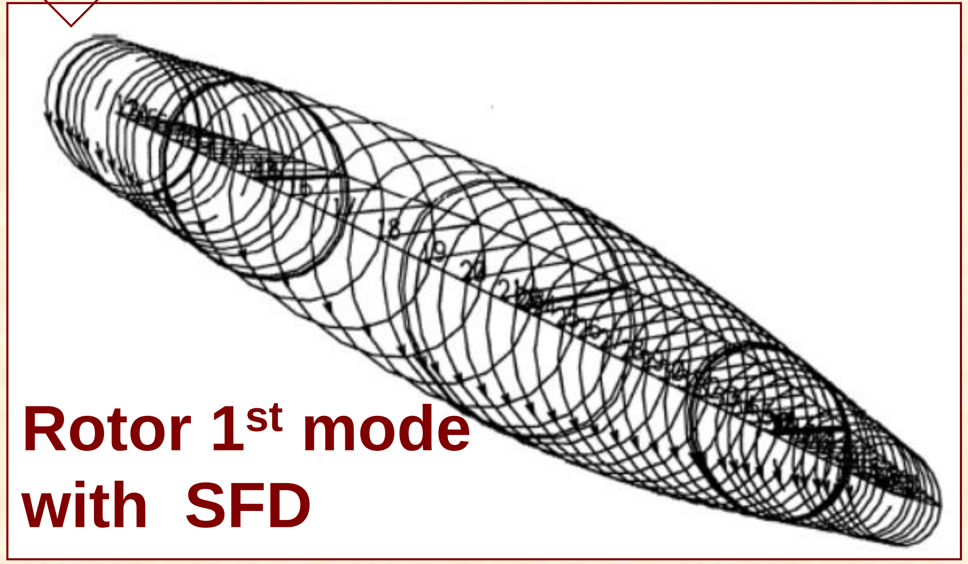
- A compressor vibrates ++ at its 1st forward mode. Bearings don't help since placed at mode-nodal locations.
- SFD support springs are soft → drop the system natural frequency and increase the effective damping ratio.
- Rotor motions greatly reduce while passing the (low) critical speed. SSV at the first forward mode eliminated.



Dampers often used to control placement of critical speed besides adding damping.

Rotor 1st forward mode w/o SFD

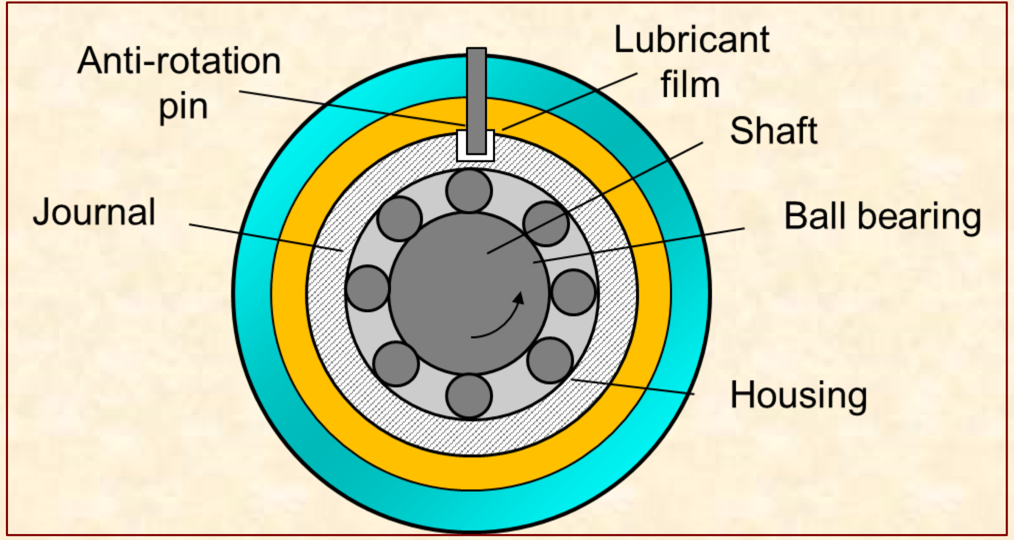




Rotor 1st mode with SFD

SFDs' fundamental design consideration

The amount of damping needed is critical.



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.

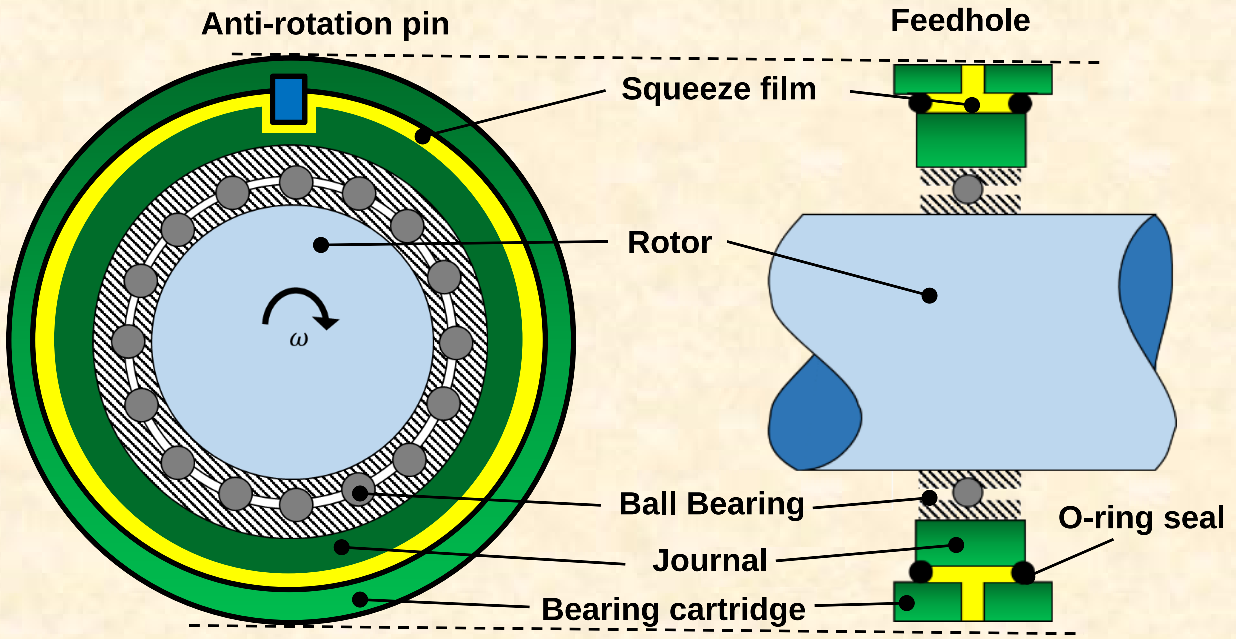
SFDs must be designed with consideration of the entire rotor-bearing system.

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

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

SFD performance is a function of

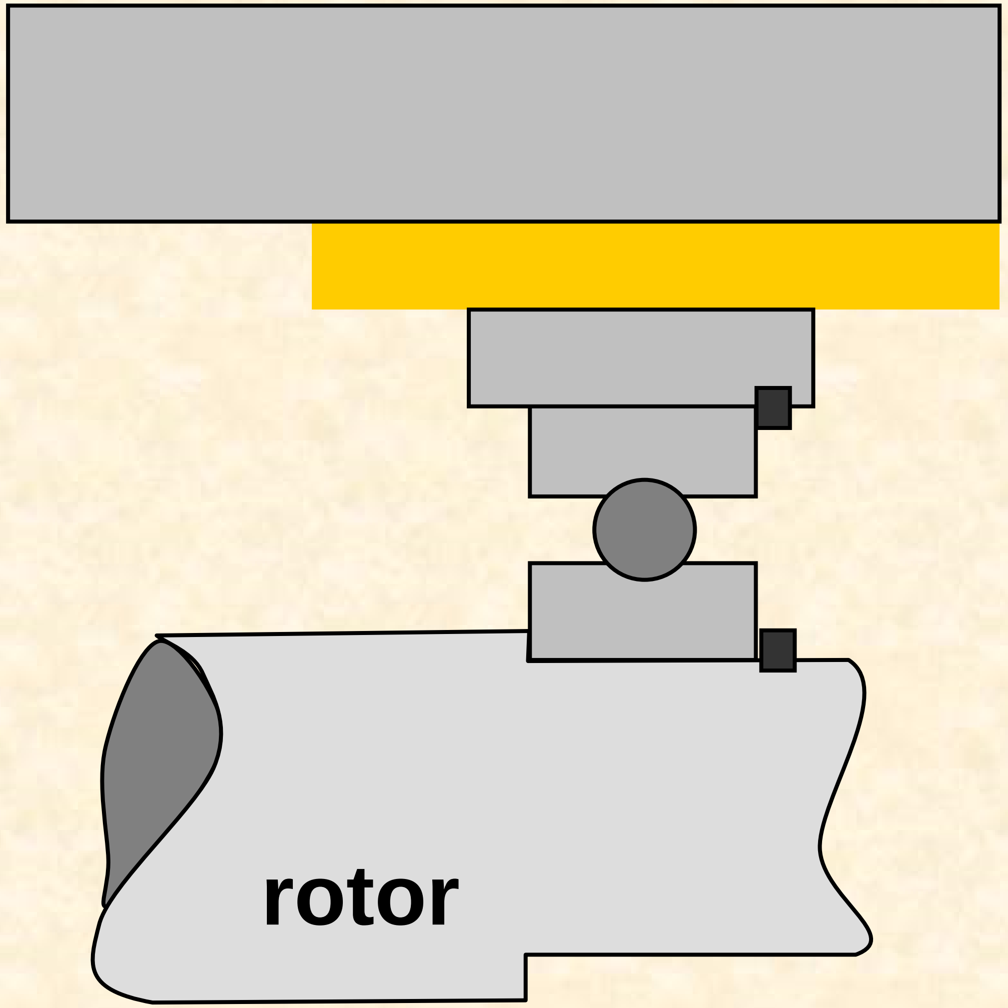
a) **Kinematics of rotor motion**
b) **Geometry** (land length, clearance, diameter)
c) **Lubricant** (density, viscosity)
d) **Supply pressure and discharge**
e) **Oil delivery and sealing mechanisms**
f) **Operating speed** (frequency)
g) **Flow Regime** (lubricant cavitation: gaseous or vapor, air ingestion and entrapment)



Industry still relies on analyses that models SFDs as a simple version of a hydrodynamic journal bearing.

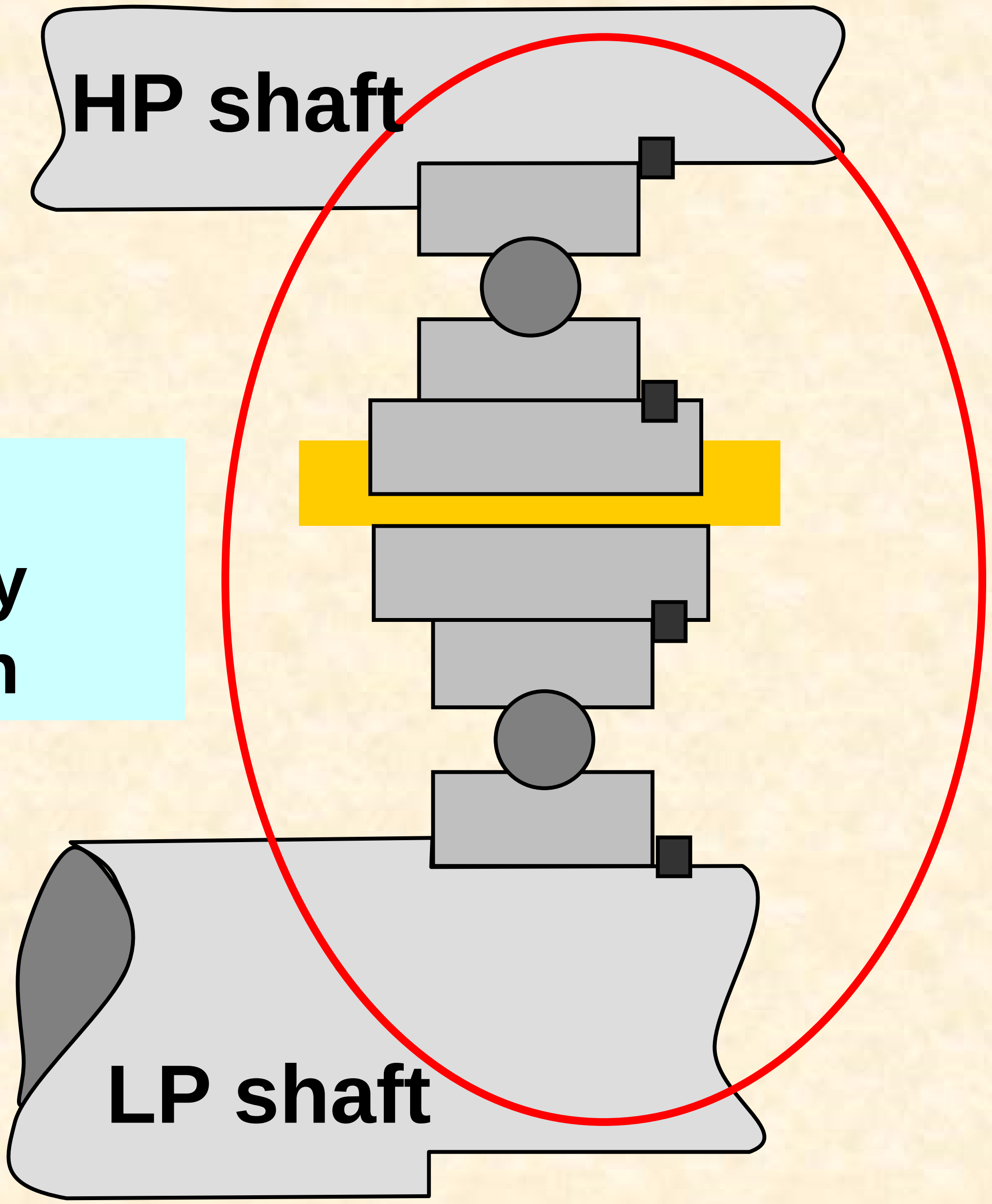
How does a SFD work? What is important?

Single frequency excitation

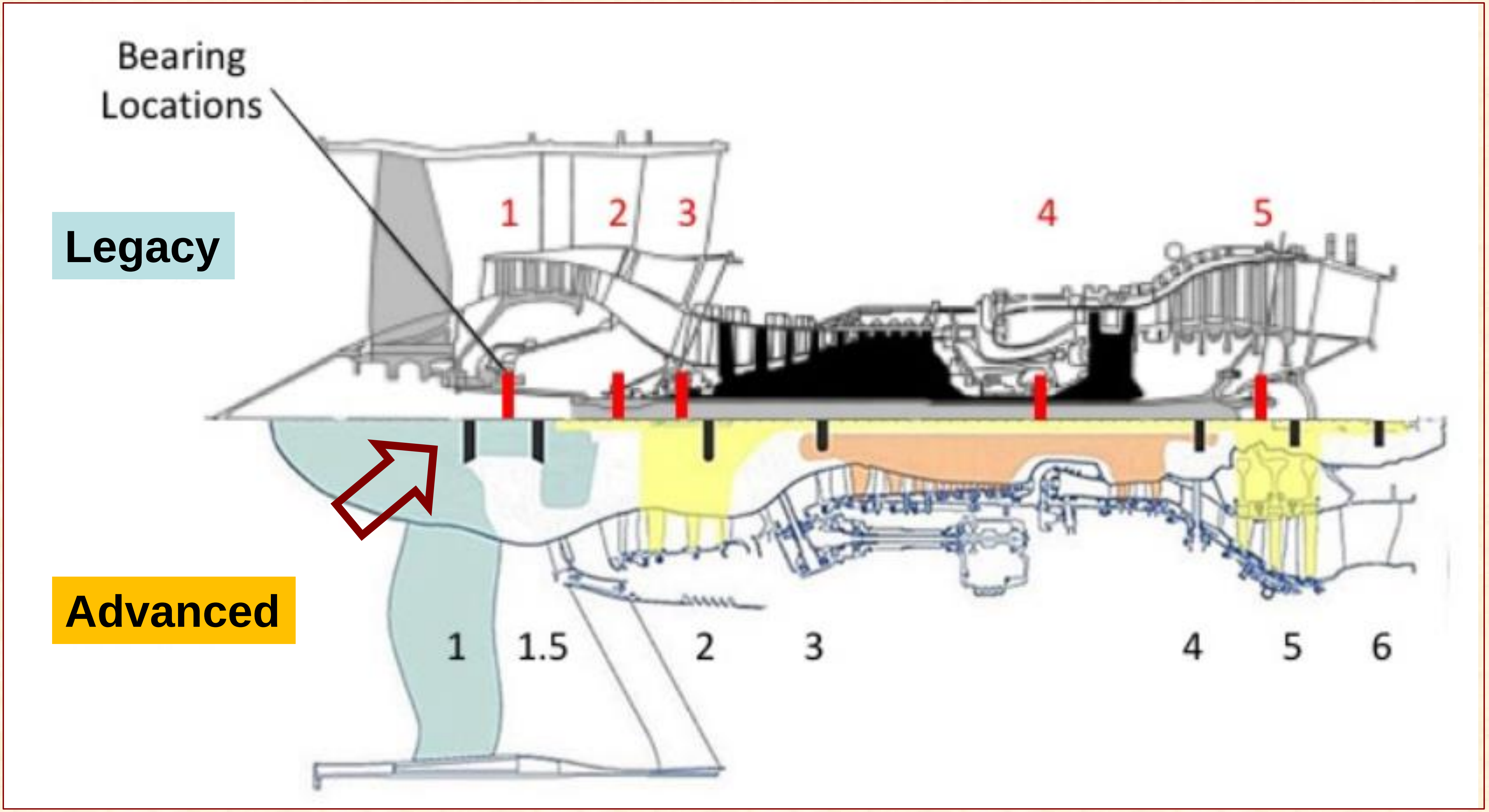


Multiple frequency excitation

Aircraft two spool rotor



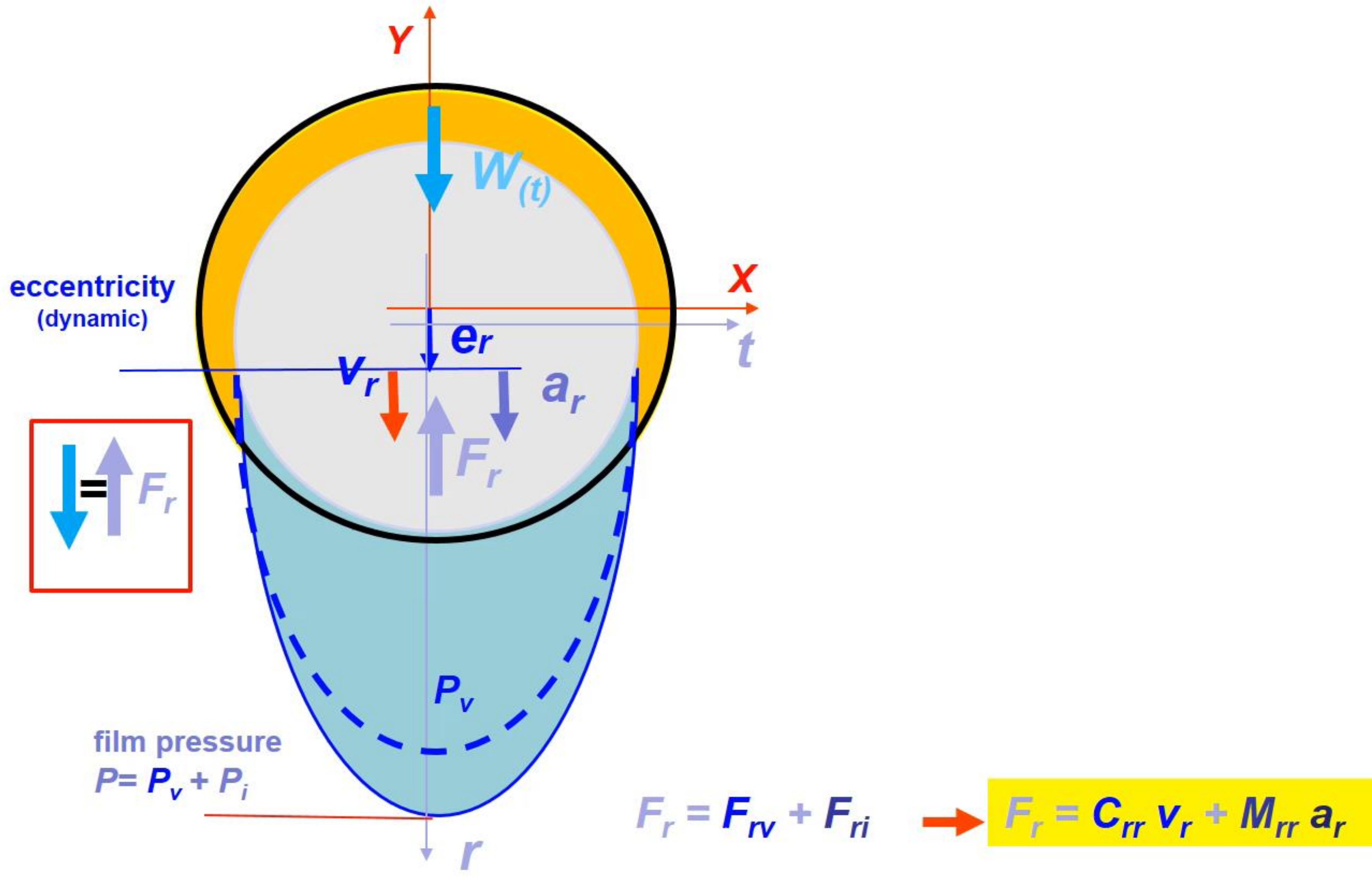
Bearings in modern aircraft



Slender & longer rotors demand more bearing supports. 4-6 SFDs are common.

Spakovszky. Z., 2021, "Instabilities Everywhere! Hard Problems in Aero-Engines." ASME Paper GT2021-60864

SFD basics: pure squeeze (plunging motion)



Pressure field *(P)* changes with time. ***P*** **~ squeeze velocity (v_r) and acceleration (a_r).** **Note pressure reversals.**

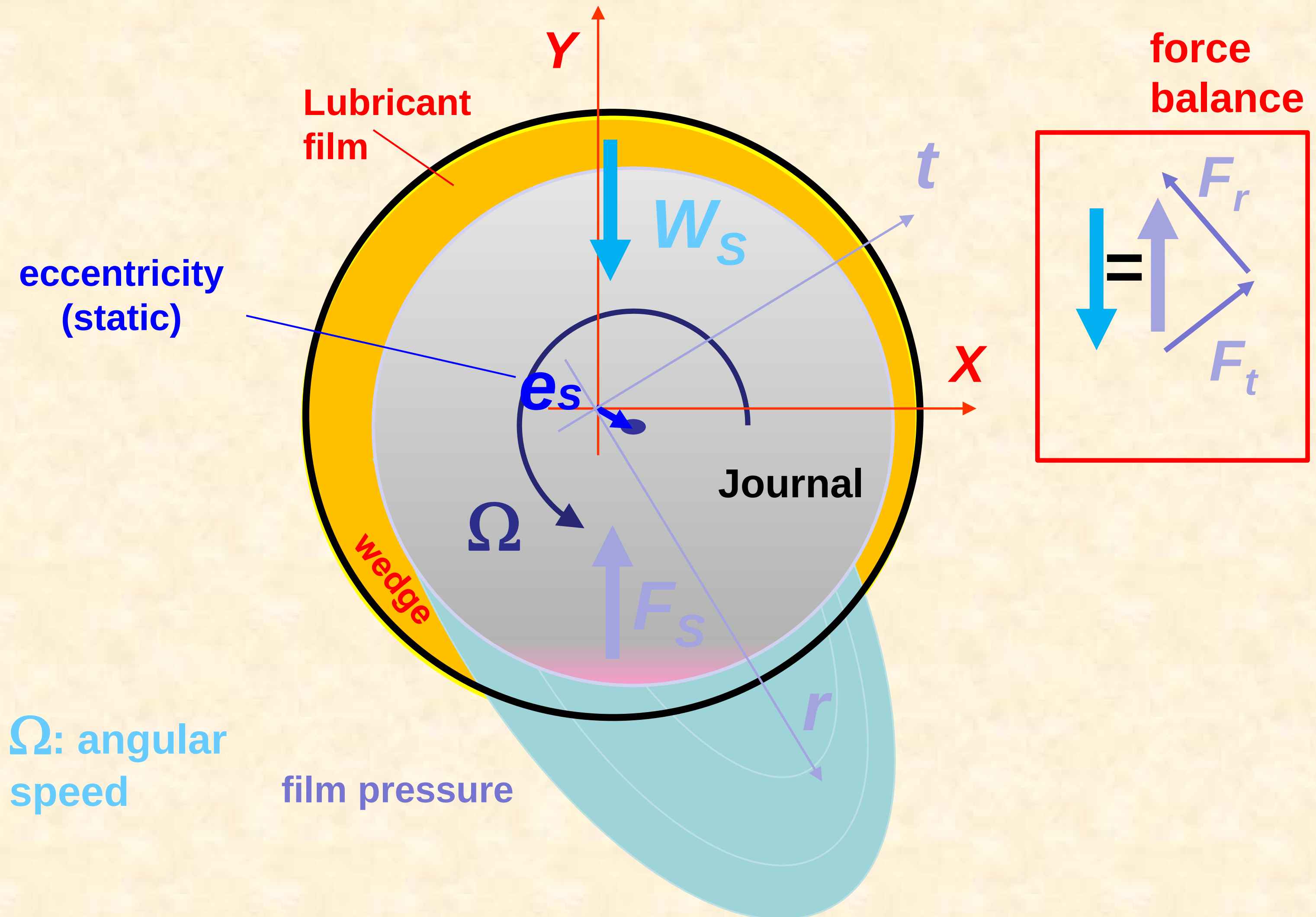
$$v_r = f_{(t)}$$
$$v_t = a_t = 0$$

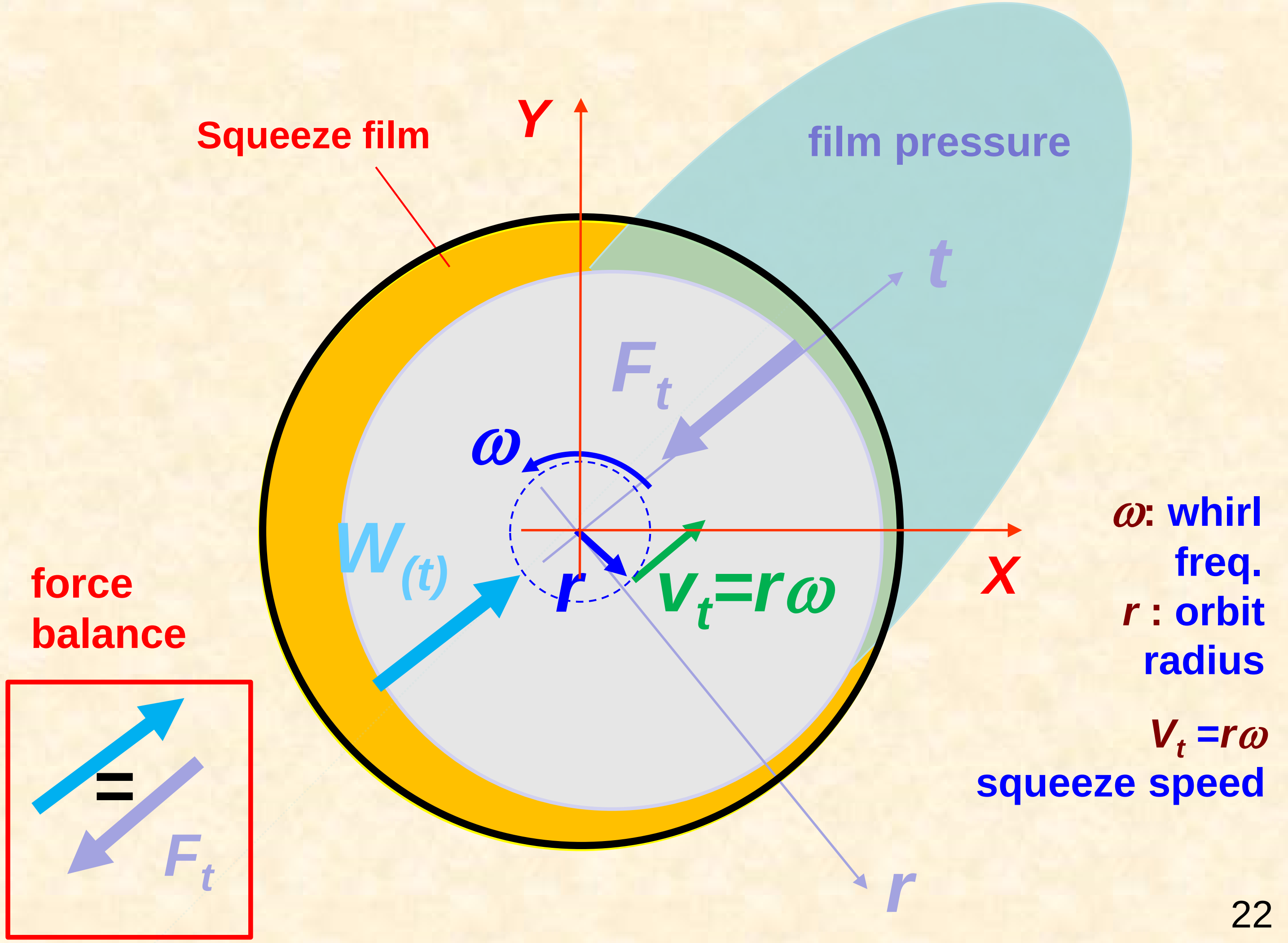
Is a SFD a non-spinning journal bearing (JB)?

SFD & JB may have a similar configuration.

In a JB, the shaft **spins** with angular speed (Ω)

In a SFD, journal center **whirls** or **precesses.**





Fluid inertia effect in a SFD; when is it important?

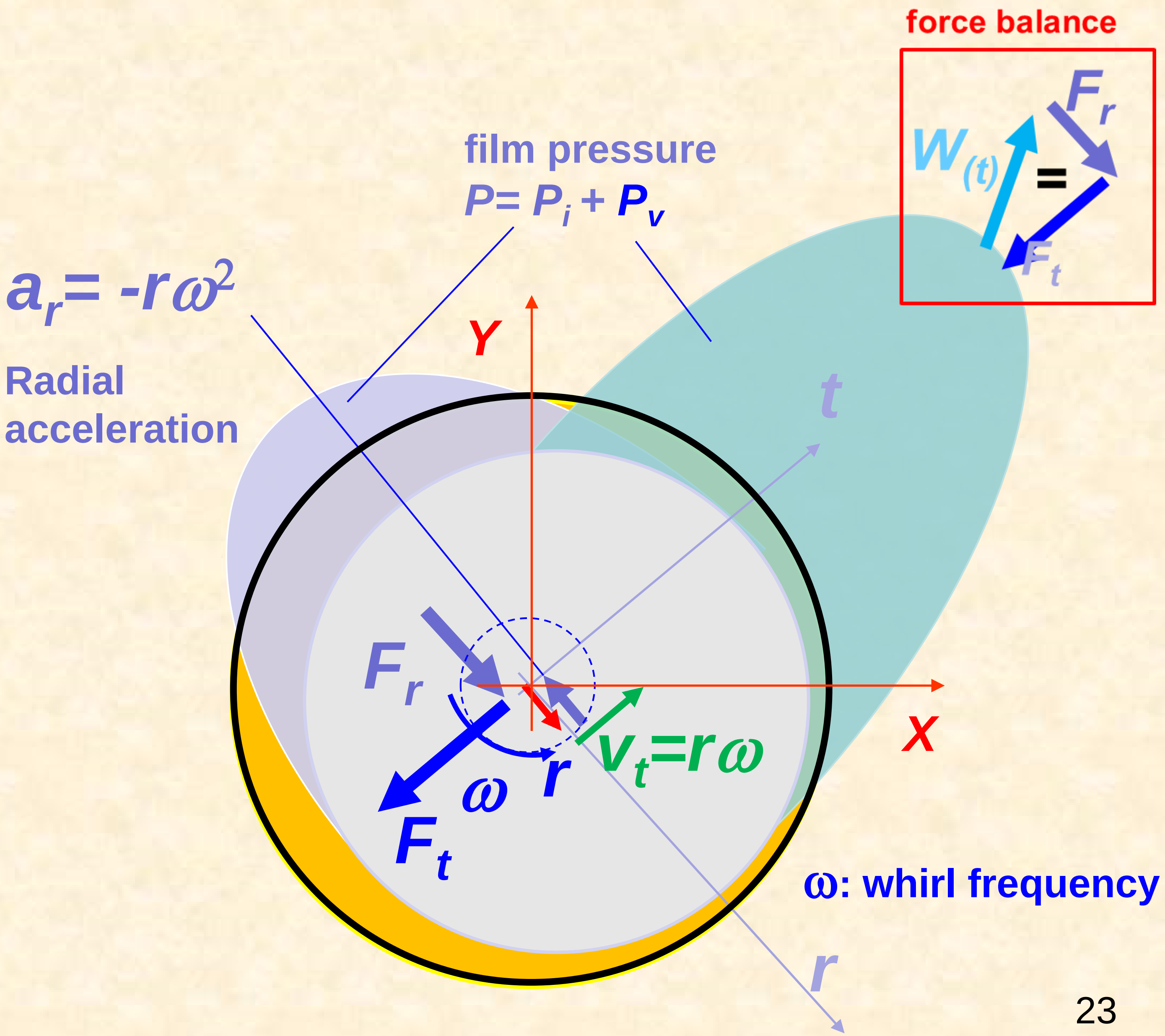
SFD with circular whirl. Fluid inertia generates a pressure field (P_i)

- **If change in whirl speed is fast, $|a_r| >> 0$, the damper reacts with a force larger than the viscous (damping) force.**

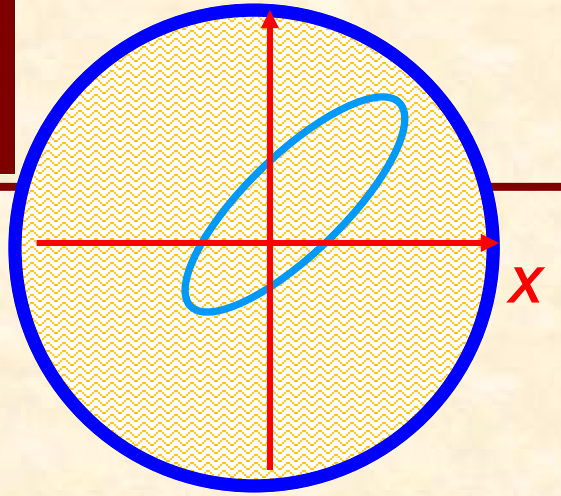
$$F_r = F_{ri} \rightarrow = -M_{rr}\, a_r$$

$$F_t = F_{tv} \rightarrow = -C_{tt}\, v_t$$

Reynolds number $Re_s = \rho \omega c^2 / \mu > 12$



SFD force coefficients



Rotordynamics models the SFD reaction force $F=\{F_X,F_Y\}^T$ with constant force coefficients → damping C & inertia M.

$$-\mathbf{F}_{(t)} = \mathbf{C}\,\dot{\mathbf{z}} + \mathbf{M}\,\ddot{\mathbf{z}}$$

$\mathbf{z}=\{x,y\}^T$: **rotor (journal) displacement** about static position (e).

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

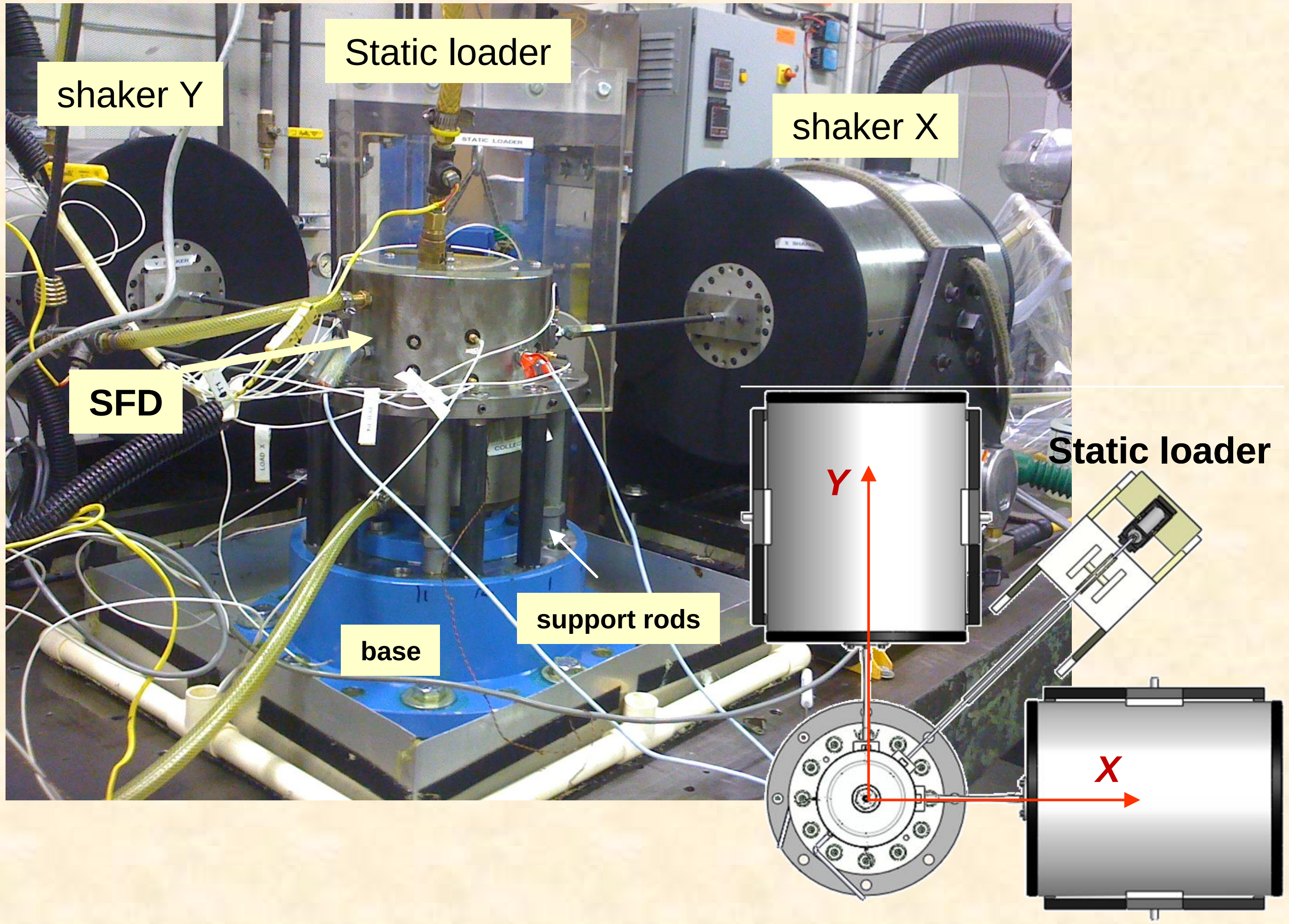
SFDs cannot generate stiffness K

USED for prediction of synchronous rotordynamic response and rotor-bearing system stability.

SFD Test Program

Explore **novel SFD designs** & benchmark **SFD empirical data**.

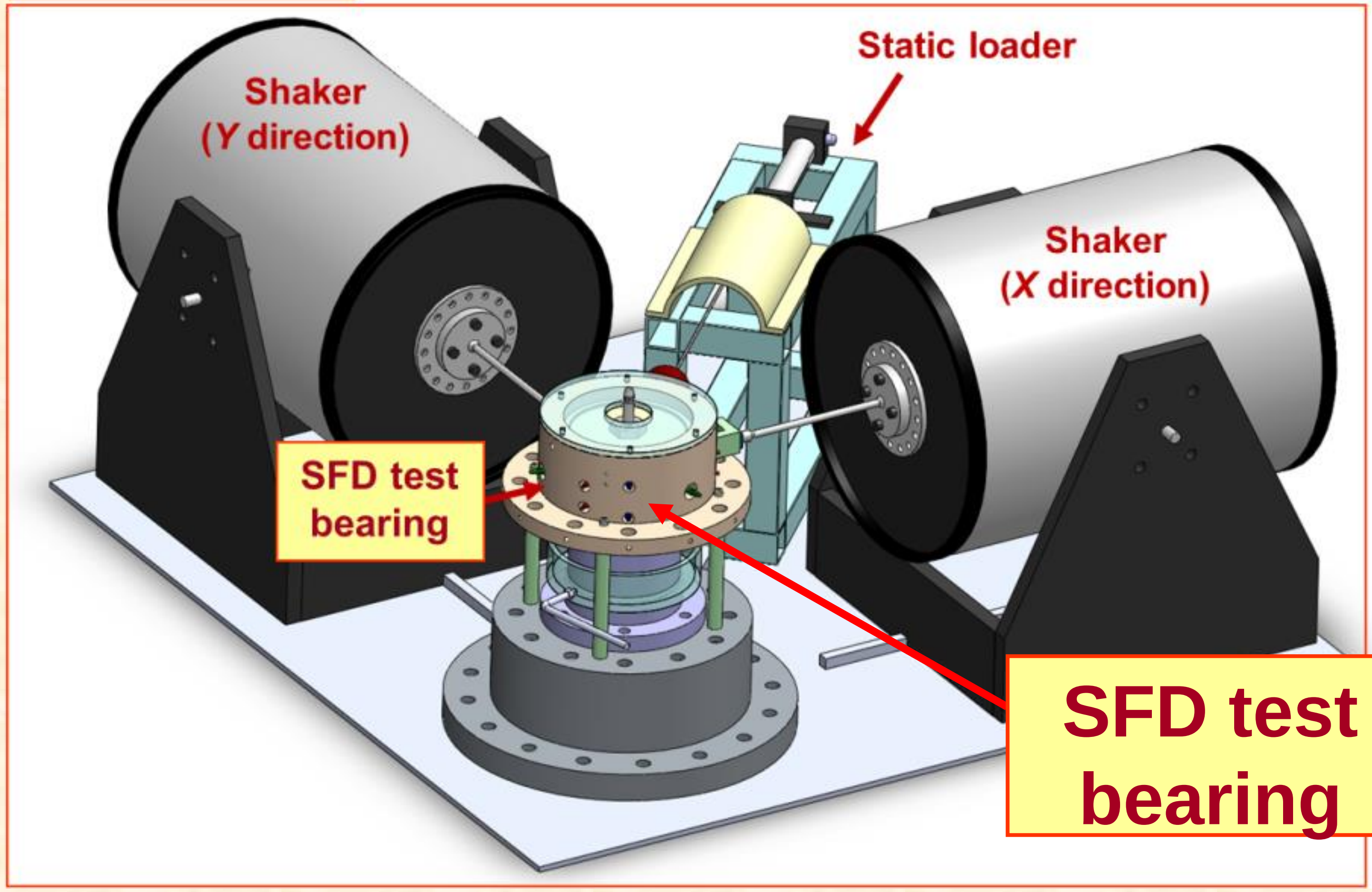
Develop & **validate SFD forced performance** model & improve its prediction.



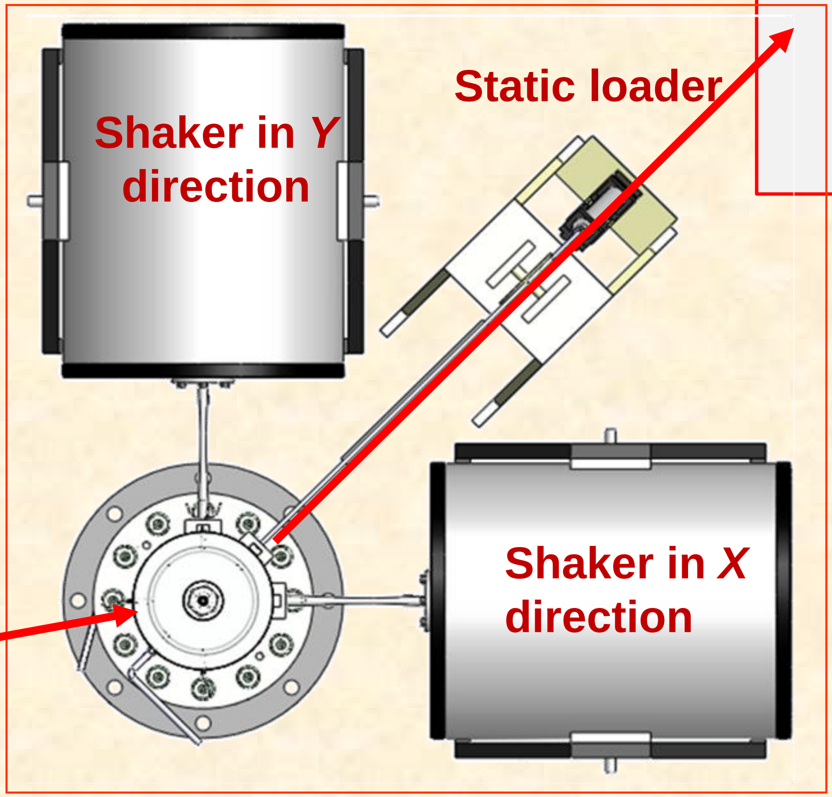
Funded by Pratt & Whitney Engines (2008-2018)

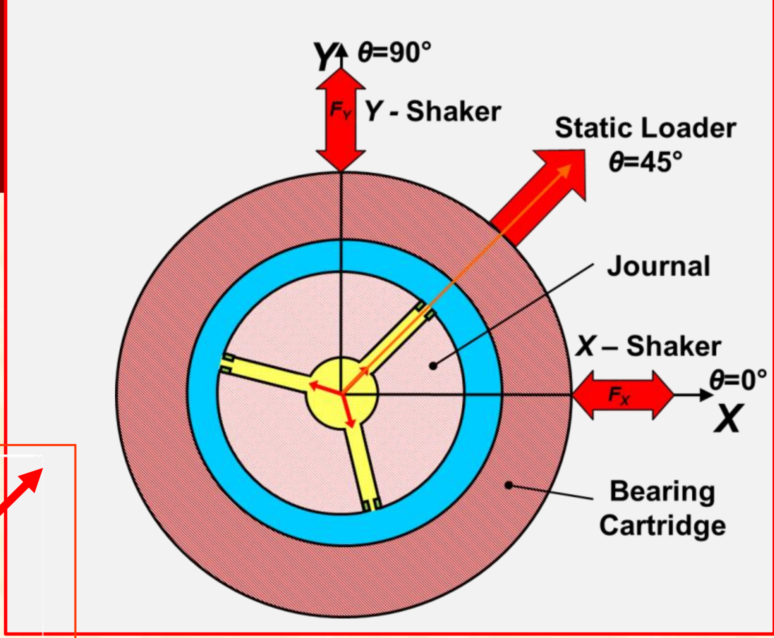
SFD test rig (2008-2018)

Isometric view



Top view



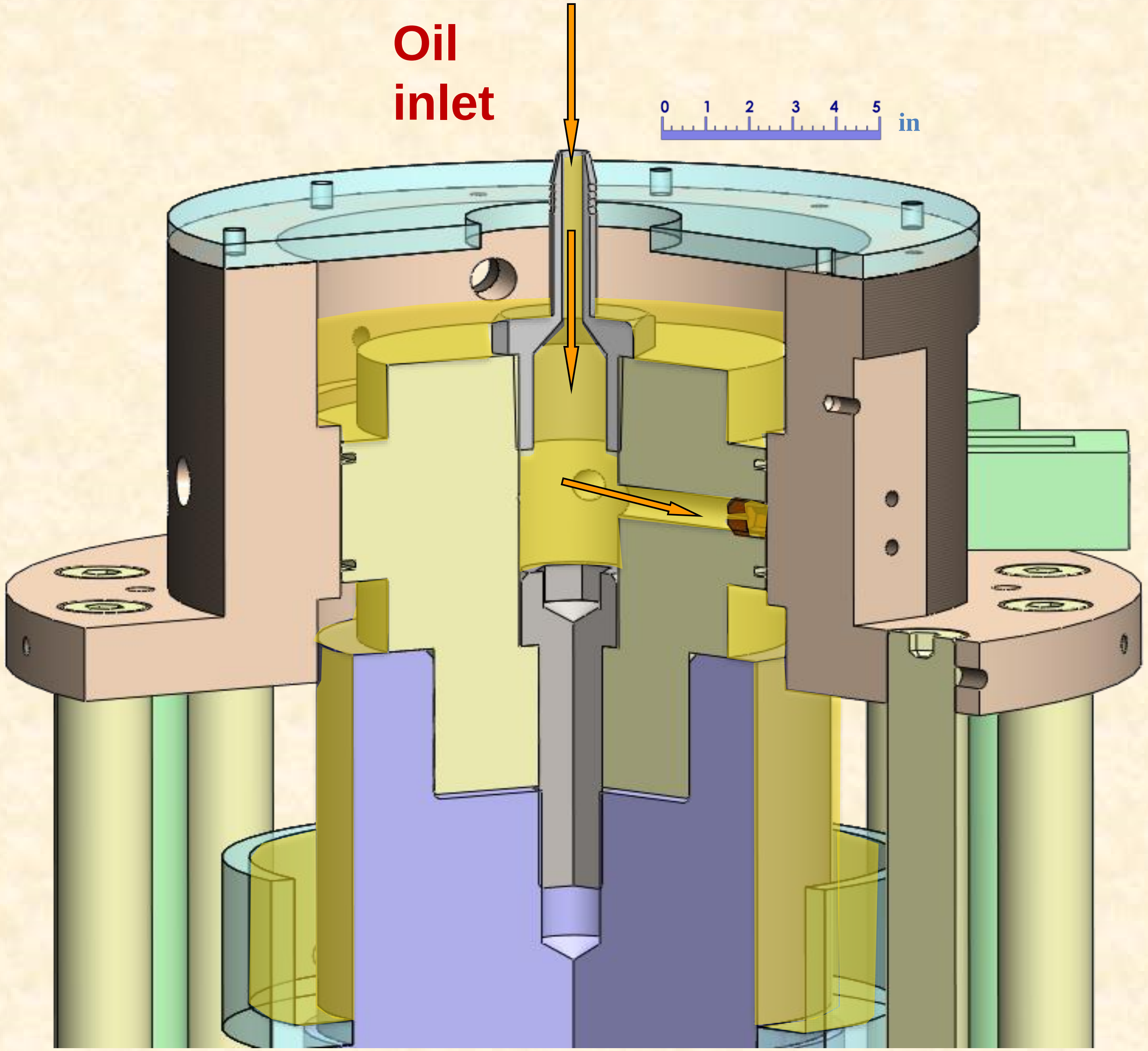


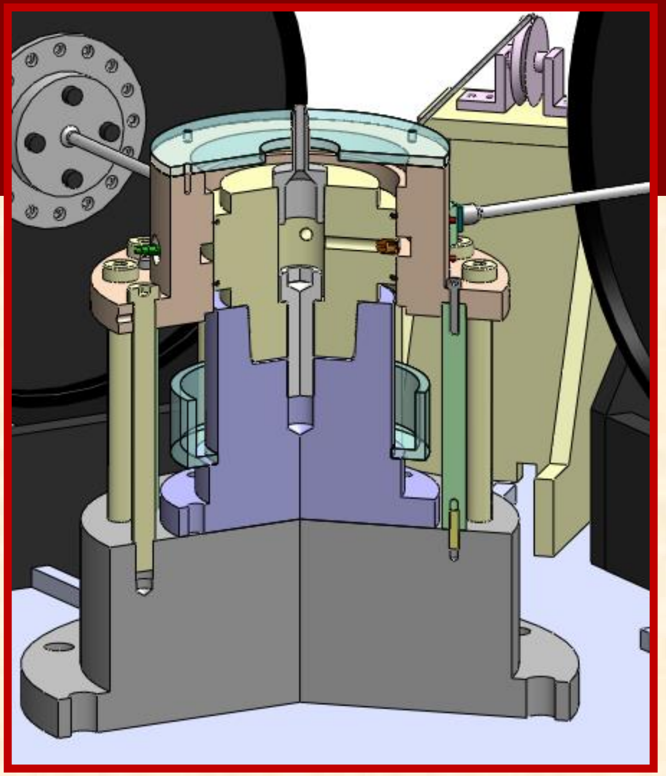
2 electro magnetic-shakers (2 kN ~ 550 lb_f)

Static loader (4 kN ~ 1 klbf) at 45°

Customizable SFD test bearing

Lubricant flow path



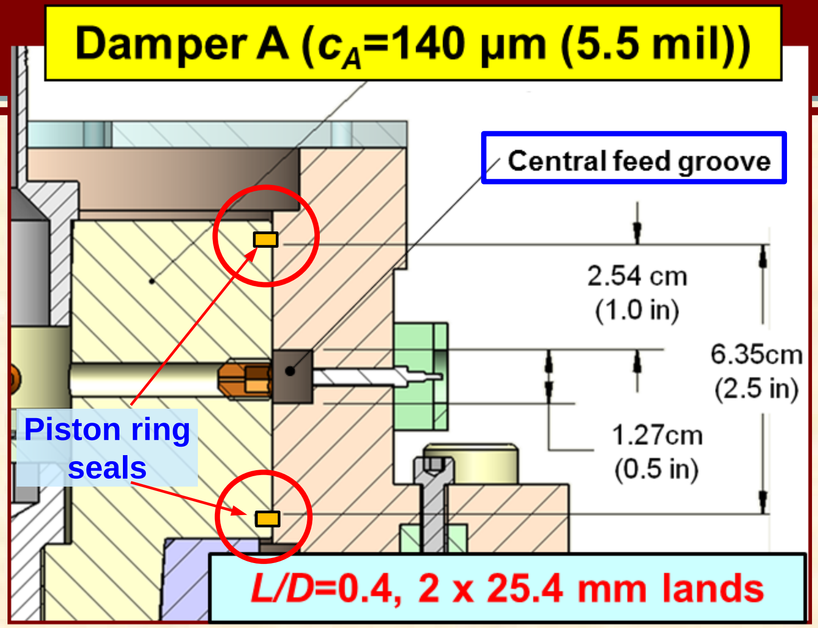


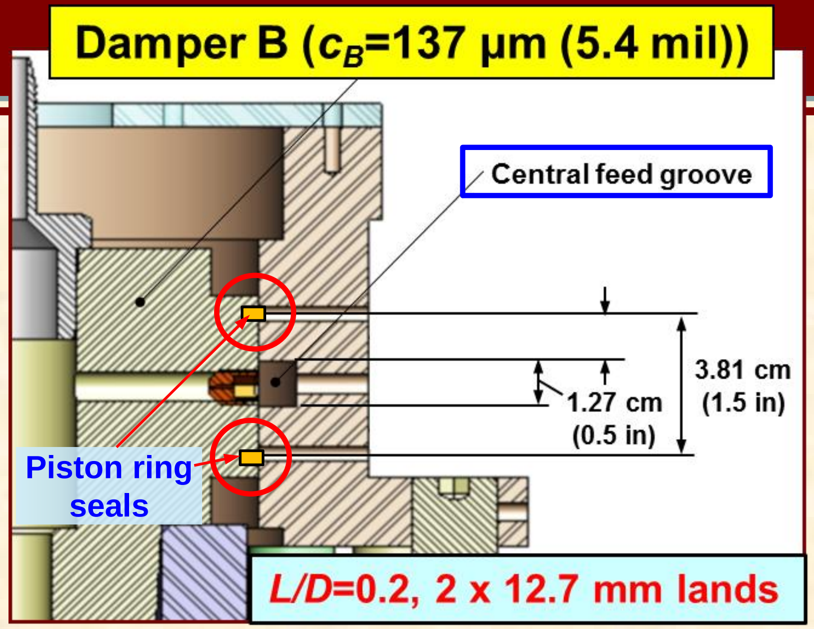
Oil physical properties similar to those in jet engines operating at high temperature.

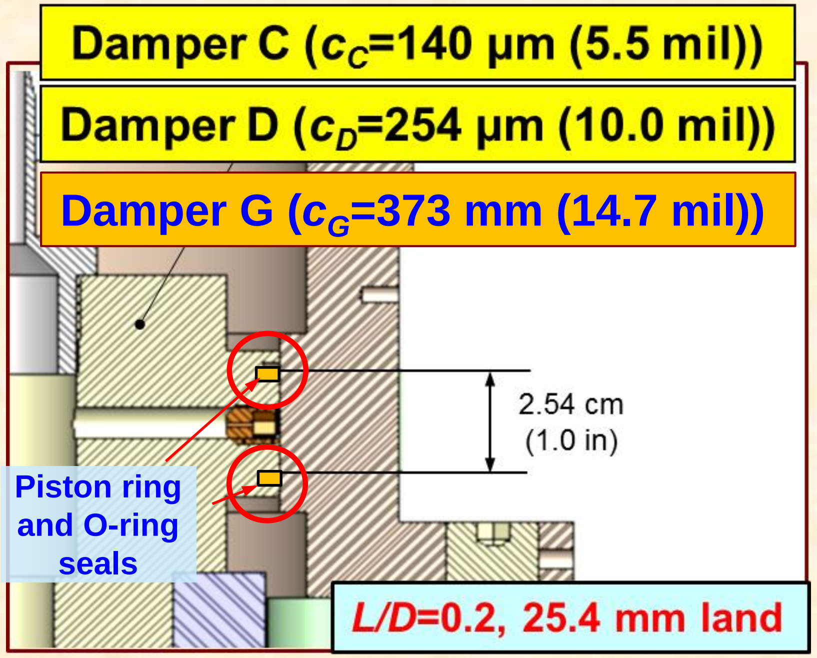
ISO VG 2 oil

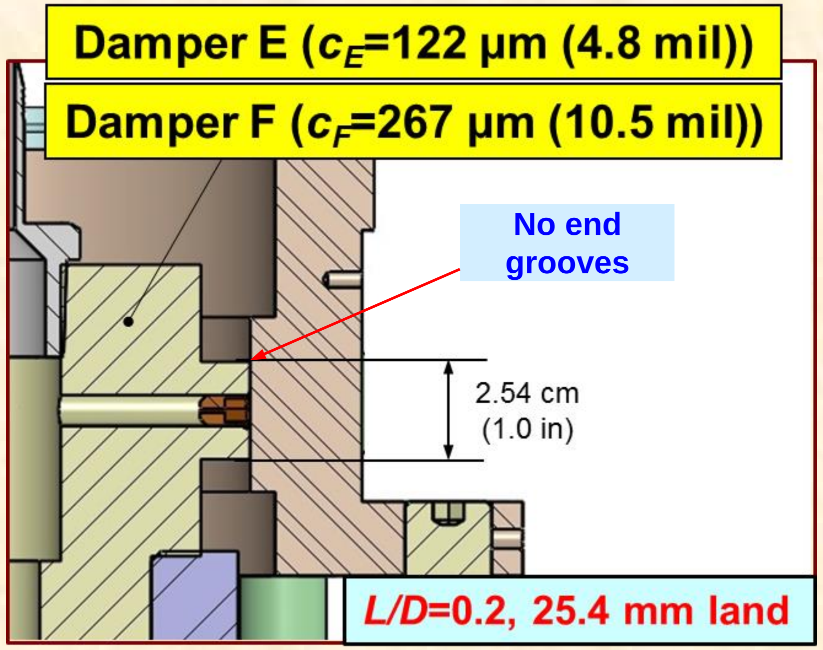
Oil inlet temperature, T_s = 23 °C
Density, ρ = 800 kg/m^3
Viscosity μ at T_s= 2.6 cPoise
Flow rate, Q_{in}= varies

To explore









novel SFD designs

& benchmark SFD empirical data & develop & validate SFD forced model.

Optimize SFD influence on rotor dynamics.

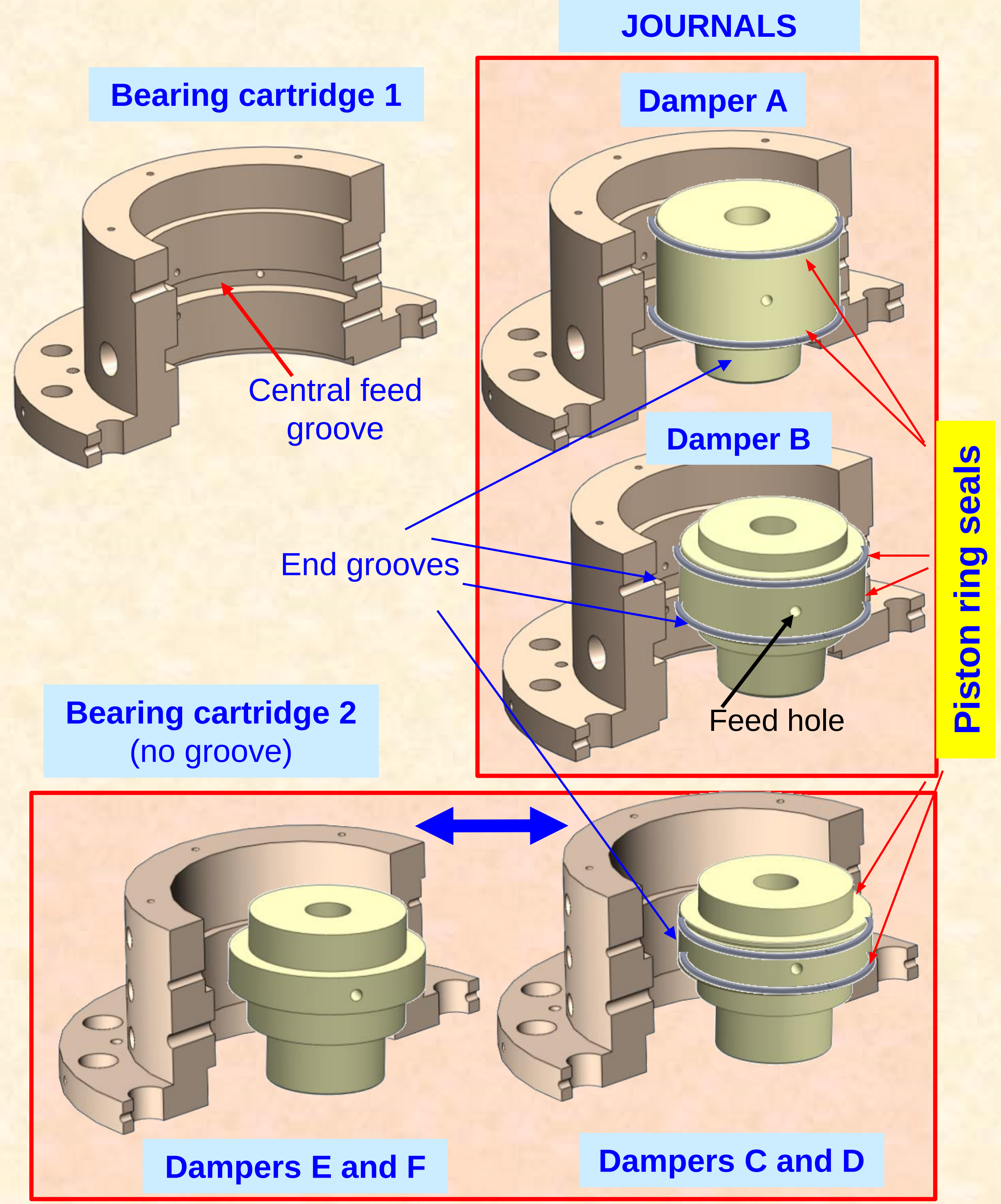
→ 25+ papers, computational tool validated by test data, and countless tech reports to sponsor.

SFD test configurations

Ultra short length desired

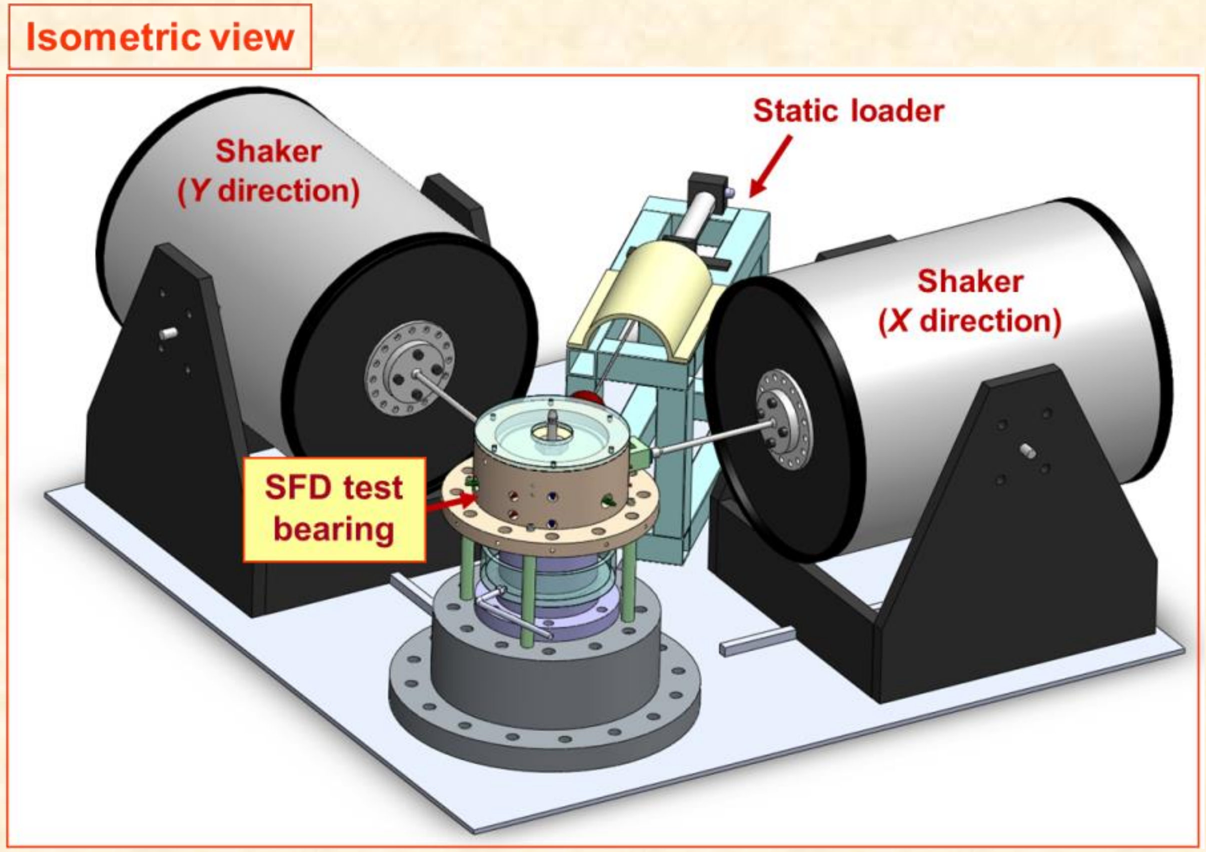
Configuration	Land Length, **Central Groove**	End Grooves	**Radial Clearance**
A*	2 X L=25.4 mm film lands - **Central groove**	Yes	c_{A-1}=141 μm c_{A-2}=251 μm
B*	2 X L=12.7 mm film lands - **Central groove**		c_B=138 μm
C*	L=25.4 mm film land - **No** feedl groove	Yes	c_C=130 μm
D*			c_D=254 μm
E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

*** SFD: open ends and sealed ends: piston rings or Orings**



Experimental estimation of force coefficients

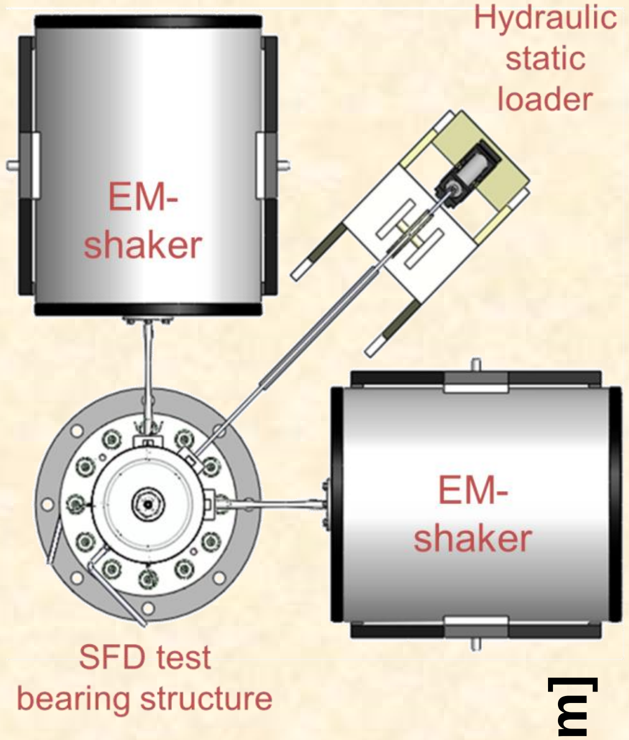
Designed and built by students at TAMU



Bearing mass, M_{BC}	15 kg
Structure stiffness, K_s	10 MN/m
damping, C_s	0.9 kN-s/m

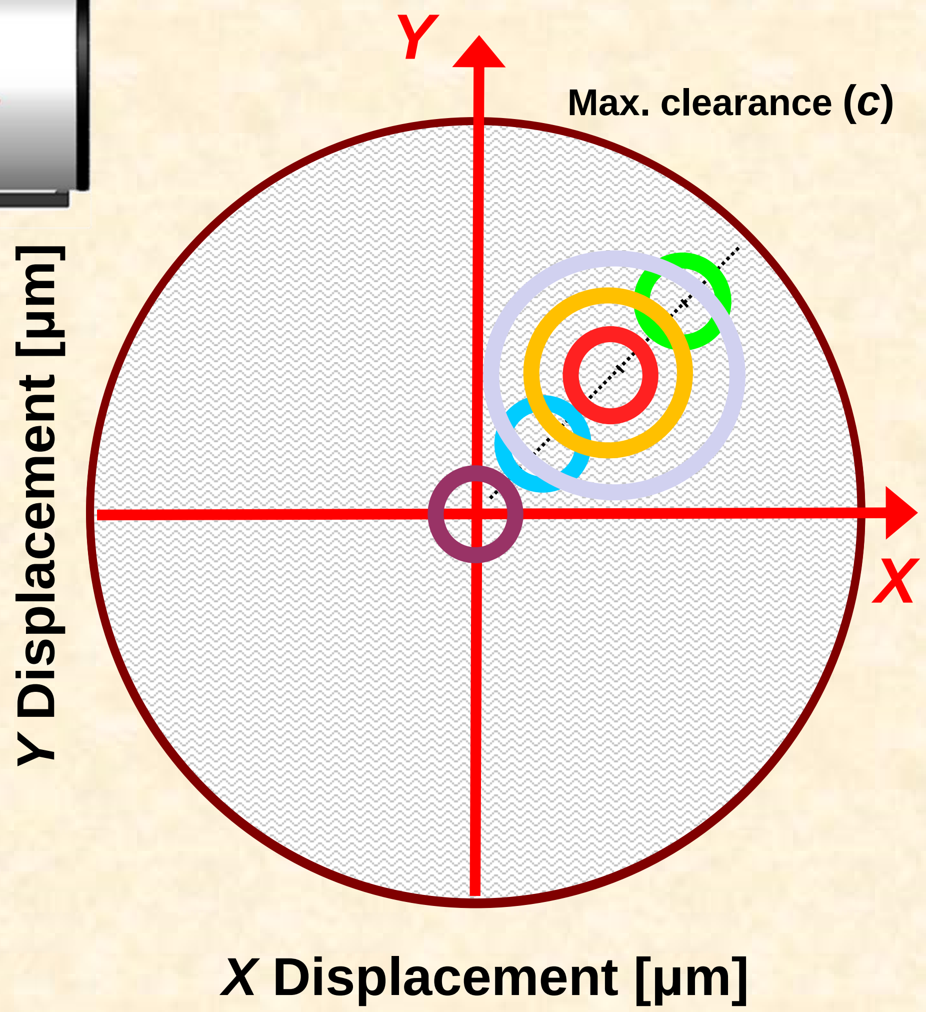
f_n=131 Hz, ζ=0.03

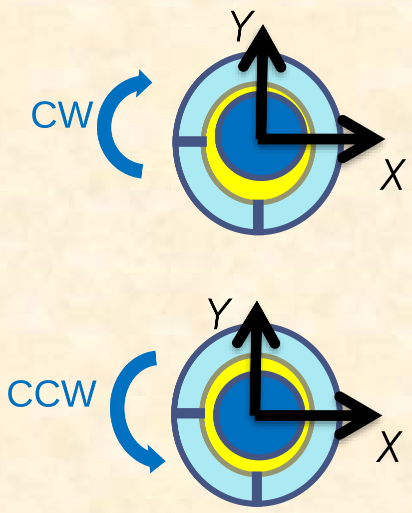
Imposed whirl motions



Evaluate SFD force coefficients from whirl orbits: amplitude (*r*) grows.

with offset or static eccentricity (e_s) – 45^o away.



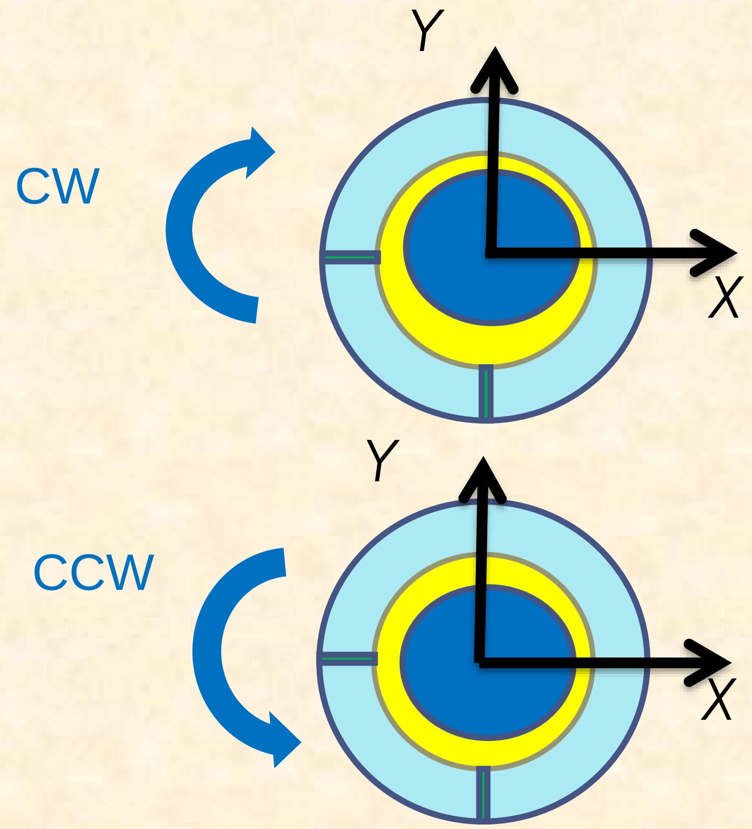


Measurement procedure and identification

Step 1 : Apply loads and measure BC motions

Shakers apply forces

$$\mathbf{F}^1 = \mathrm{Re}\left(\begin{bmatrix} F_X{}^1 \\ iF_Y{}^1 \end{bmatrix} e^{i\omega t}\right) \qquad \mathbf{F}^2 = \mathrm{Re}\left(\begin{bmatrix} F_X{}^2 \\ -iF_Y{}^2 \end{bmatrix} e^{i\omega t}\right)$$



Record BC displacements and accelerations

$$\mathbf{z}^1 = \begin{bmatrix} x^1_{(t)} \\ y^1_{(t)} \end{bmatrix} = \begin{bmatrix} X^1 \\ Y^1 \end{bmatrix} e^{i\omega t} \qquad \mathbf{z}^2 = \begin{bmatrix} x^2_{(t)} \\ y^2_{(t)} \end{bmatrix} = \begin{bmatrix} X^2 \\ Y^2 \end{bmatrix} e^{i\omega t}$$

$\mathbf{a}^1$ $\qquad$ $\mathbf{a}^2$

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

EOM: Frequency Domain

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

Unknown Parameters:

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

Identification of parameters

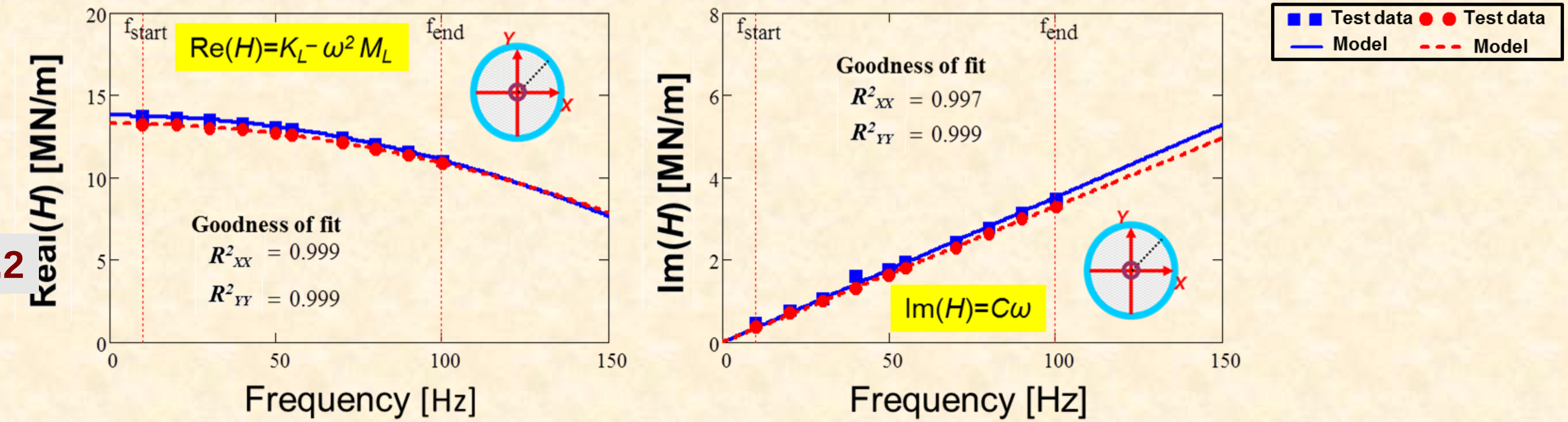
Step 2 : Transform to frequency domain and curve fit H_L's

Complex dynamic stiffness

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

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

r/c_A=0.2



TYPICALLY physical model $\mathrm{Re}(H_{XX}) \rightarrow K\text{-}\omega^2 M$ and $\mathrm{Im}(H_{XX}) \rightarrow C\omega$ agree with experimental data. Damping C is constant over frequency range

SFD coefficients

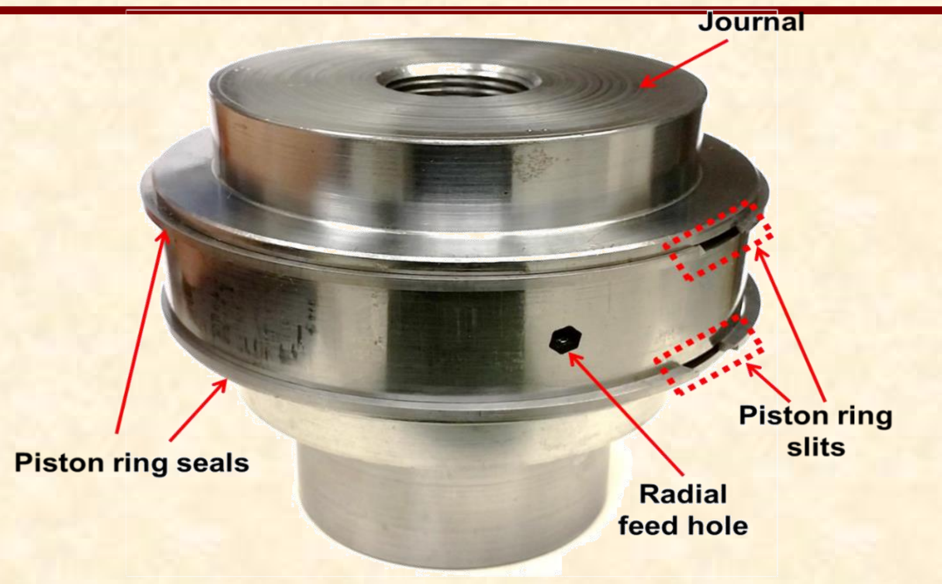
$$(K, C, M)_{SFD} = (K, C, M)_L - (K, C, M)_S$$

SFD

Test system (lubricated)

Dry structure

Comprehensive flow model for prediction of SFD forced performance



Bubbly mixture (2001, 2019) and orbit-model (2016) …… with a major departure!

San Andrés, L., and Koo, B., **2019**, "Model and Experimental Verification of the Dynamic Forced Performance of a Tightly Sealed Squeeze Film Damper Supplied with a Bubbly Mixture," ASME 2019-90330

San Andres, L., and Jeung,S-H, **2016,** "Orbit-Model Force Coefficients for Fluid Film Bearings: A Step Beyond Linearization," ASME J. Eng. Gas Turb. Pwr., 138(2)

Diaz and San Andres, **2001,** "A Model for Squeeze Film Dampers Operating with Air Entrainment and Validation with Experiments**,**" ASME J. Tribol., 123.

Squeeze film pressure and boundary conditions

Extended Reynolds Eq.

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

Temporal fluid inertia

density

viscosity

Gas density — Oil density

$$\rho_m = \beta_{(P)}\,\rho_g + \left(1-\beta_{(P)}\right)\rho_{oil}$$

$$\mu_m = \beta_{(P)}\,\mu_g + \left(1-\beta_{(P)}\right)\mu_{oil}$$

Gas viscosity — Oil viscosity

Gas volume fraction GVF = β

$$\beta_{(P)} = \frac{1}{1+\dfrac{P_{(\Theta,z,t)}-P_v}{P_s-P_v}\left(\dfrac{1-\beta_s}{\beta_s}\right)}$$

See also ASME GT2022-81990

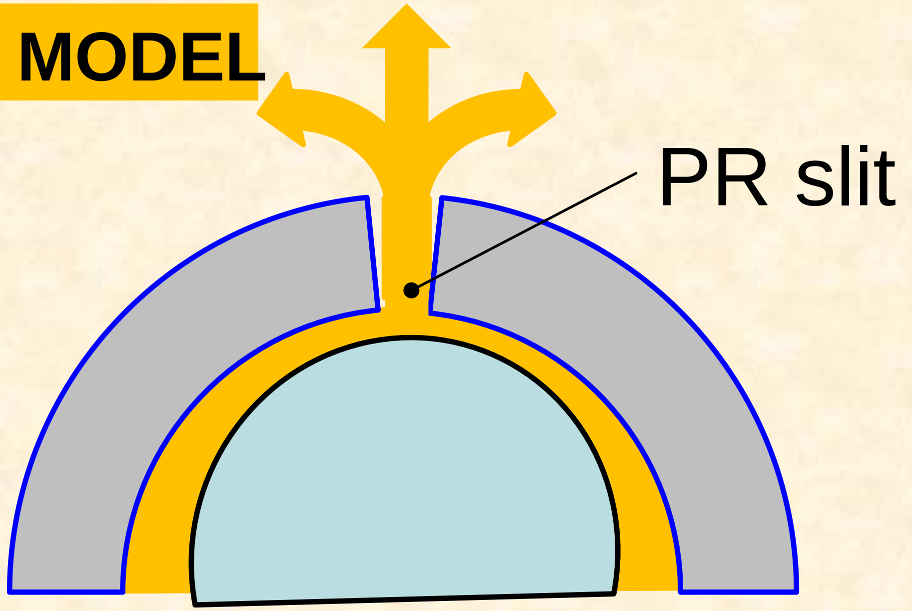
feedhole

$$M_{in} = C_{in}A_{in}\sqrt{2\rho\left|P_s - P_{(\theta_{in},0,t)}\right|}$$

PR-slit

$$M_{exit} = C_{slit}A_{slit}\sqrt{2\rho\left|P_{(\theta_{slit},\frac{L}{2},t)} - P_a\right|}$$

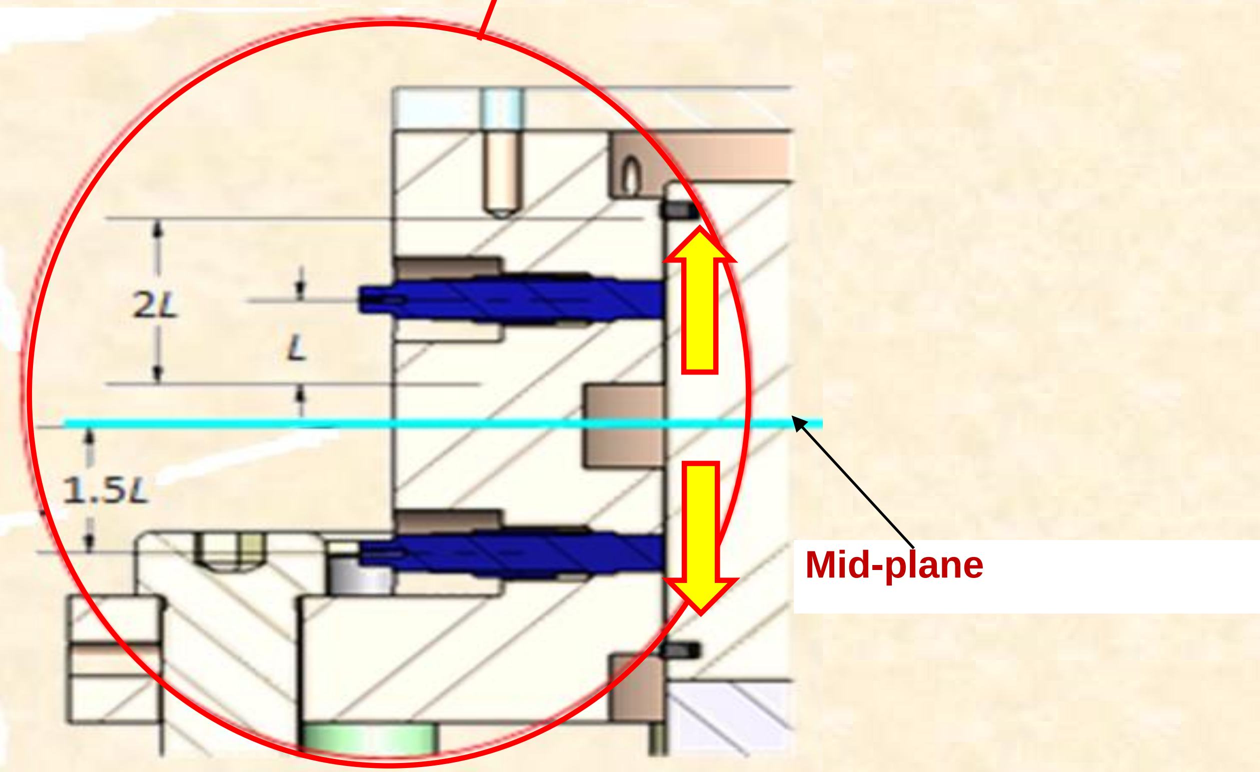
OLD MODEL



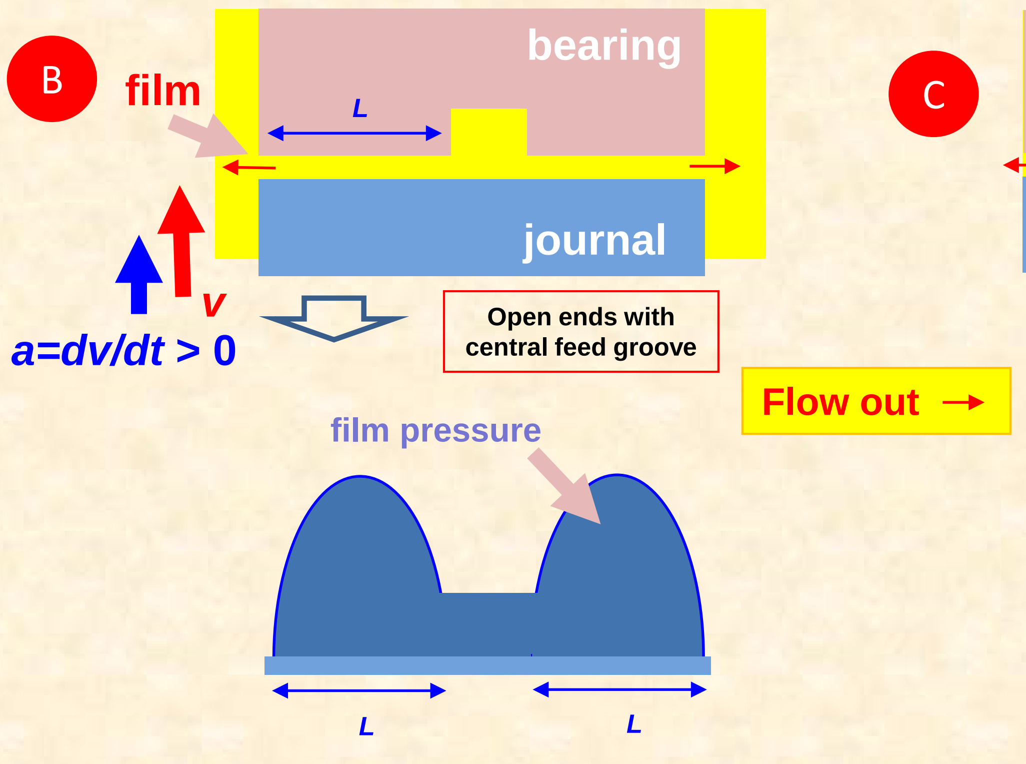
PR flow: a major departure from simple practice: $Q \sim C_{seal}\,\Delta P$

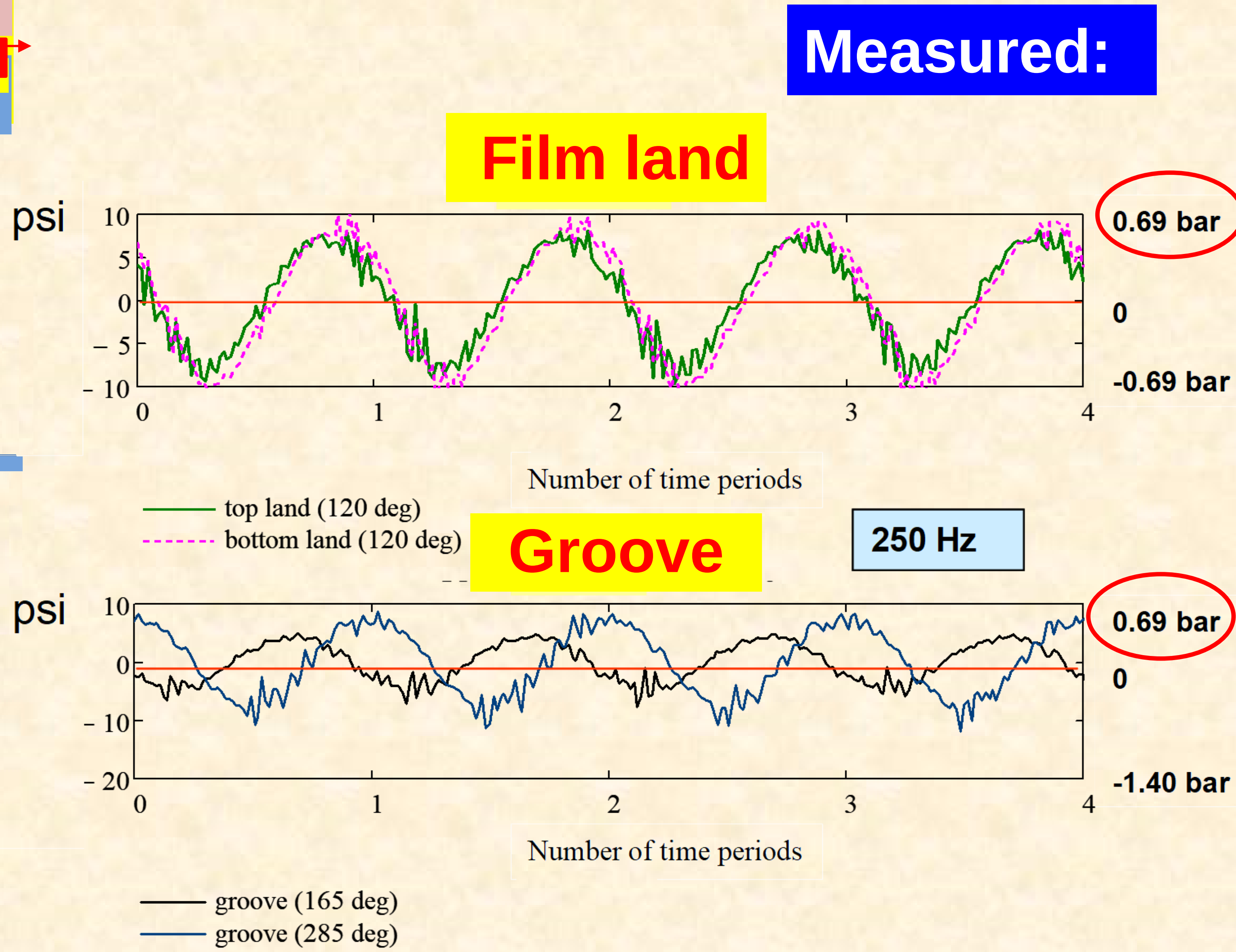
Do dynamic pressures appear in a deep feed groove?

Conventional knowledge asserts that deep grooves keep a <u>constant pressure</u> that pushes lubricant into the adjacent film lands.



Do deep grooves isolate film lands?

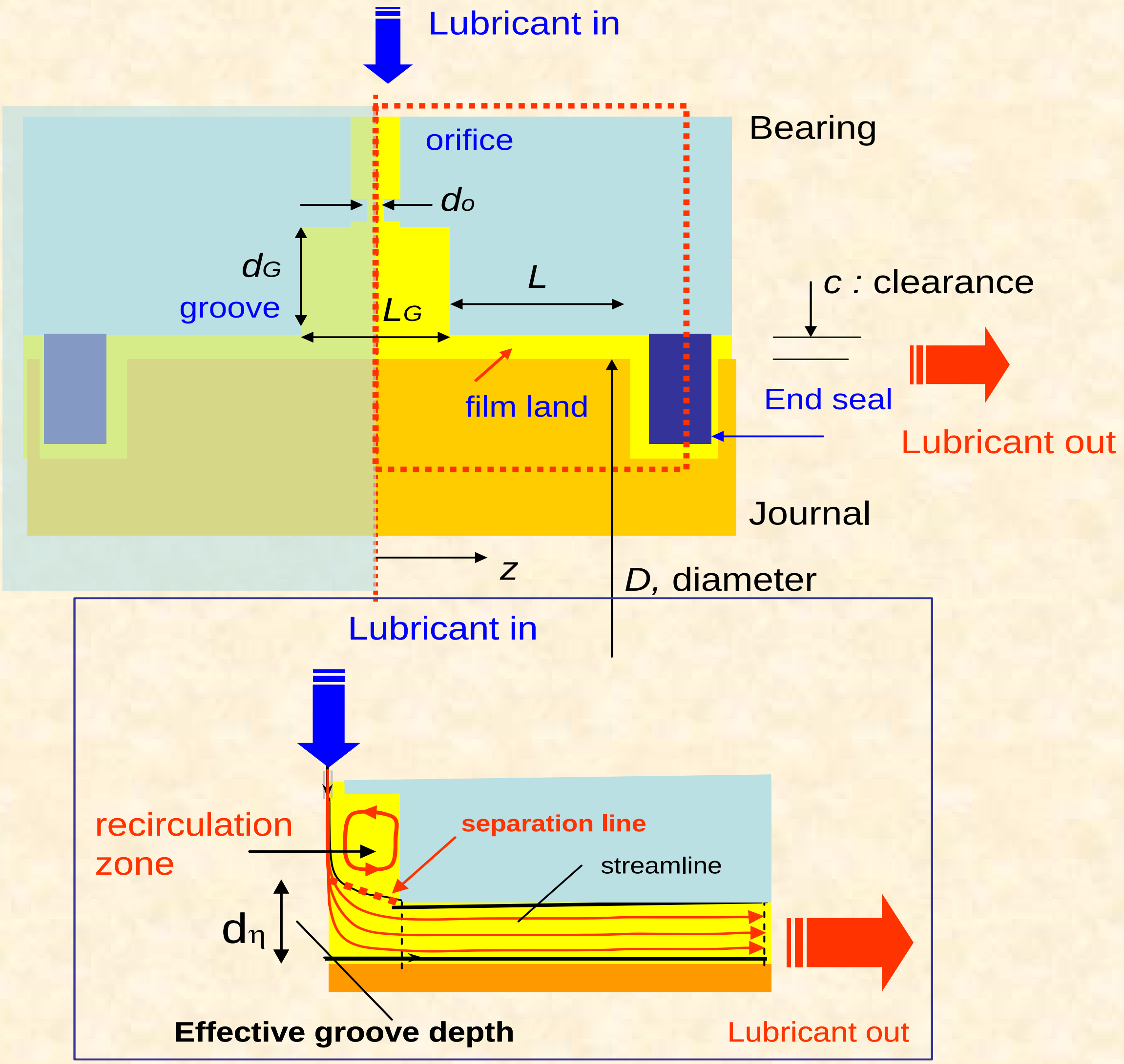




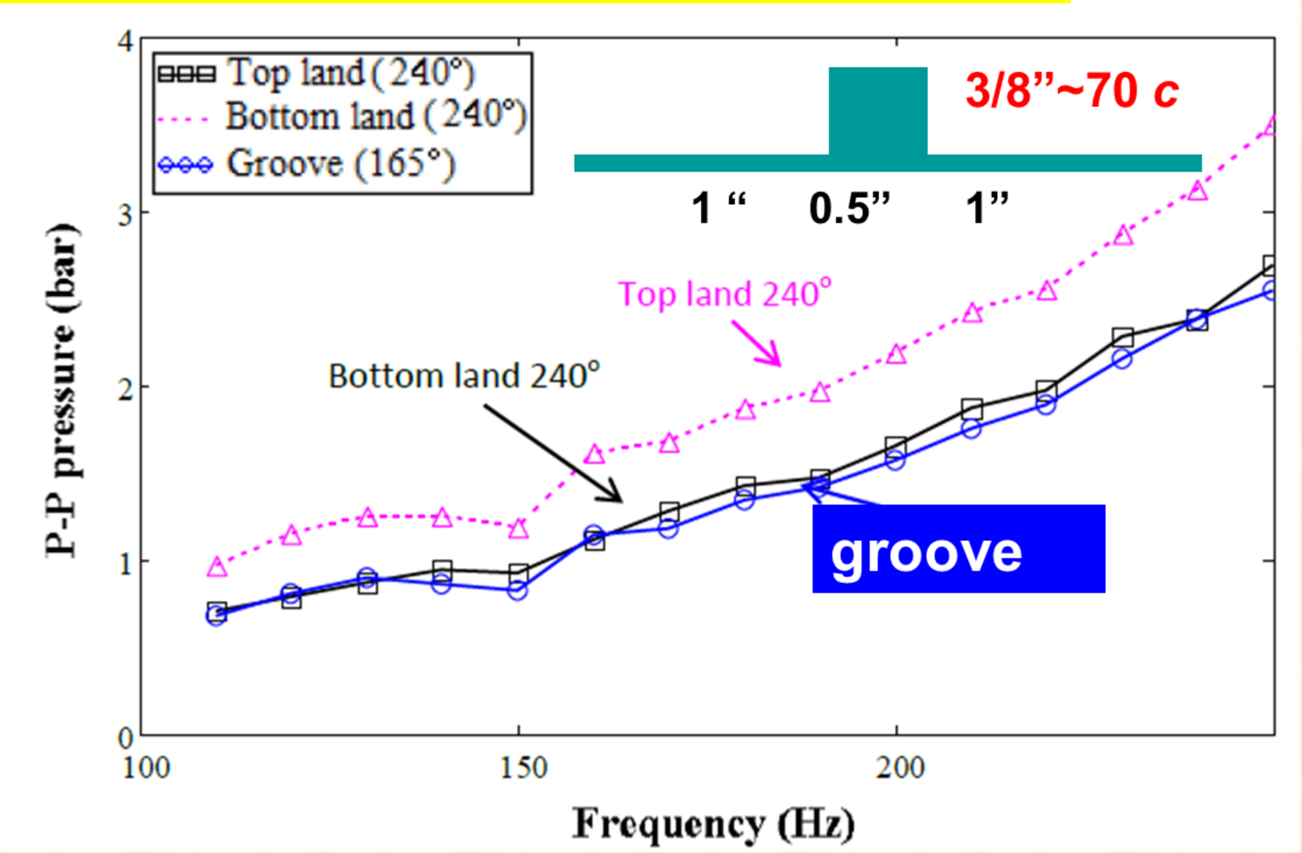
No! grooves lead to a significant squeeze film (pressure) action

Flow model SFD with grooves

San Andrés & Delgado, GT2011-45274



SFD with PR seals

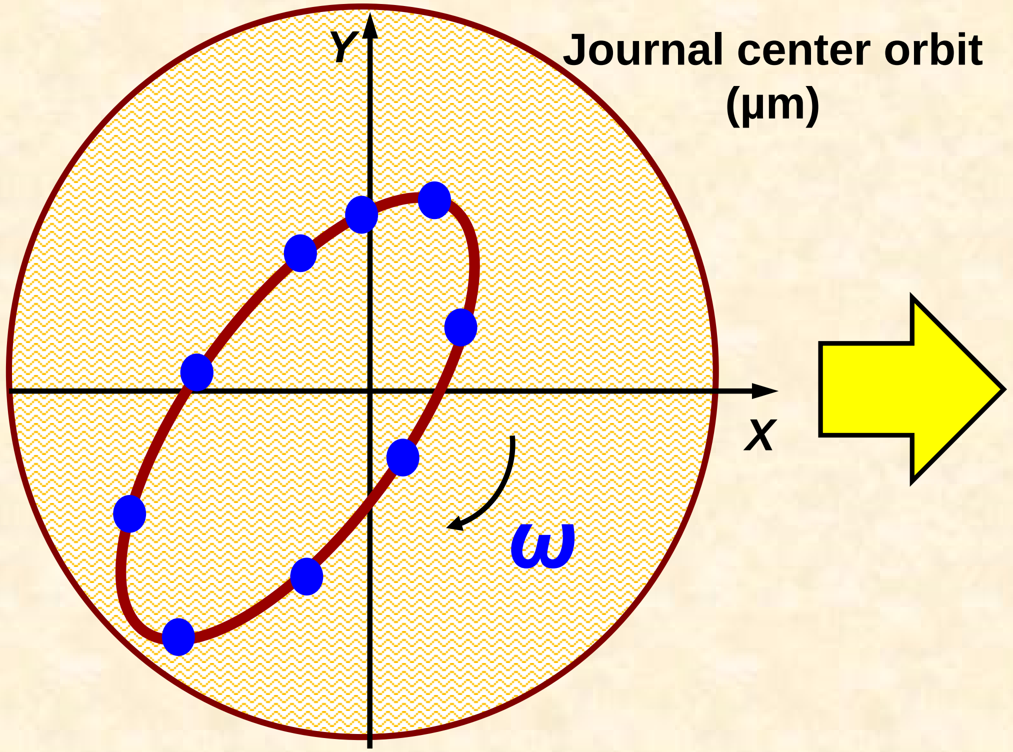


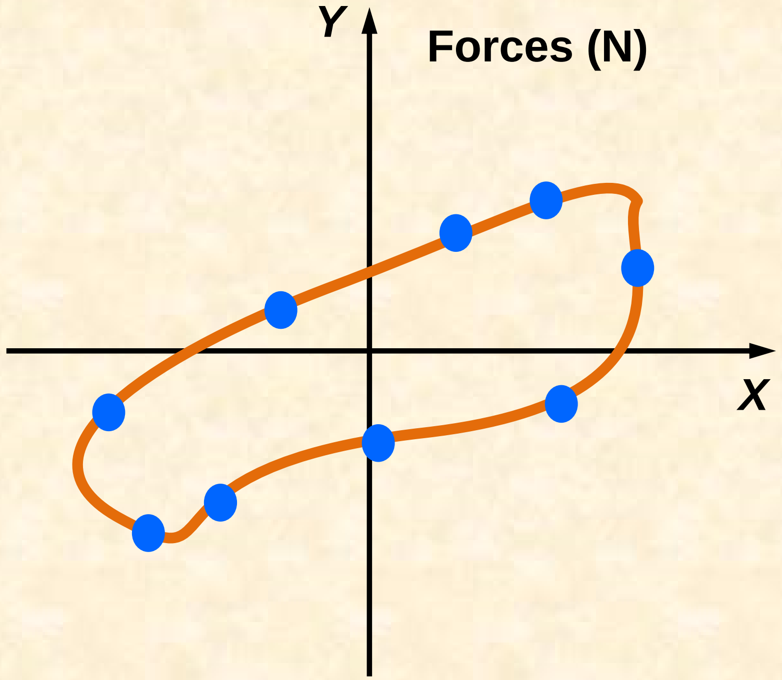
Pressures in groove are as large as those in the film lands.

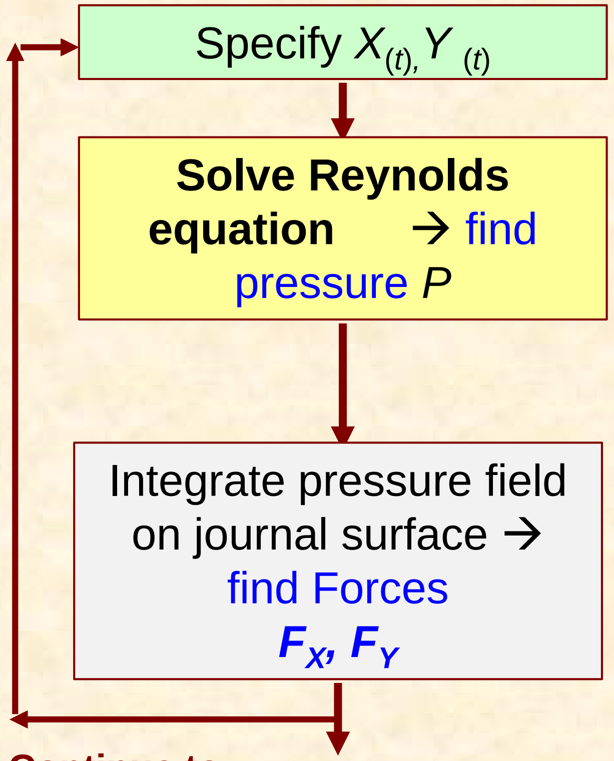
Accounts for dynamic pressure generation in deep grooves as measurements show!

Journal center kinematics and forces

Journal motion (*X* vs*Y*) → bearing reaction forces (*F_X* vs *F_Y*).







Model not restricted to small amplitude motions.

Procedure reproduces experimental one and *estimates* (numerically) force coefficients over a wide frequency range.

See also ASME GT2022-81990

Practical questions answered by R&D:

- **How do the film length and clearance affect SFD force coefficients?**
- **Does the amplitude / shape of whirl motion affect the force coefficients?**

today

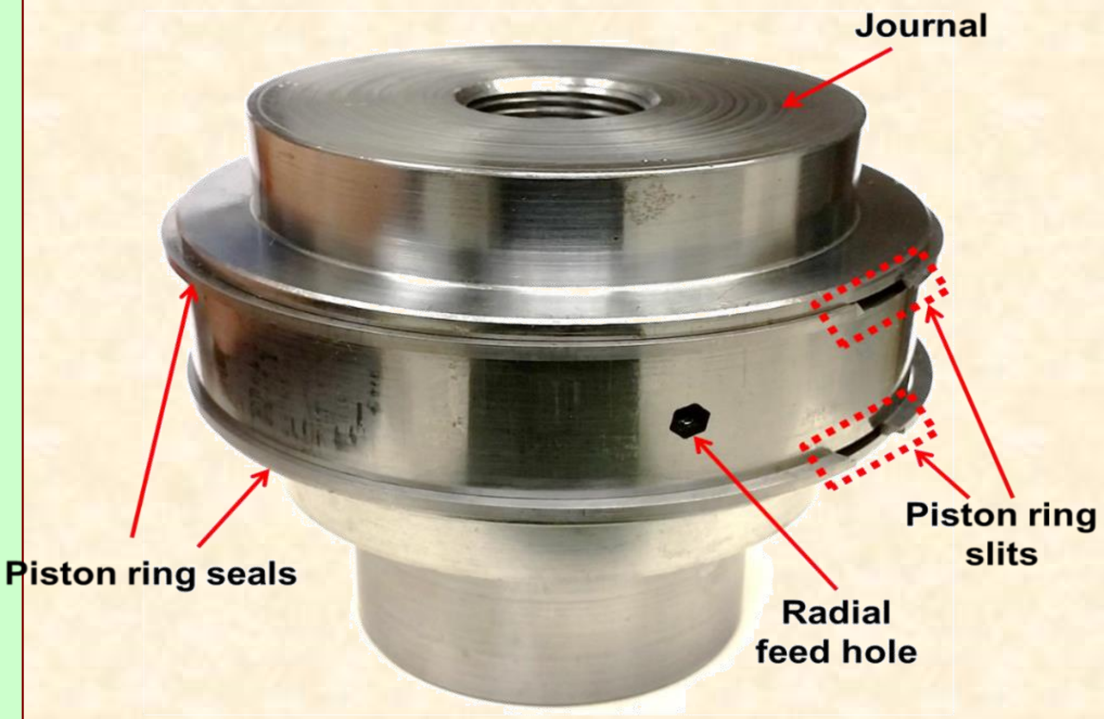
- **Do end seals work? how much more damping do they enable?**

- **Is a damper configuration with feed holes as effective as one containing a feed groove?**
- **What if one feed hole plugs, is a damper still effective?**
- **What if the damper operates largely off-centered; does its performance become nonlinear?**

Learn more

San Andrés, L., Jeung, S.-H, Den, S., and Savela, G., 2016 "**Squeeze Film Dampers: A Further Experimental Appraisal of their Dynamic Performance,**" *Proceedings of the 45th Turbomachinery Symposium, Houston, TX*

A SQUEEZE FILM DAMPER SEALED WITH PISTON RINGS OR WITH ORINGS: WHICH ONE TO SELECT?



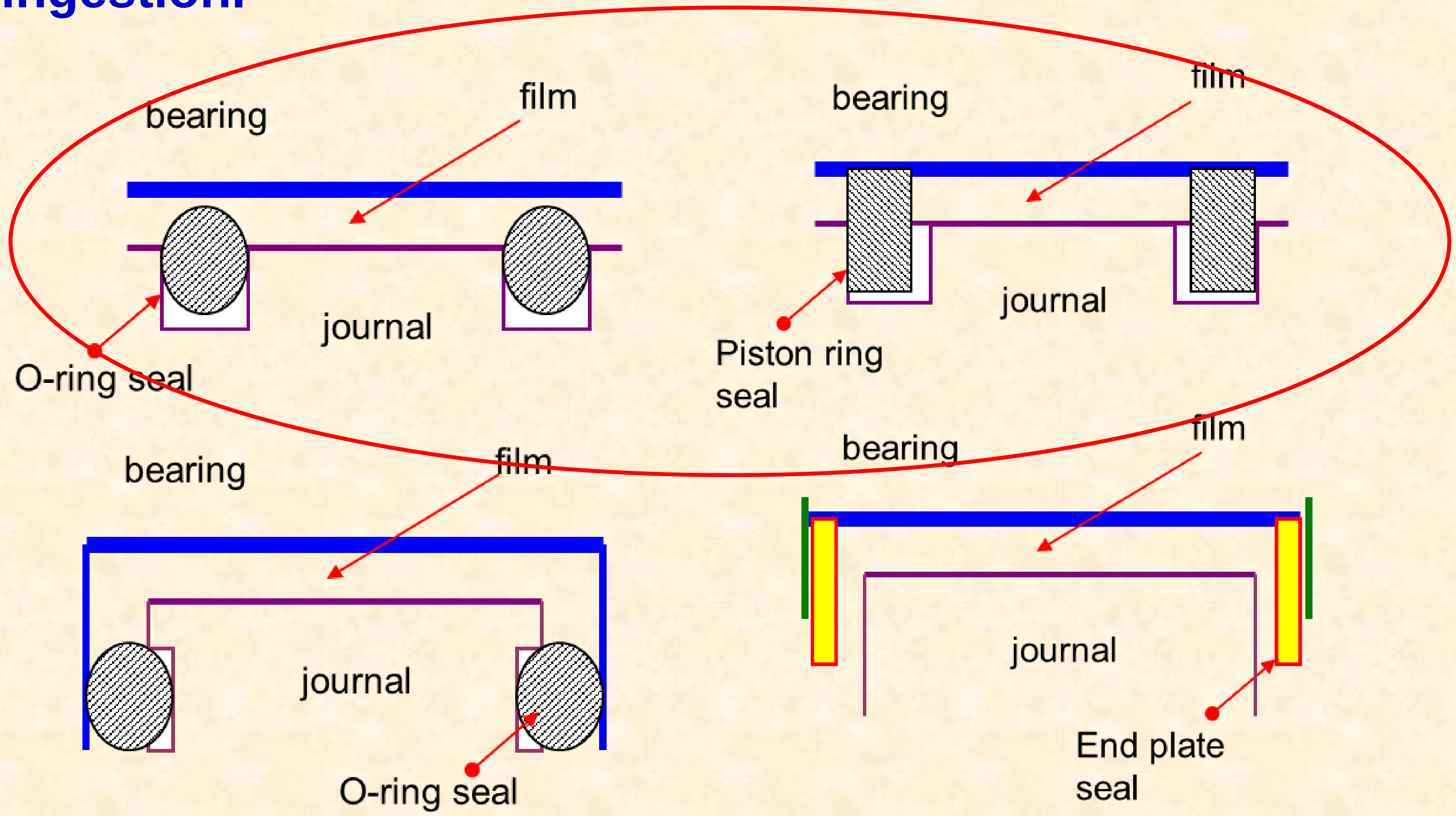
San Andrés, L., Jeung, S-H, Koo, B., **2018**, ASME GT2018-76224

San Andrés, Koo, B., **2019**, ASME GT 2019-90330

End seals for SFDs

Compressors use O-rings, and commercial jet engines implement piston rings.

Reduce thru flow and increase damping but cannot always prevent air ingestion.



O-ring issues:

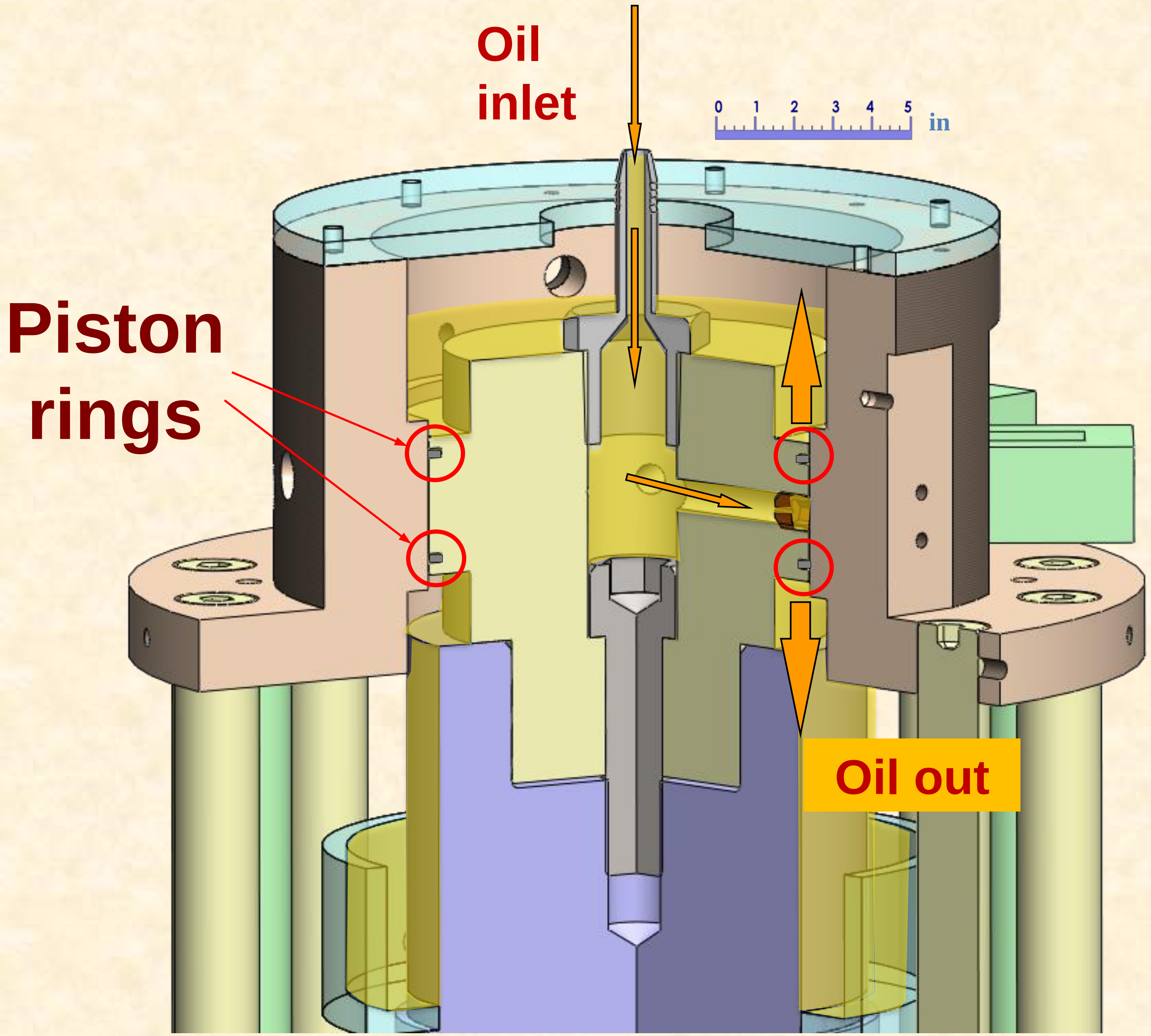
Special groove machining,
Material compatibility,
Add viscoelastic effect.

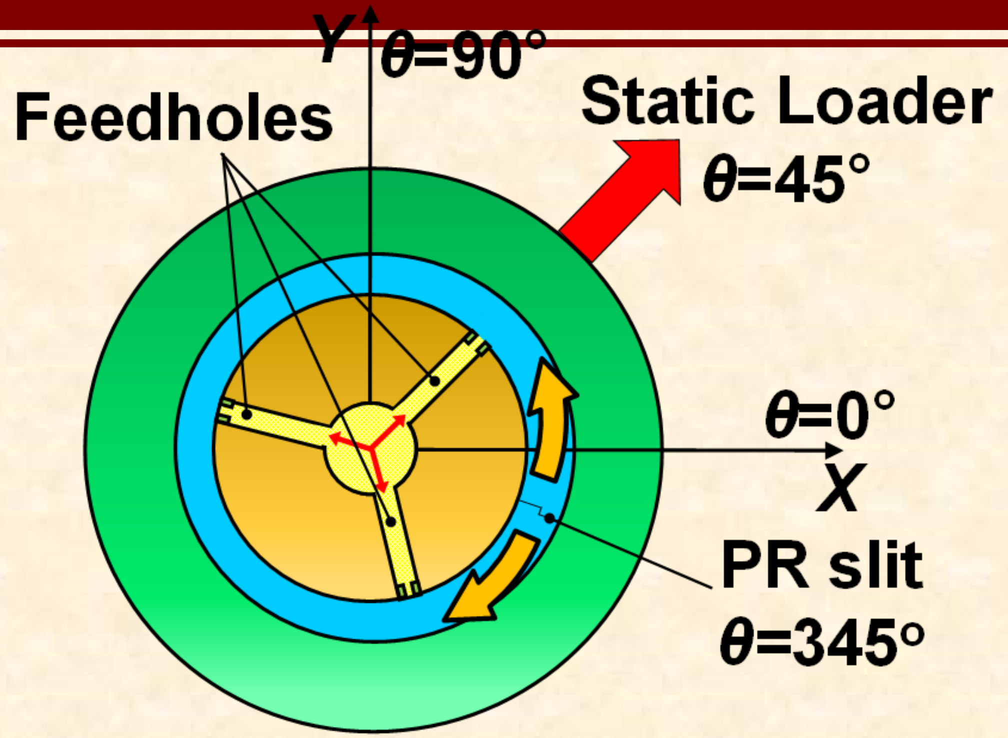
Piston ring issues:

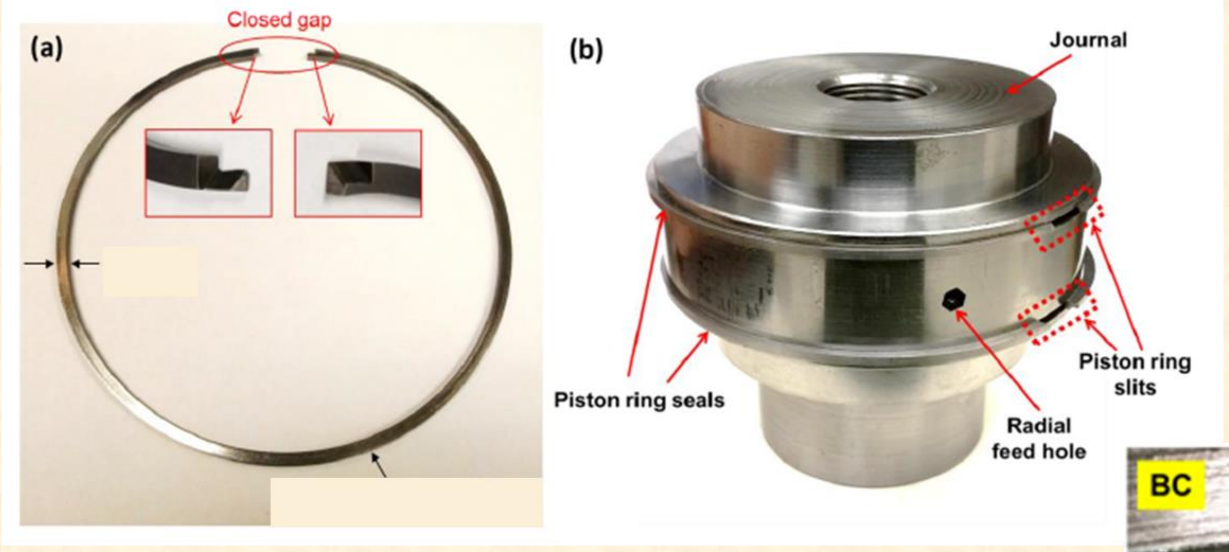
Cocking and locking
Slits – leak too much

Design is highly empirical, except for end plate seals.

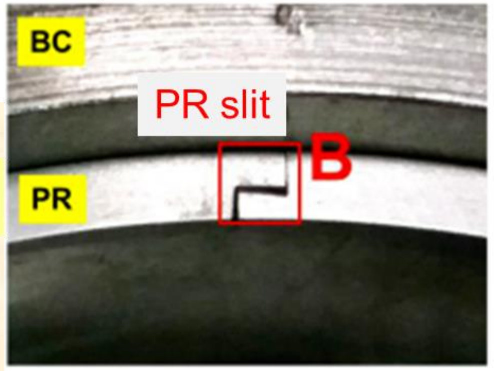
PR-SFD lubricant flow path





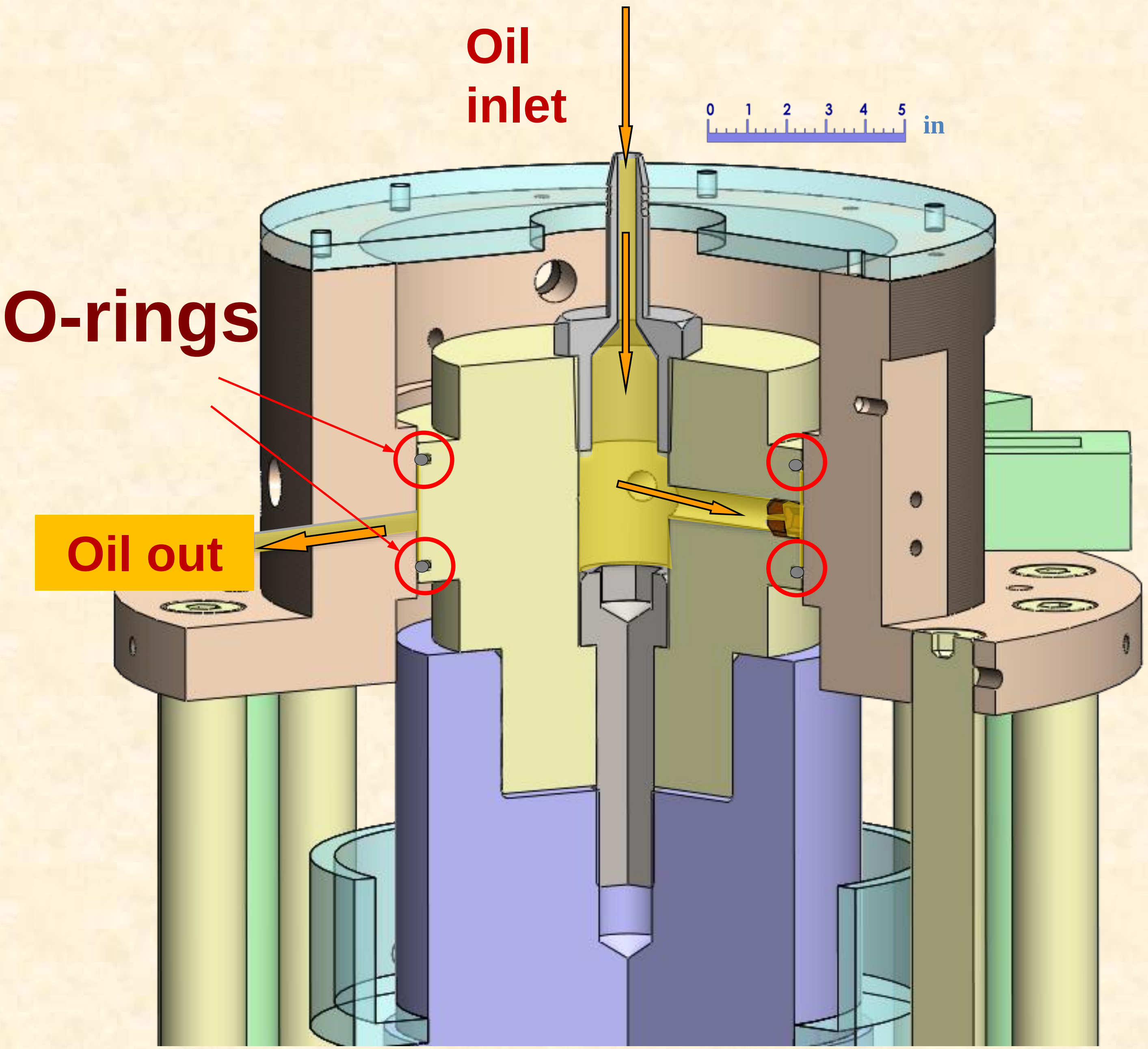


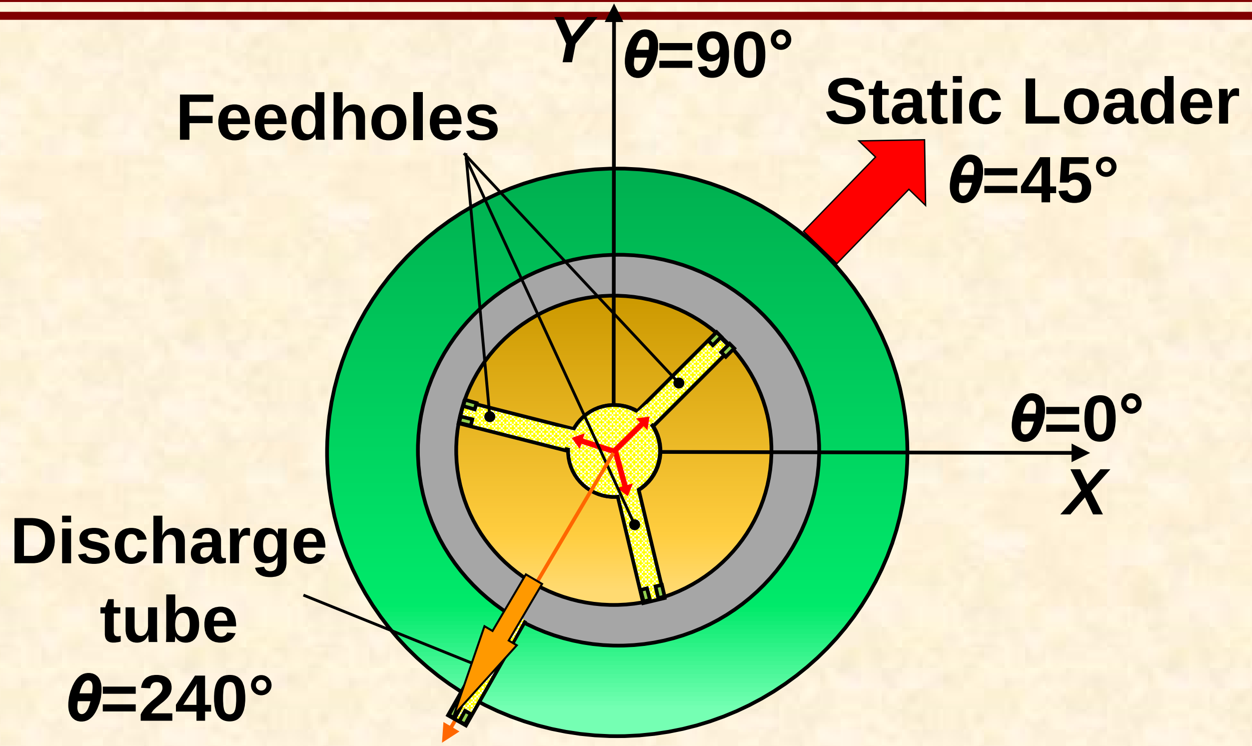
PR slits allow oil leakage and **air ingestion**

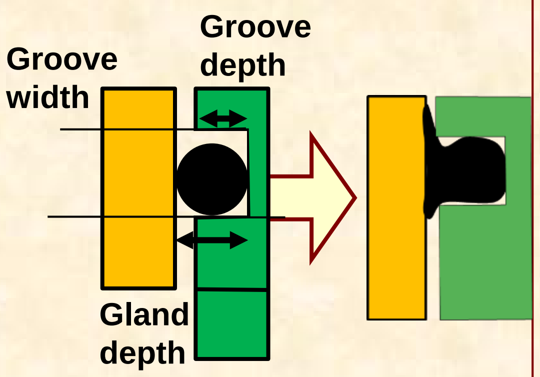


Proper installation of PR with slit on correct side is important.

OR-SFD lubricant flow path



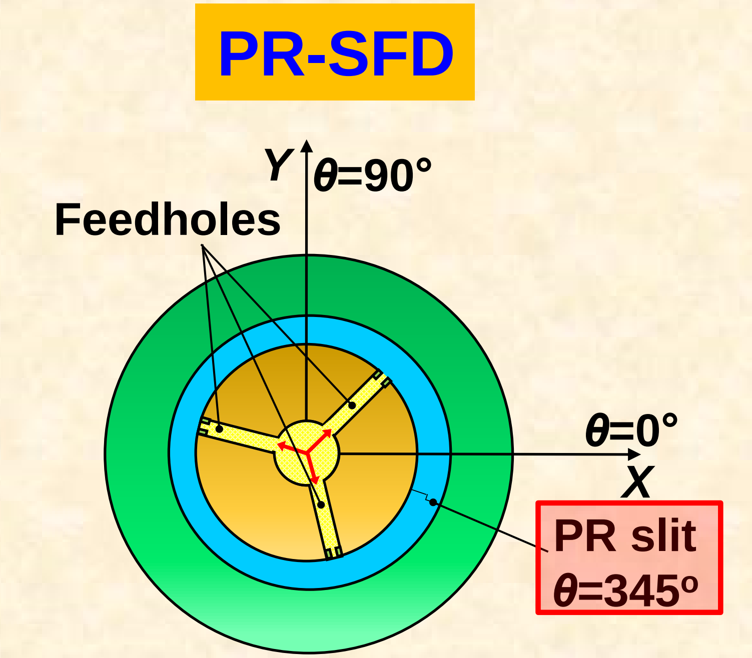


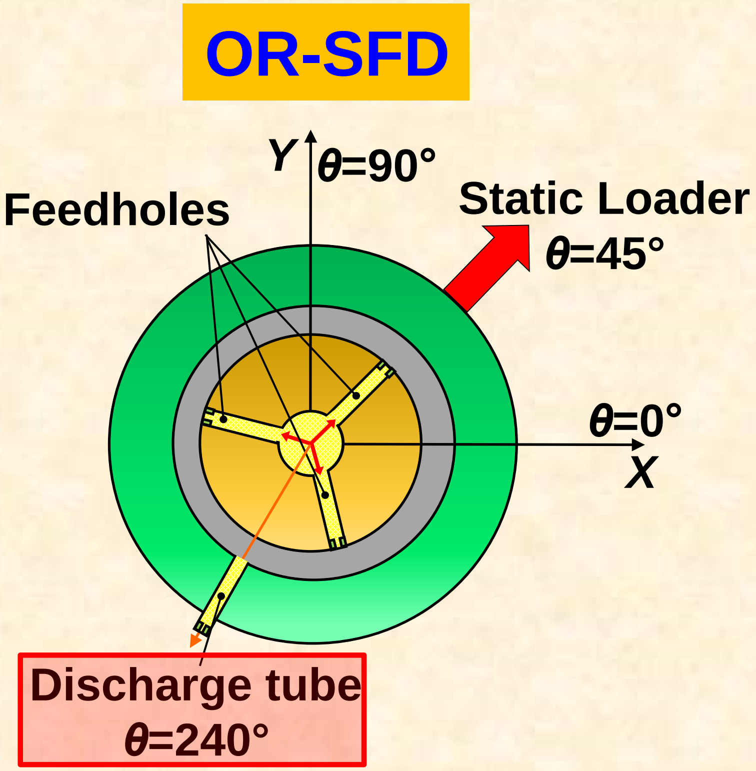


Adequate squeeze and volume fill ensure proper sealing. If too low, the OR does not seal; if too tight OR permanently deforms and becomes overly stiff.

ORs fully seal film land; hence oil evacuation through hole needed.

Short length damper (L/D=0.2, D/c=340)





Journal diameter, D	**127 mm**
Axial film land length, L	**25.4 mm**
Radial clearance, c	**0.373 mm**
Feedhole diameter, ϕ_{in}	**2.5 mm**
angular location, θ_{in}	45°, 165°, 285°
OR-SFD Discharge hole diameter, ϕ_{exit}	1.0 mm
hole location, θ_{exit}	240°
PR-SFD slit location, θ_{slit}	345°

Max squeeze velocity $v_s = r\omega \sim 60$ mm/s
→ $Re_s = (\rho / \mu)\, \omega\, c^2 \sim 65$

ISO VG 2 oil

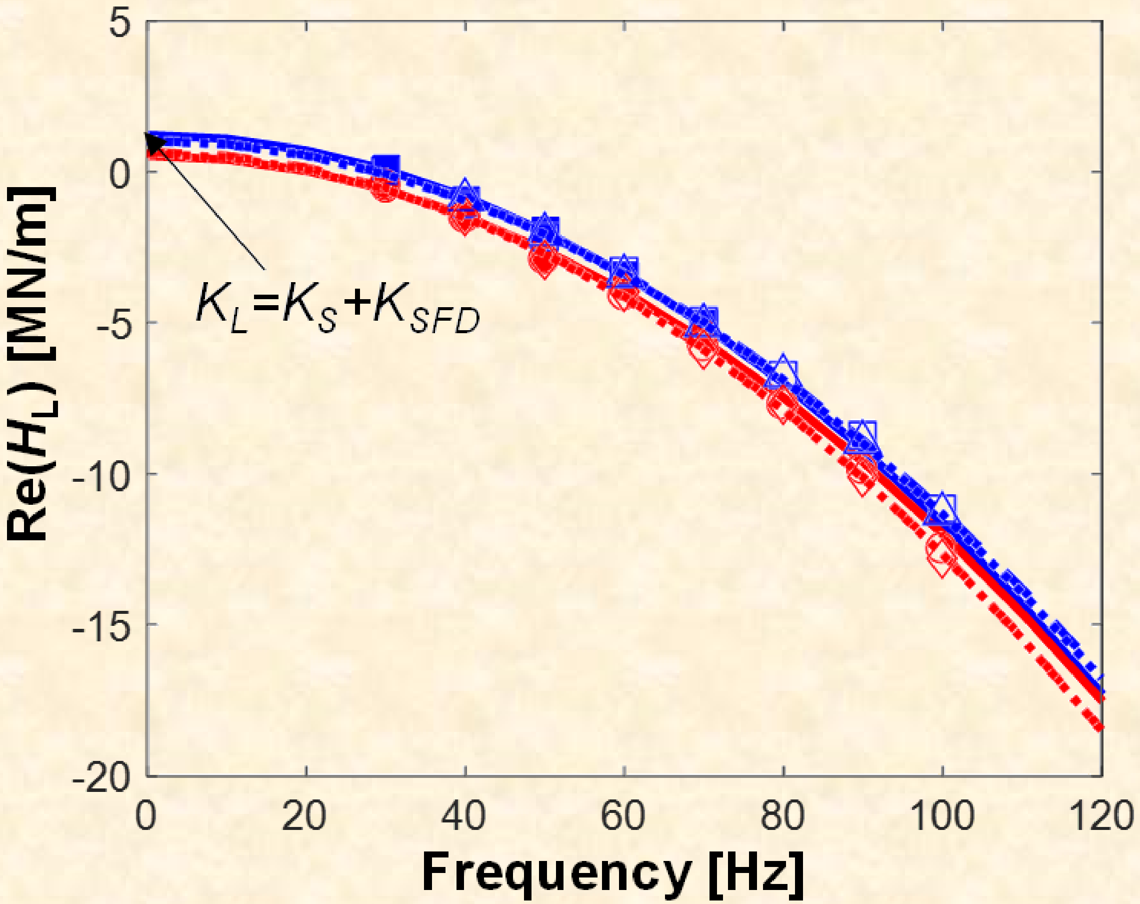
Oil inlet temperature T_s = 23 °C
Density ρ = 799 kg/m³
Viscosity μ at T_s = 2.7 cPoise

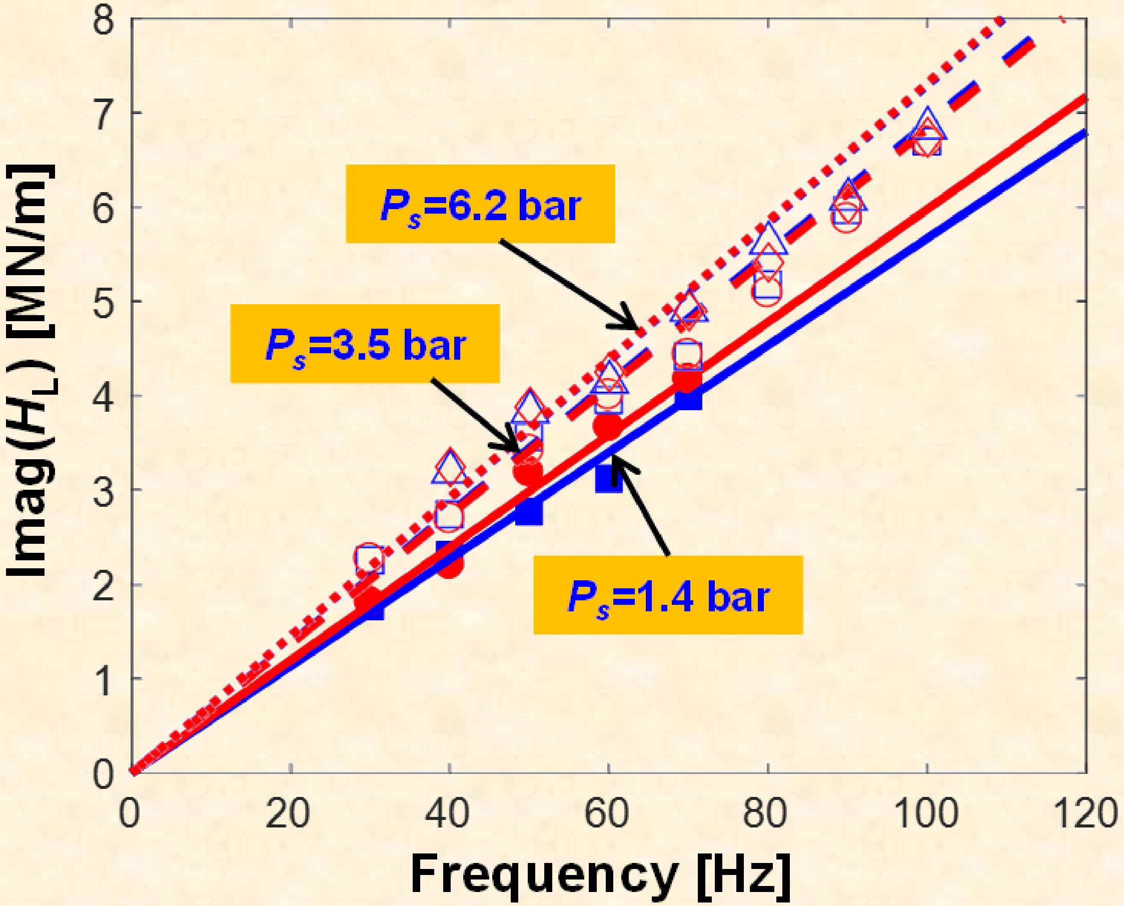
Typical PR-SFD complex stiffness

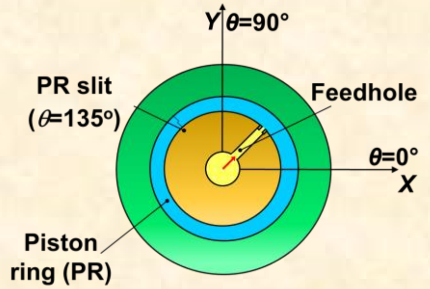
c=0.37 mm, orbit size r/c=0.3

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

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



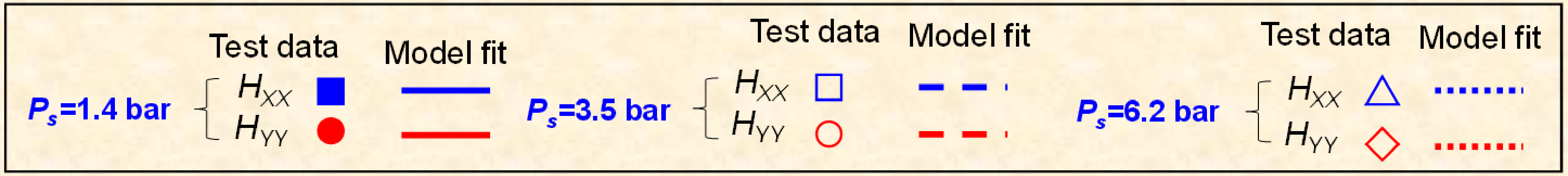




K_{S+SFD} ~ K_S=1.6 MN/m.

Damping C ↑ as P_s ↑.

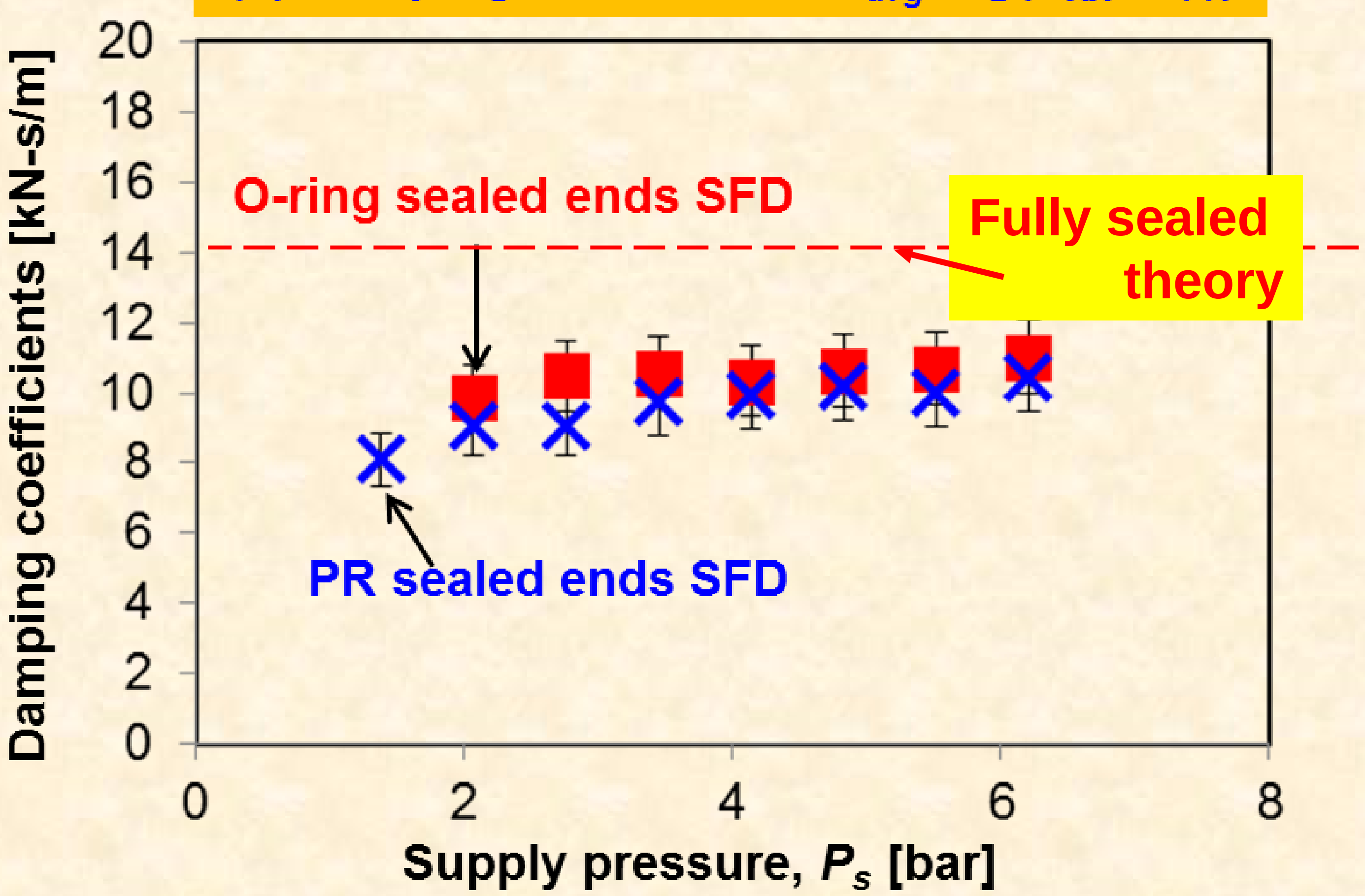
Added mass M ~ same.



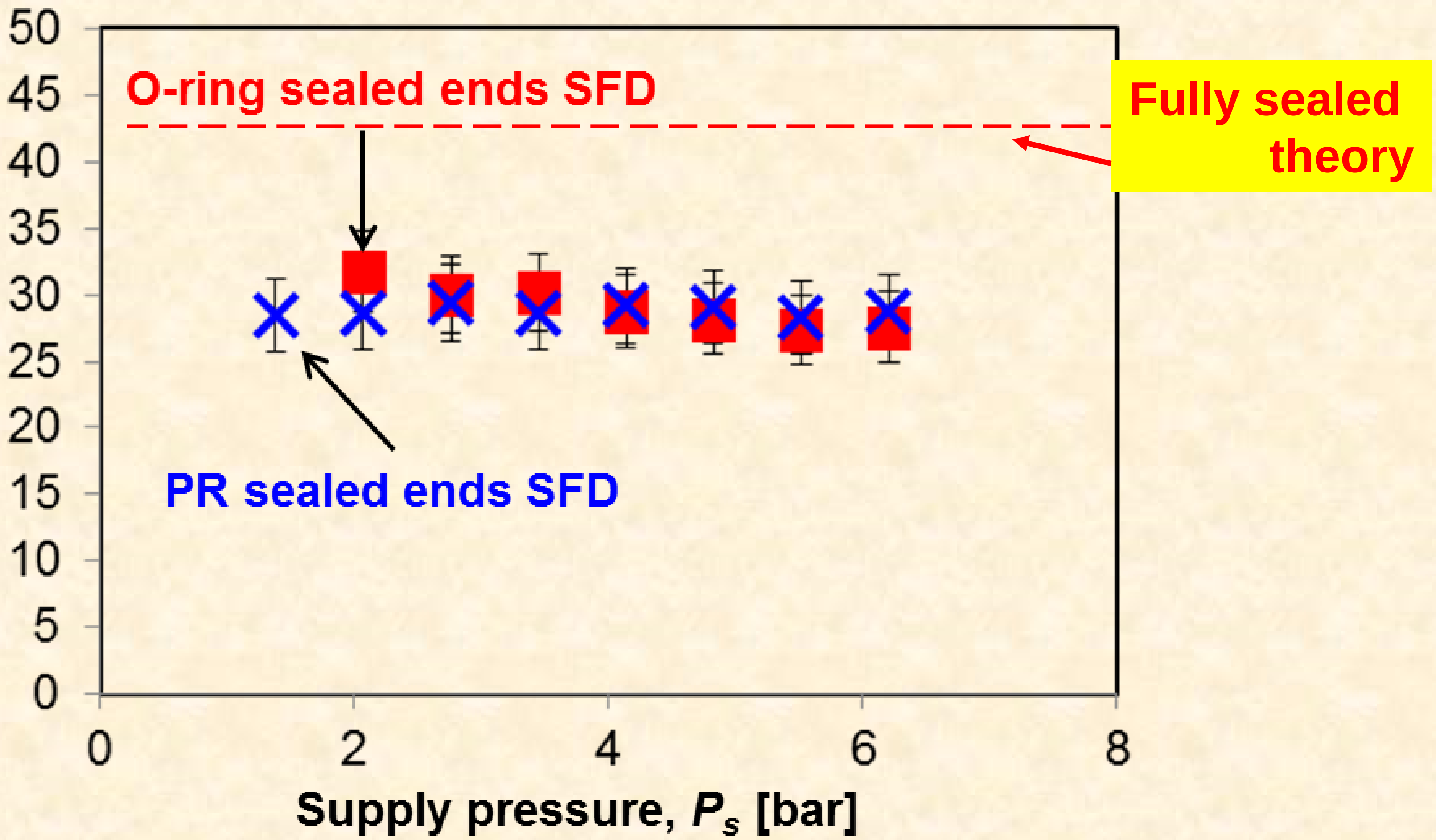
→ **Extract SFD force coefficients from curve fits to complex stiffnesses**

OR-SFD vs PR-SFD *C* and *M* vs. supply pressure

(a) Damping coefficients, $C_{avg} = \frac{1}{2}(C_{XX}+C_{YY})$



(b) Added inertia coefficients, $M_{avg} = \frac{1}{2}(M_{XX}+M_{YY})$

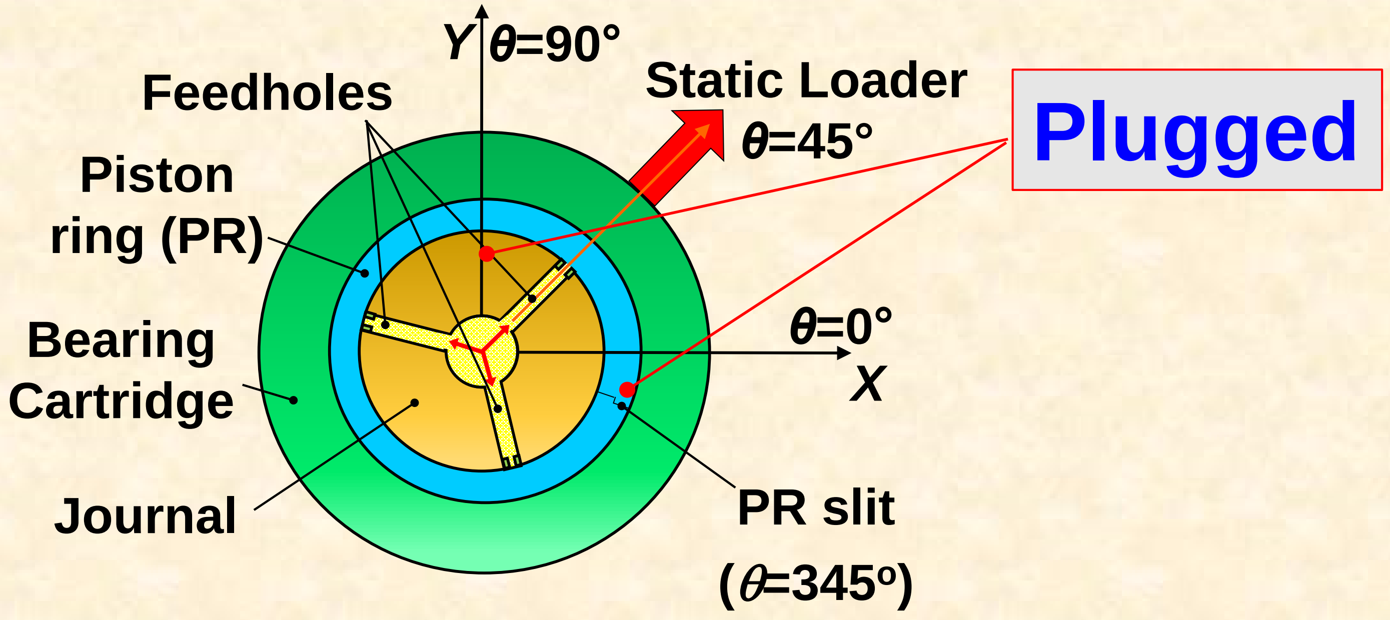


- **Damping coefficients *C* ↓ as lubricant supply pressure ↓**
- **Damping *C* for OR-SFD is 11% larger than *C* for PR-SFD.**
- **Added mass *M* ~30 kg as supply pressure decreases.**

***c*=373 μm, *r*/*c*=0.3, *ω*=10-100 Hz → Max. $v_s = r\omega$ = 70 mm/s, Max. Re_s=26**

Effect of number of open feed holes on SFD forced performance

OR-SFD vs. PR-SFD



Dimensionless force coefficients

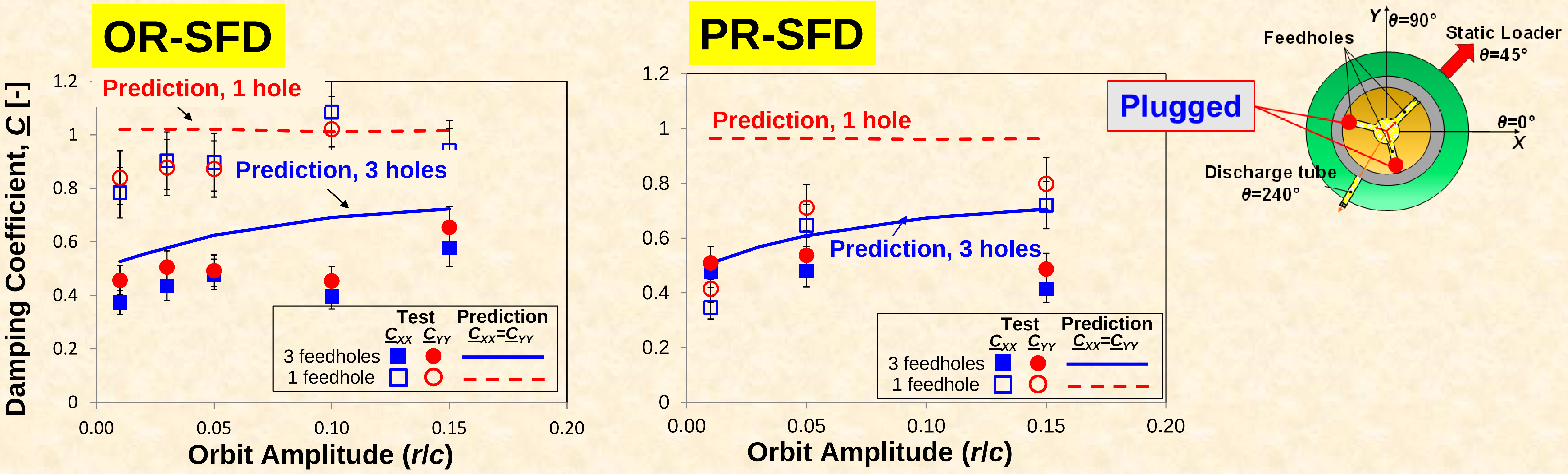
$\underline{C}=C/C^*$ $\underline{M}=M/M^*$

Long bearing model

$$C^* = 12\pi\mu L\left(\frac{D}{2c}\right)^3 \qquad M^* = \frac{\rho\pi L}{c}\left(\frac{D}{2}\right)^3$$

=12.85 kN s/m =44 kg

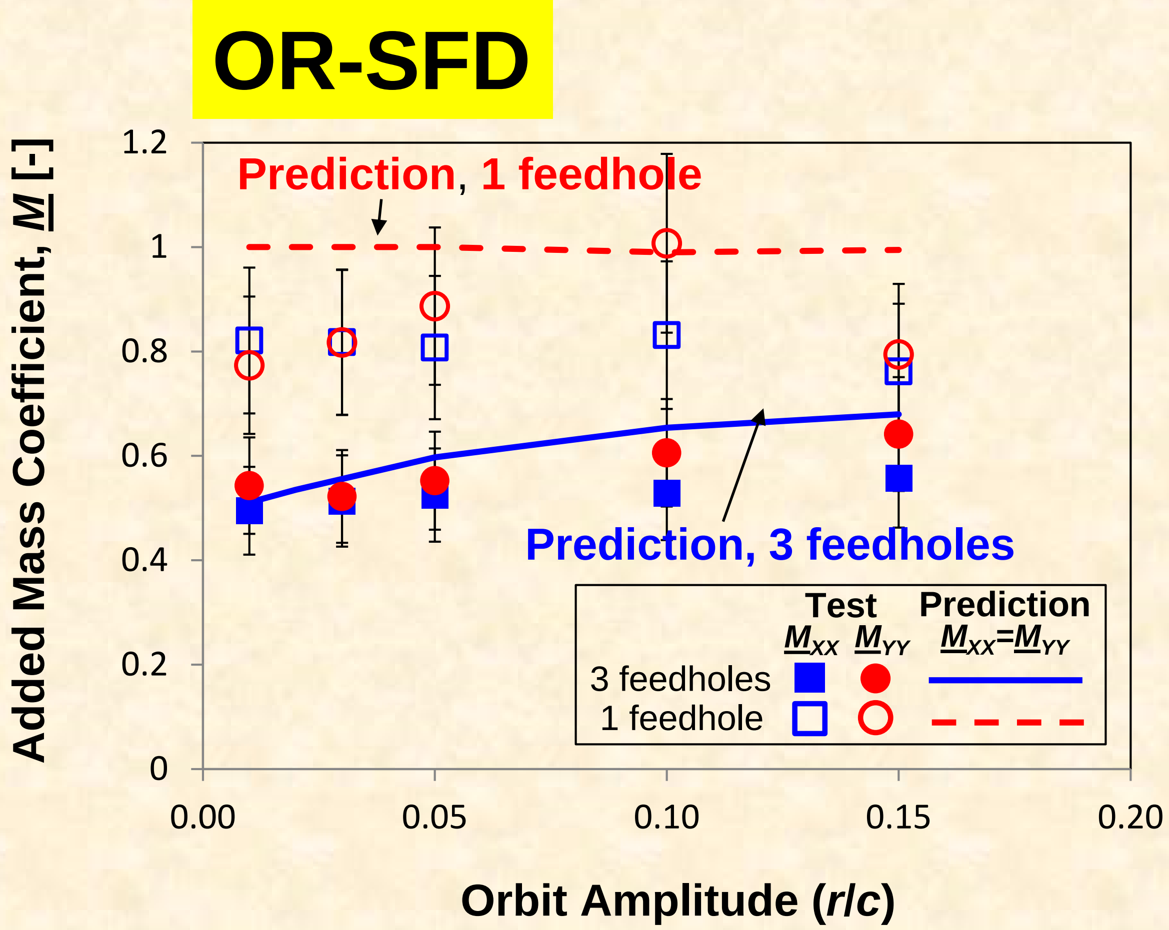
of feedholes → damping C_{SFD} P_s=69 kPa(g)

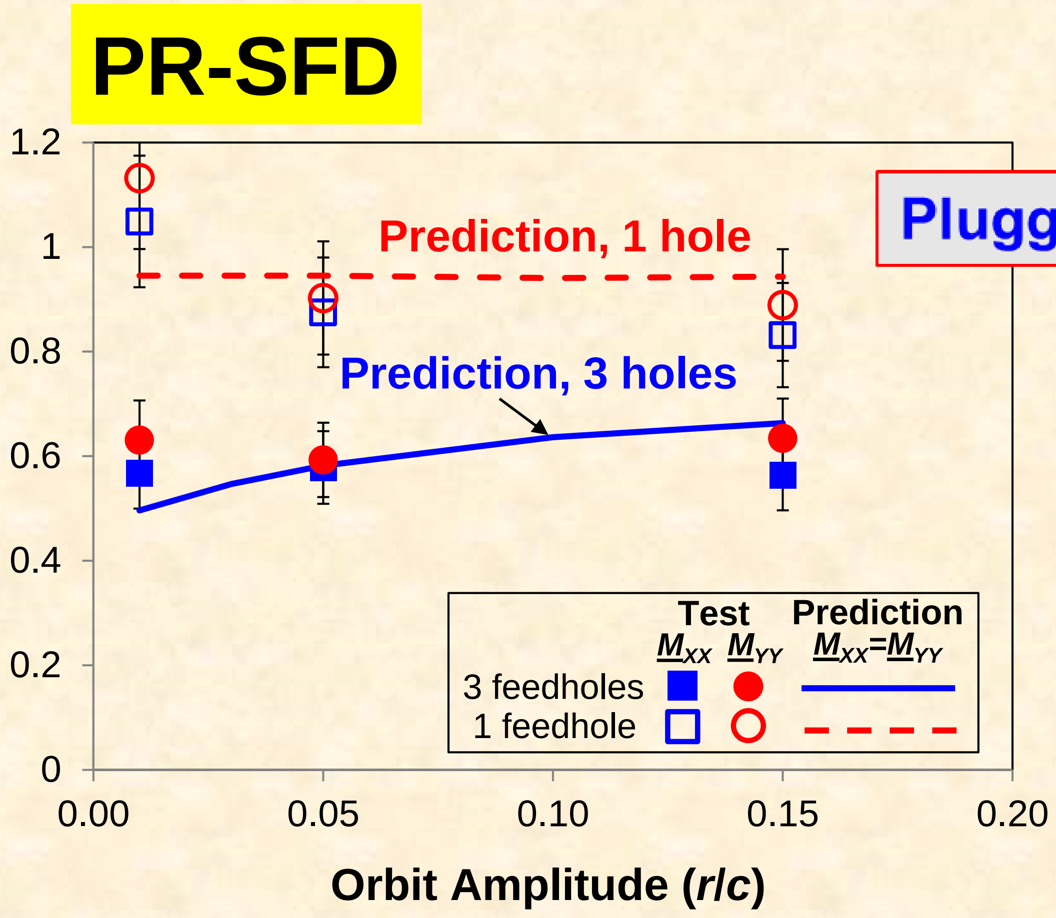


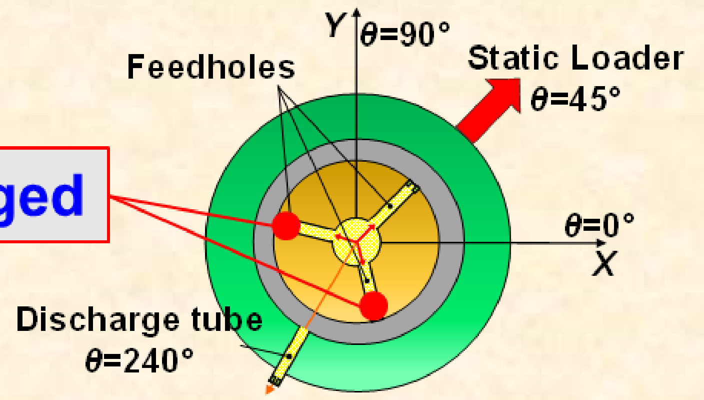
Damper with one open feedhole produces 60% +damping than SFD with three holes.

For damper with one feedhole, $C_{PR\text{-}SFD} < C_{OR\text{-}SFD}$ due to air ingestion through PR slits.

of feedholes → inertia M_{SFD} P_s=69 kPa(g)







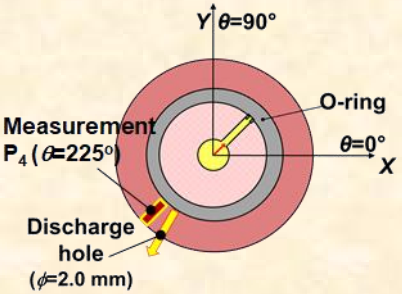
Damper with one feedhole produces more inertia (~80%) than damper with three feedholes. OR-SFD and PR-SFD produce same *M*.

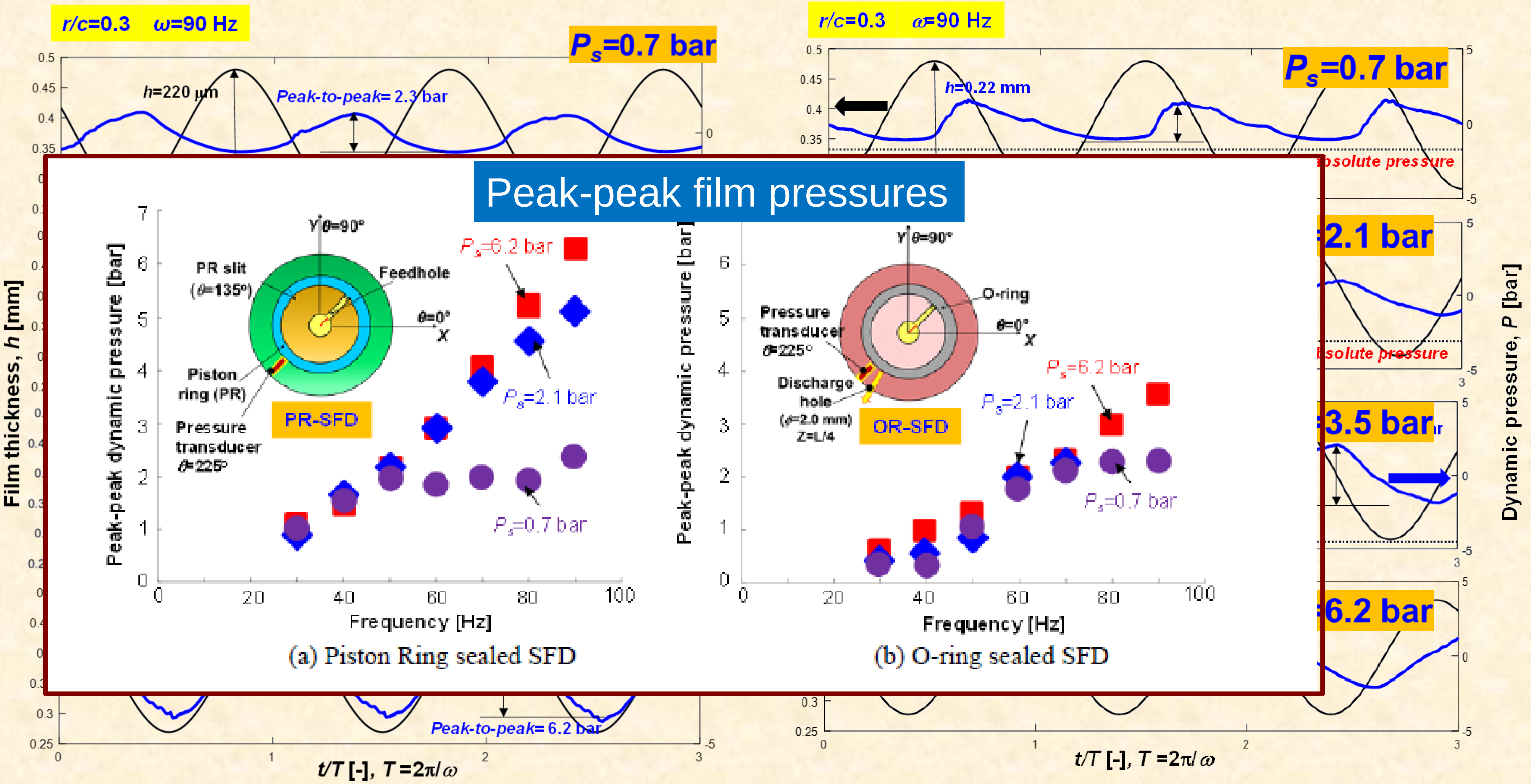
Although ends are sealed, test coefficients are 50% of long bearing model due to pressure distortions.

PR-SFD & OR-SFD pressure profiles

v_s = 63.2 m/s = (r = 0.3 c x ω = 90 Hz)

PR-SFD shows air ingestion at low feed pressures.





PR-SFD

OR-SFD

Closure OR-SFD vs PR-SFD

(a) O-ring damper provides more damping as it avoids air ingestion. O-rings add stiffness and viscoelastic damping to test system.

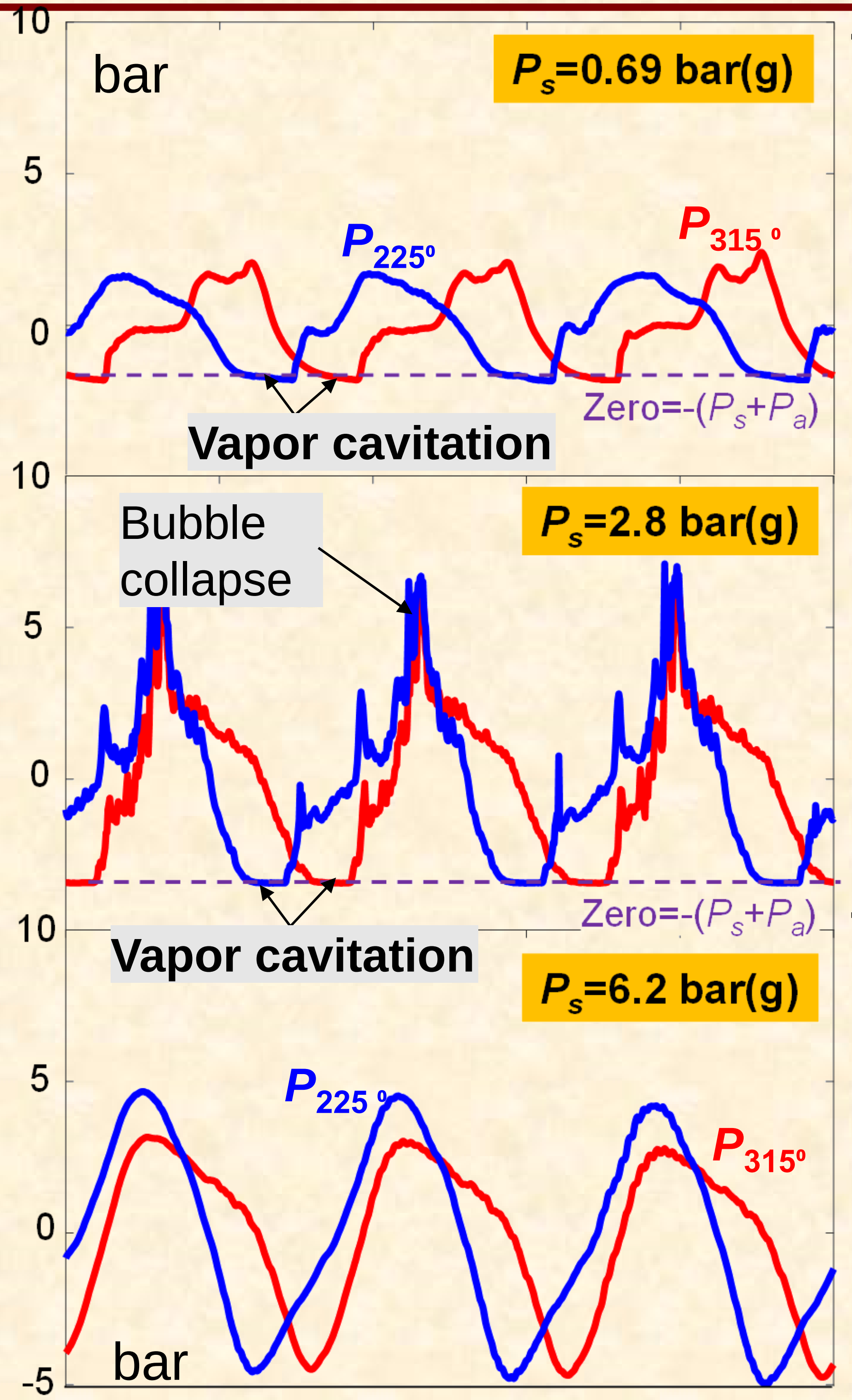
(b) For both PR-SFD and OR-SFD, the more # feedholes, the lower the damping coefficient.

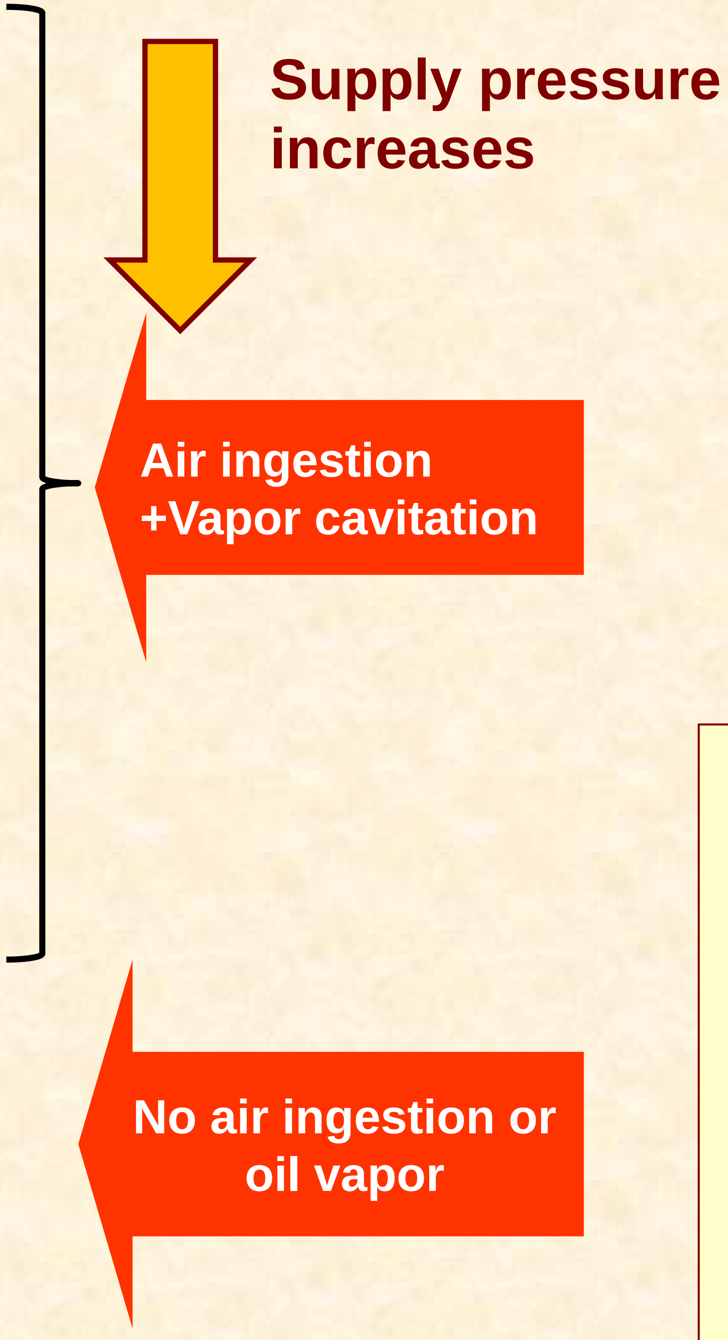
(c) SFD coefficients from dynamic film pressure are largely in error. Pressure field does not simply rotate!

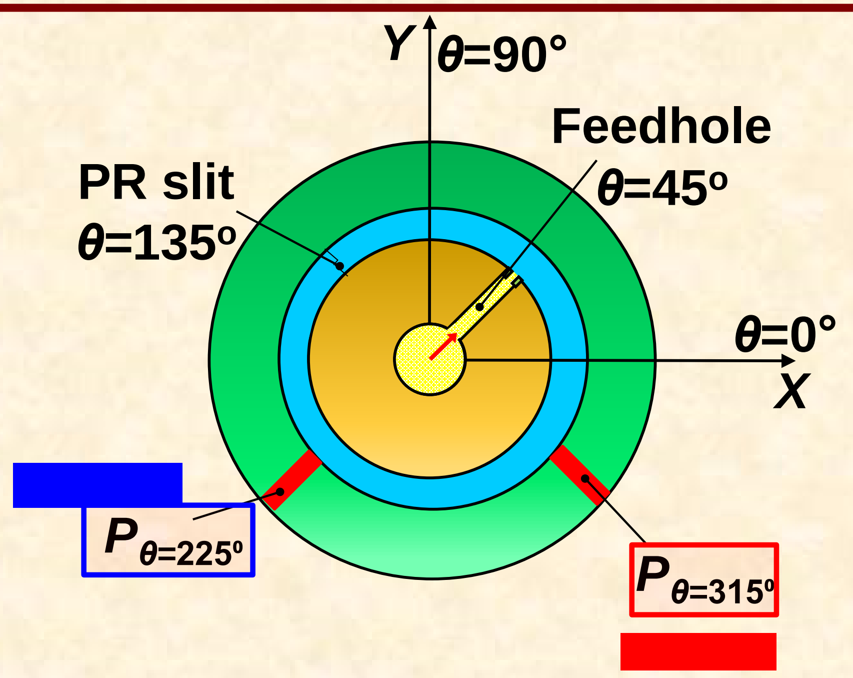
(d) Film pressures show oil vapor cavitation and persistent air ingestion for operation at a low supply pressure and/or with a large squeeze velocity (v_s).

PR-SFD more film pressures

60 Hz, r=0.65c: v_s~90 mm/s → Re_s~65







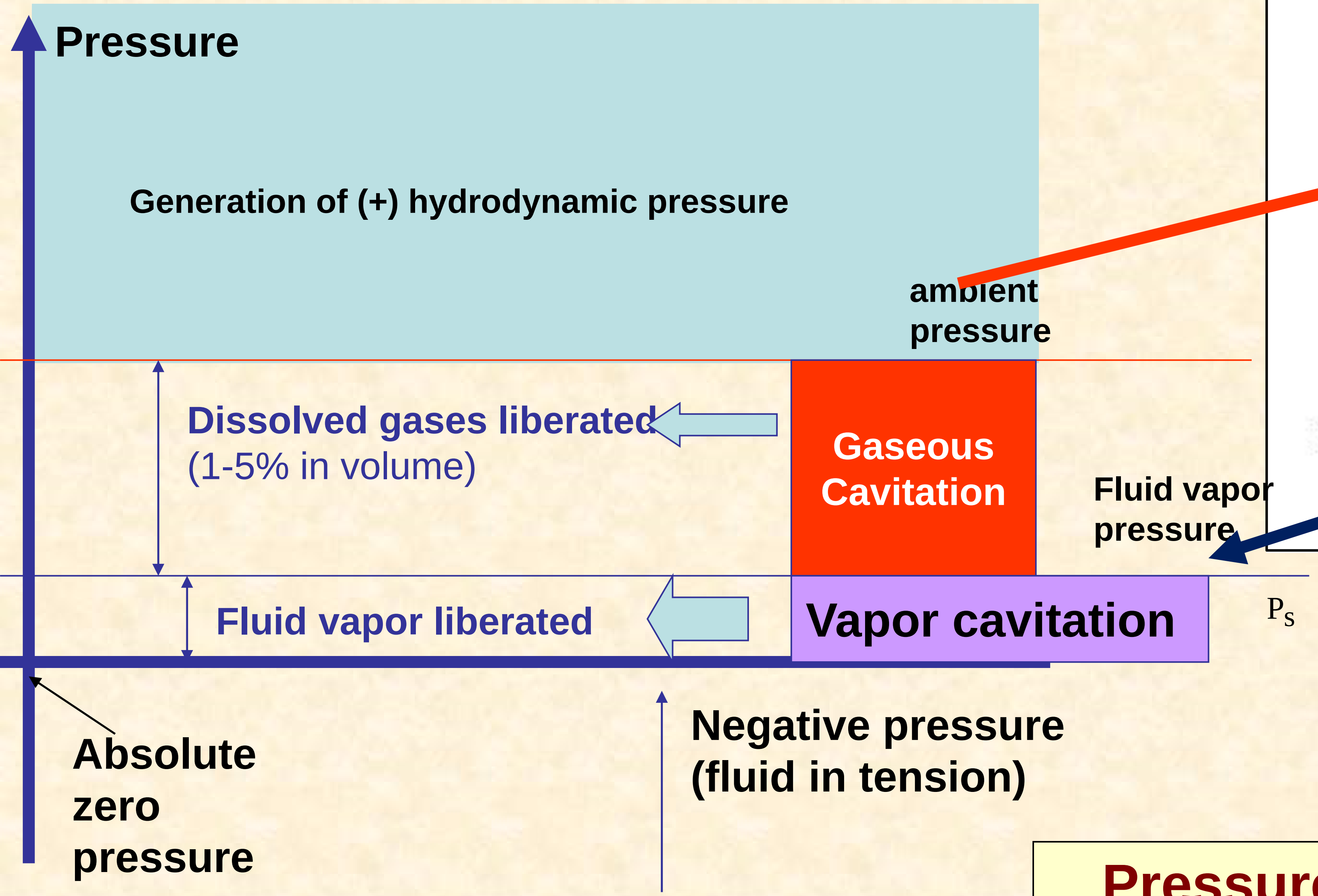
Pressures show both oil vapor cavitation & air ingestion with large amplitude/high frequency spikes from bubbles collapsing.

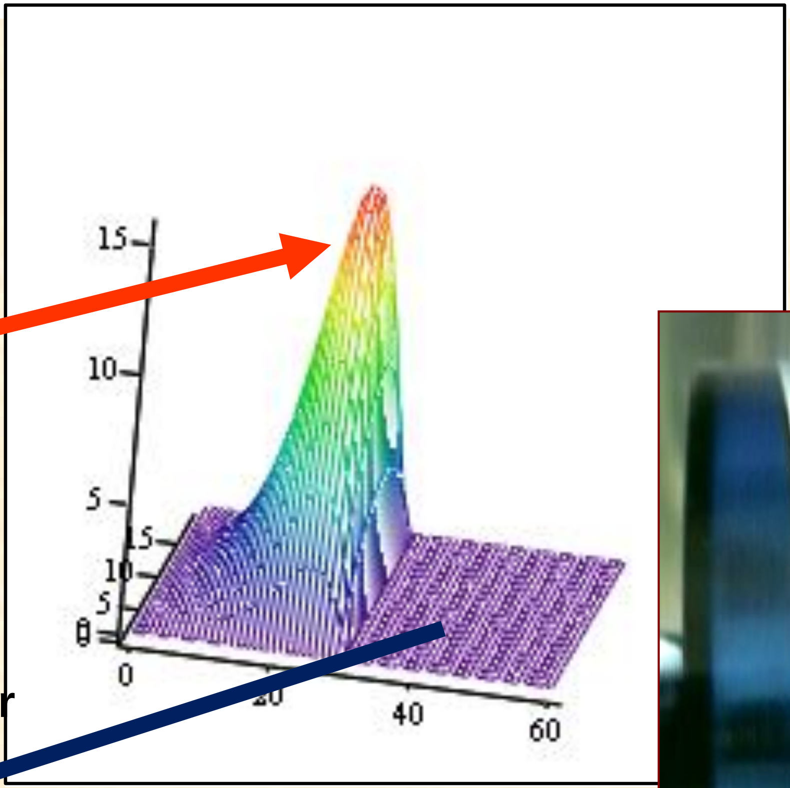
Pressures are distorted & do NOT displace with whirl speed!

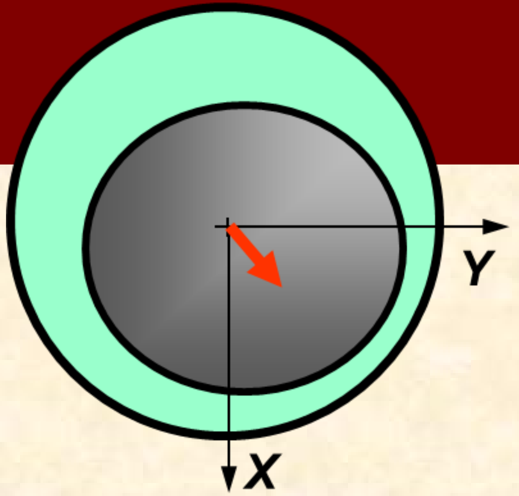
Oil cavitation vs. air ingestion and entrapment?

Major issue for reliable SFD forced performance!

Cavitation in oil lubricated bearings



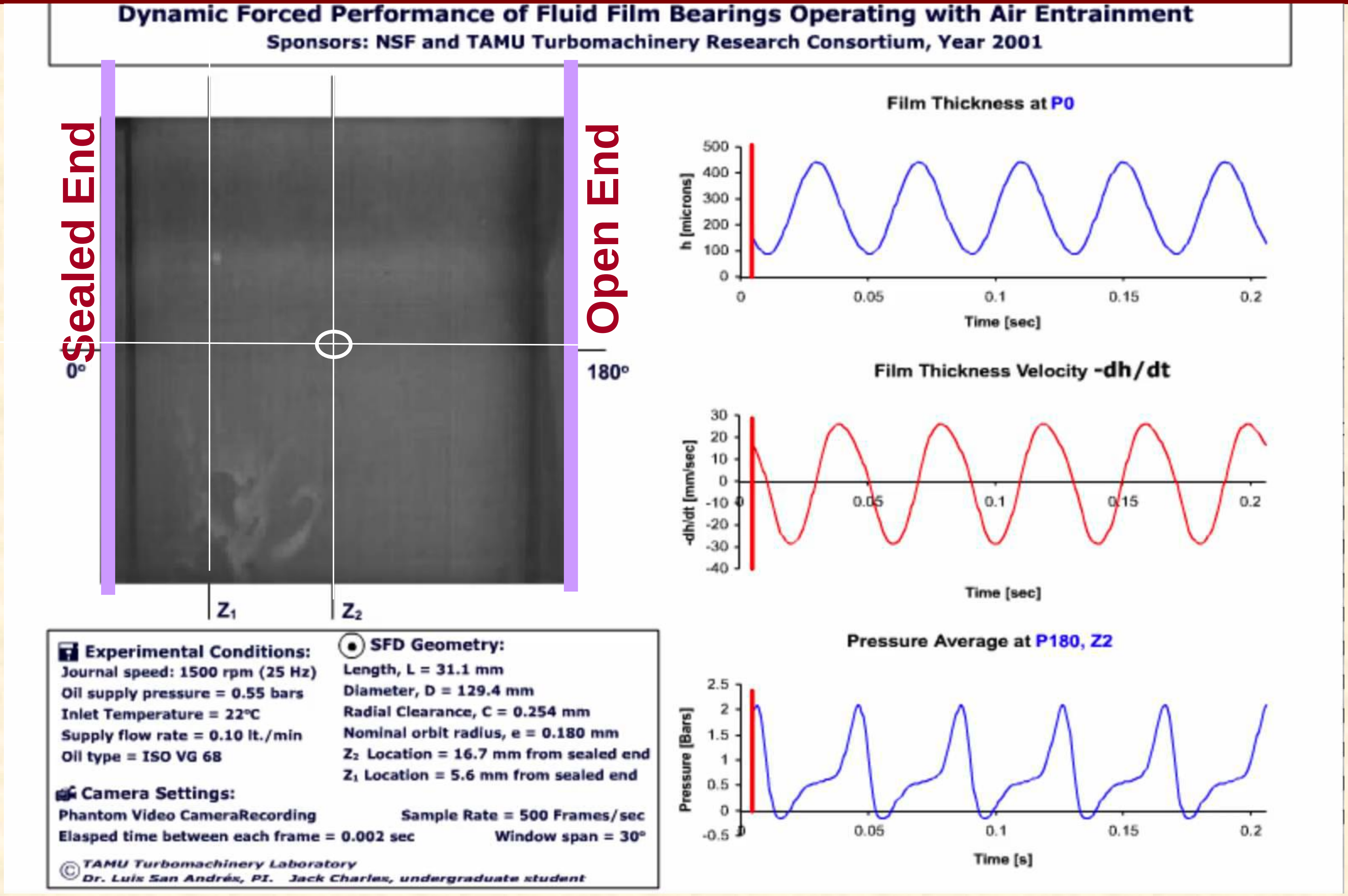


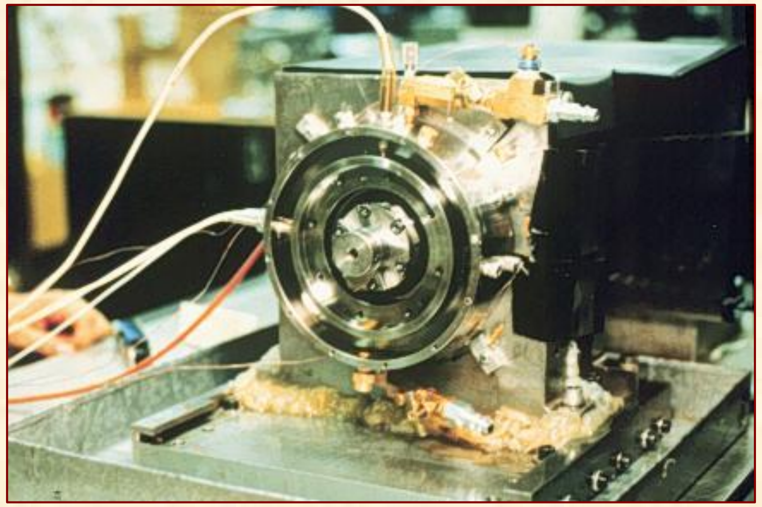


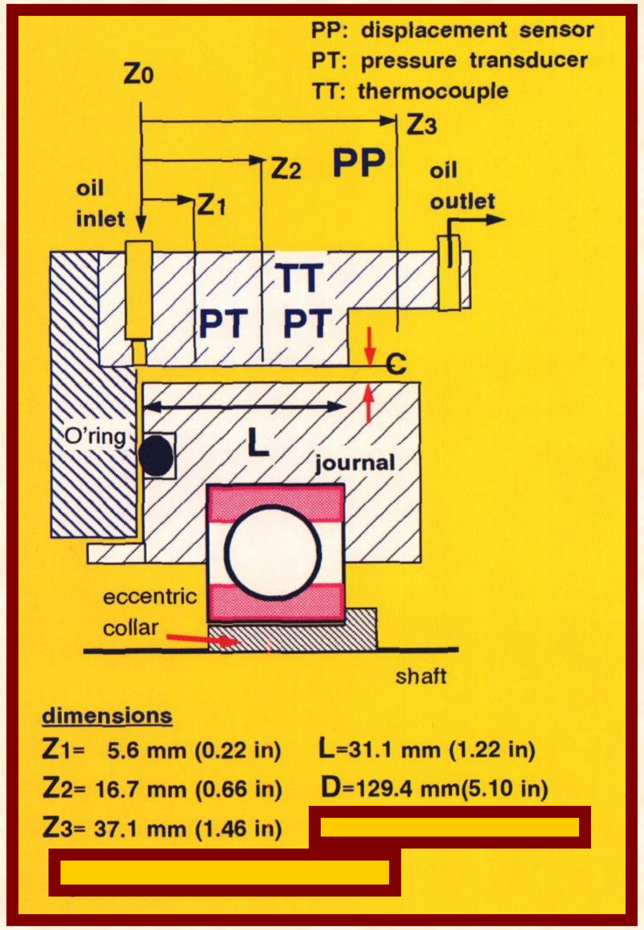
Pressure is uniform (constant) inside cavitation "bubble" – Flow reformation at trailing edge of bubble.

55

SFD flow visualization of air ingestion (2001 NSF)



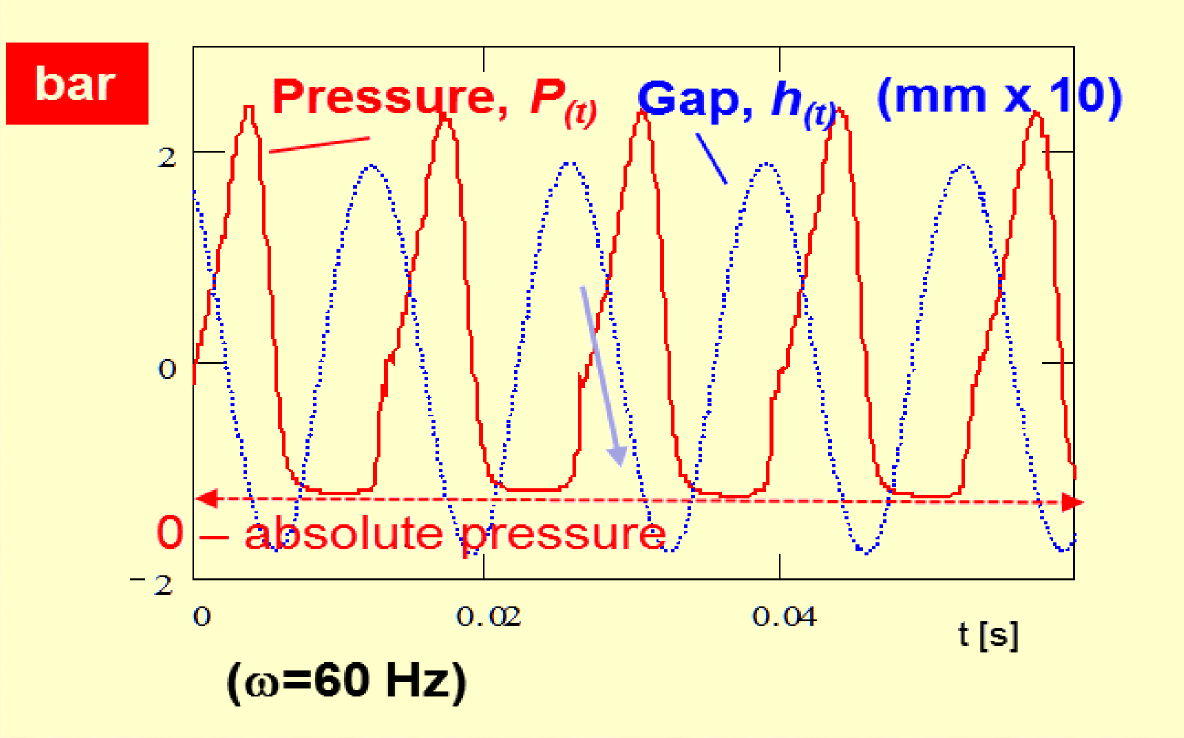




Air ingestion & entrapment persist under dynamic load conditions.

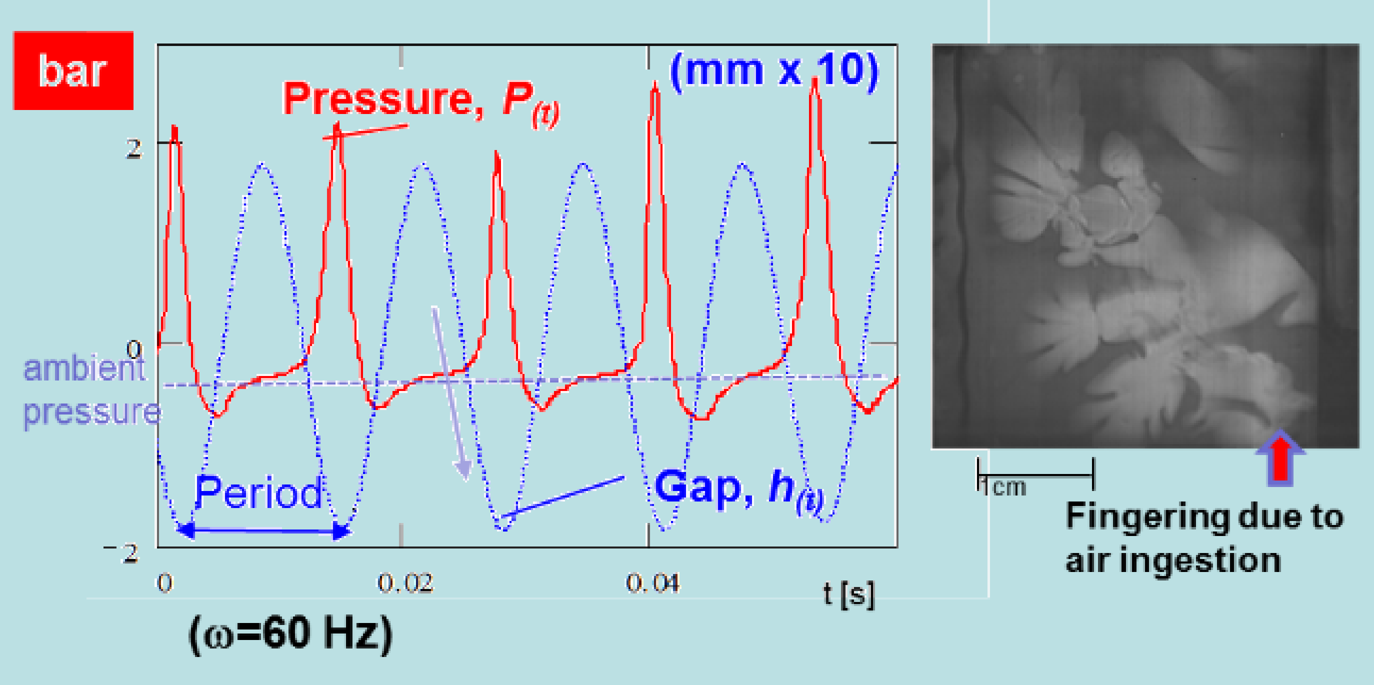
Lubricant cavitation vs. air ingestion in SFDs

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



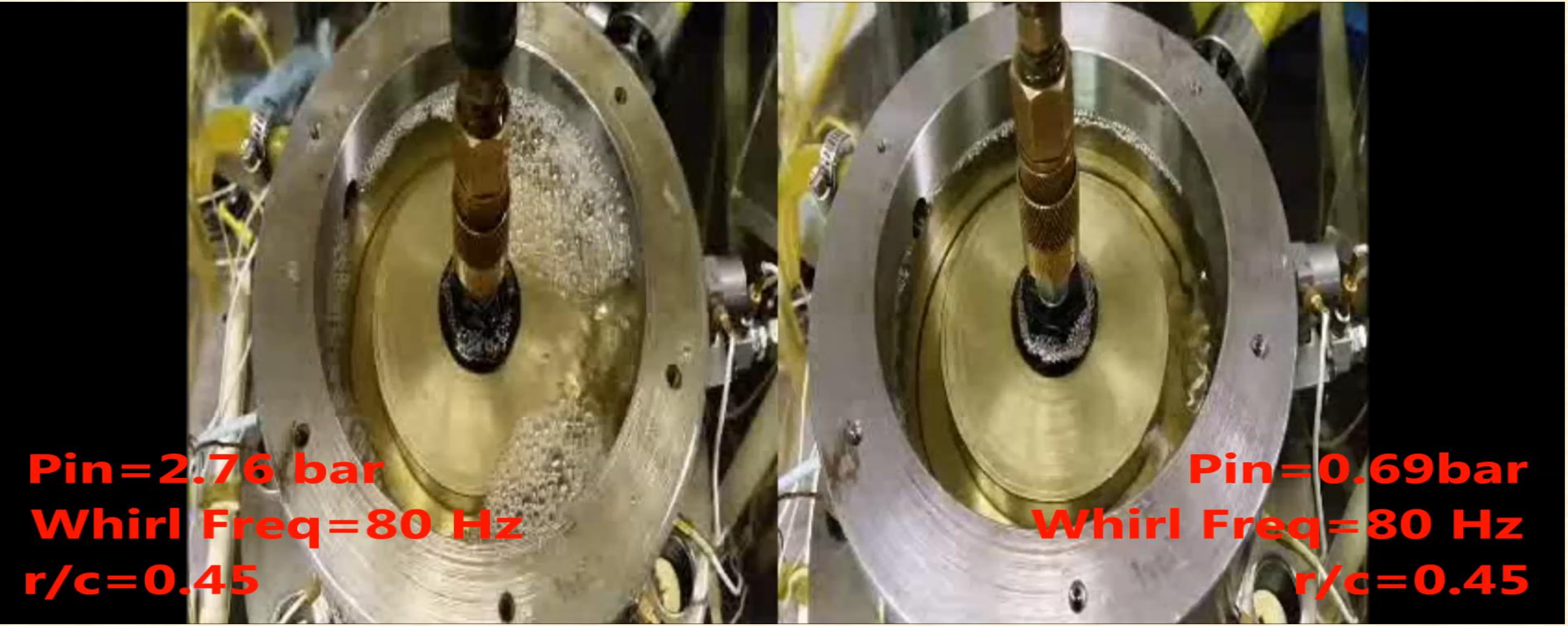
Gas cavitation: Cavitation zone, contains released **dissolved gas** in lubricant and appears steady in a rotating frame.

Air ingestion and entrapment: as local gap opens, **air is drawn** to fill in empty volume.



Onset of air ingestion

→High and low oil feed pressures



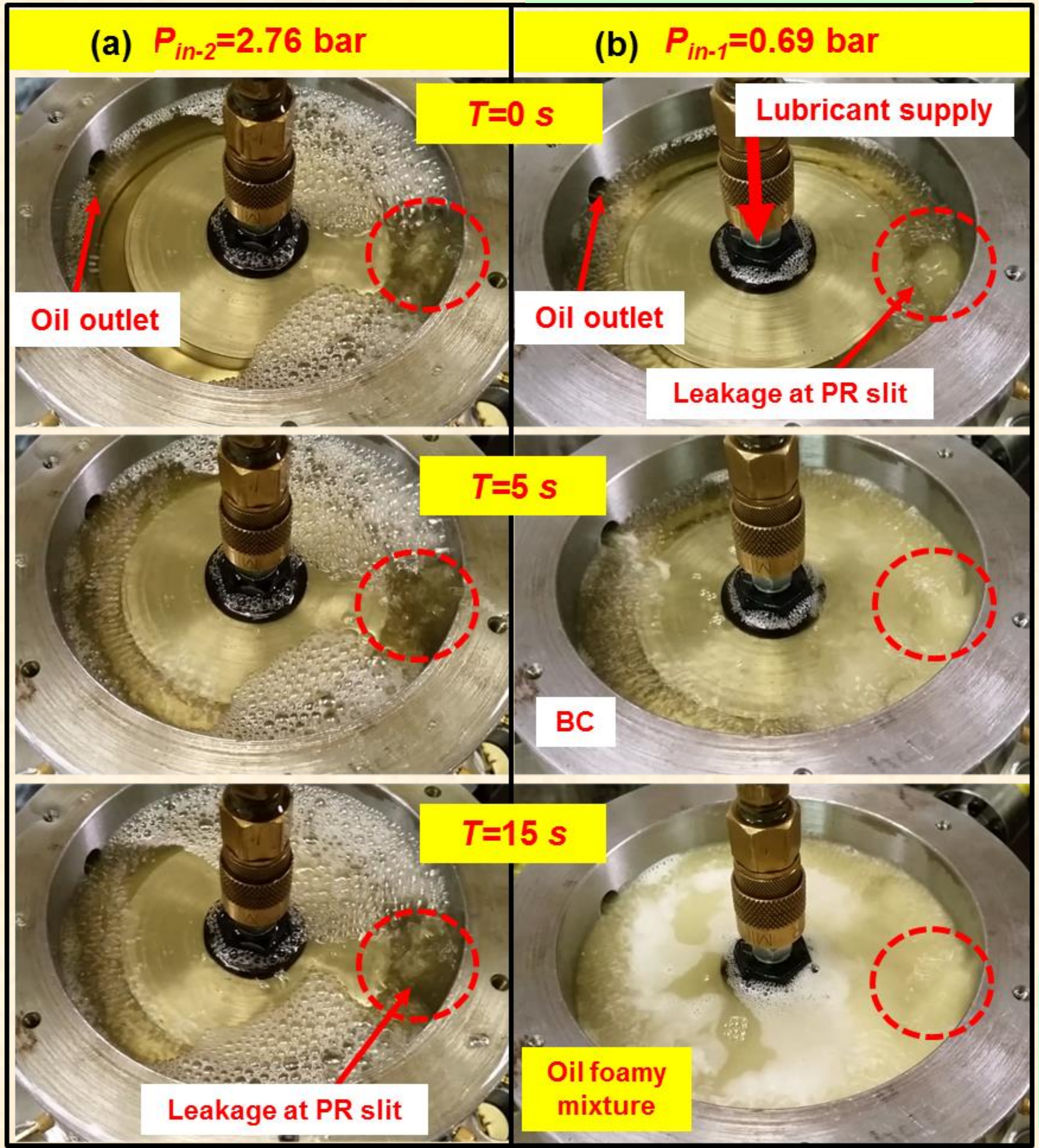
Oil foamy mixture evolves through piston ring slit.

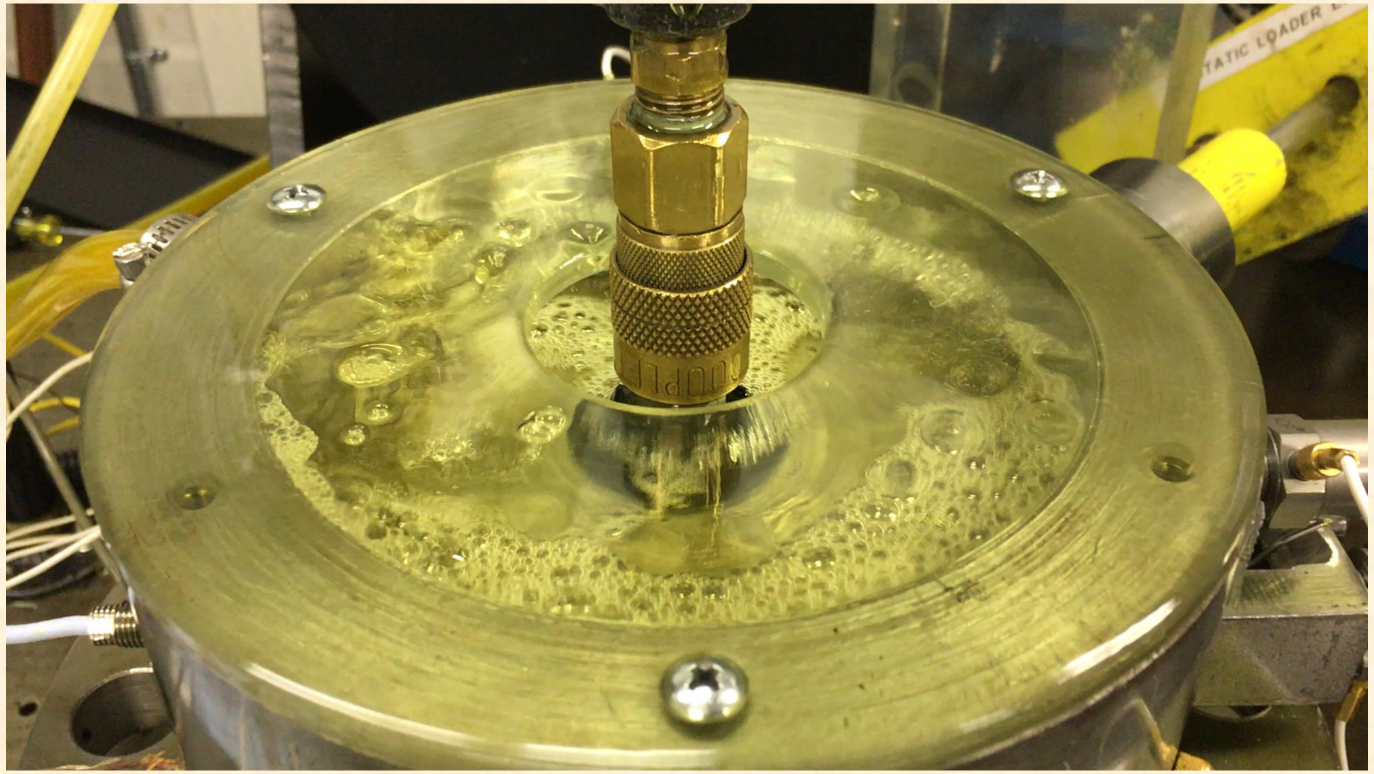
PR-SFD

Time evolution of exit flow

r/c =0.45, ω=80Hz

high feed pressure | low feed pressure

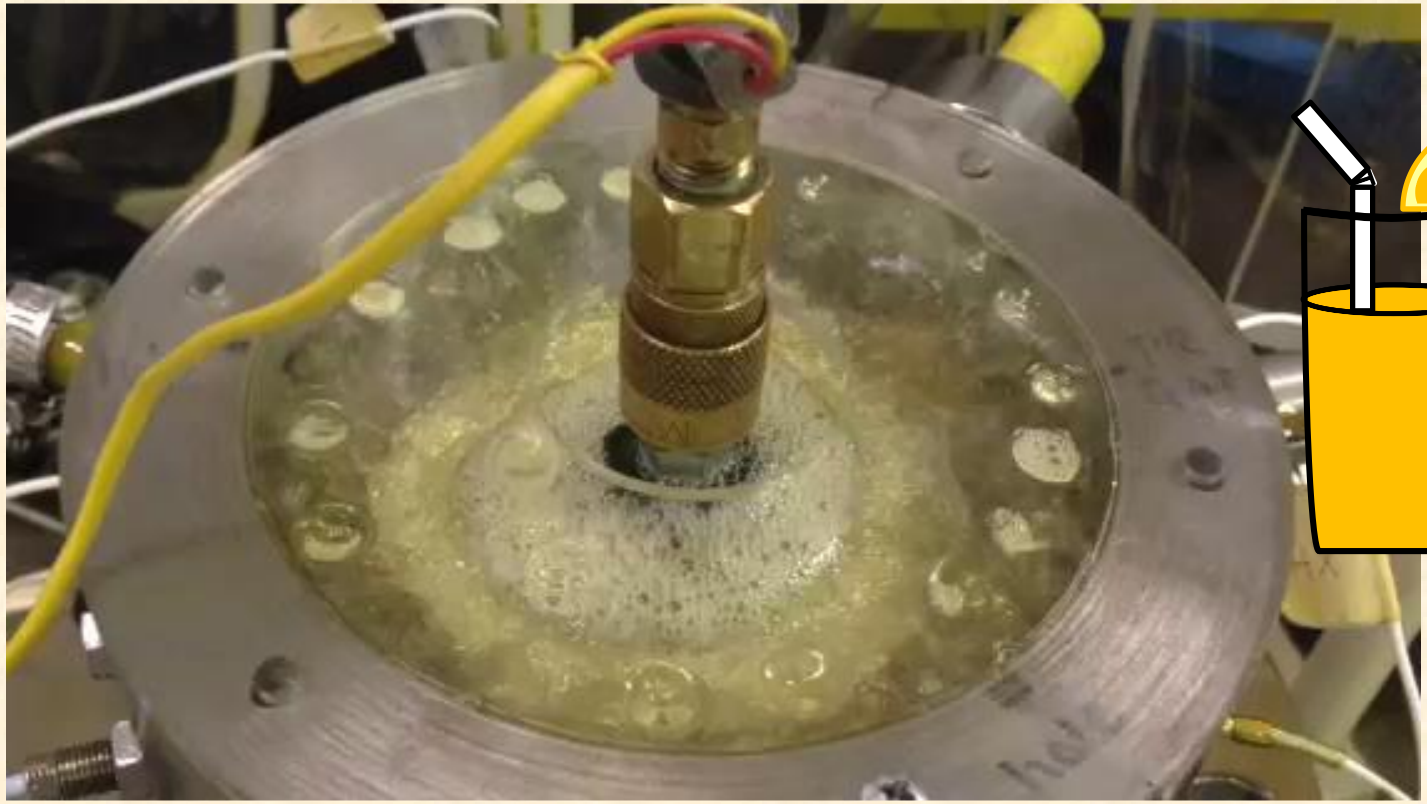


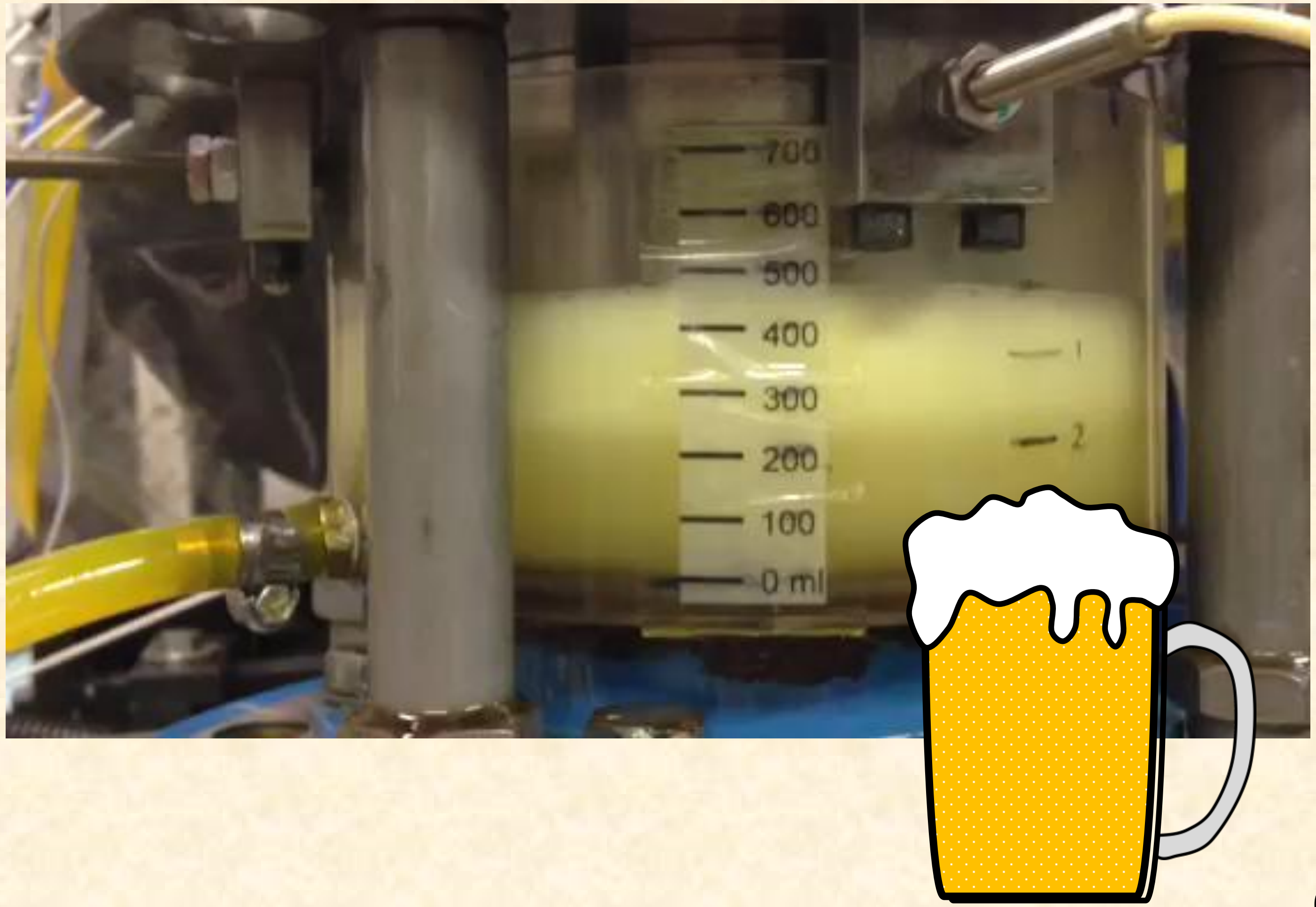


→ **Bubbly mixture makes a foam.**

SFD operational issue

Air ingestion and entrapment





Bubbly lubricant leaves SFD through top and bottom ends.

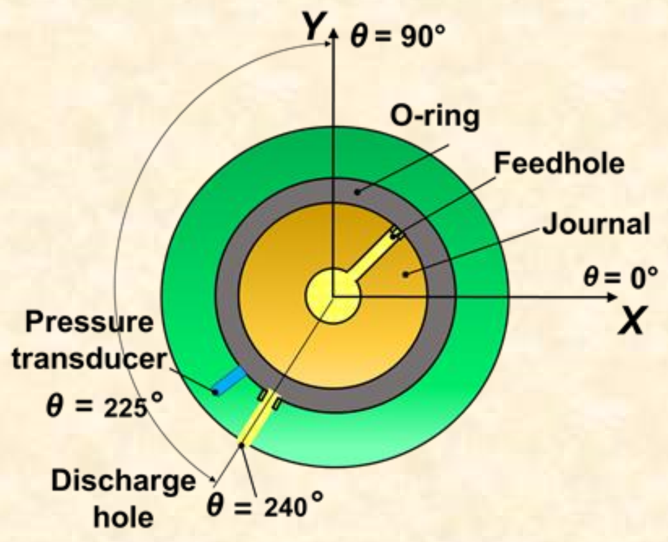
PR-SFD

How much gas is ingested?

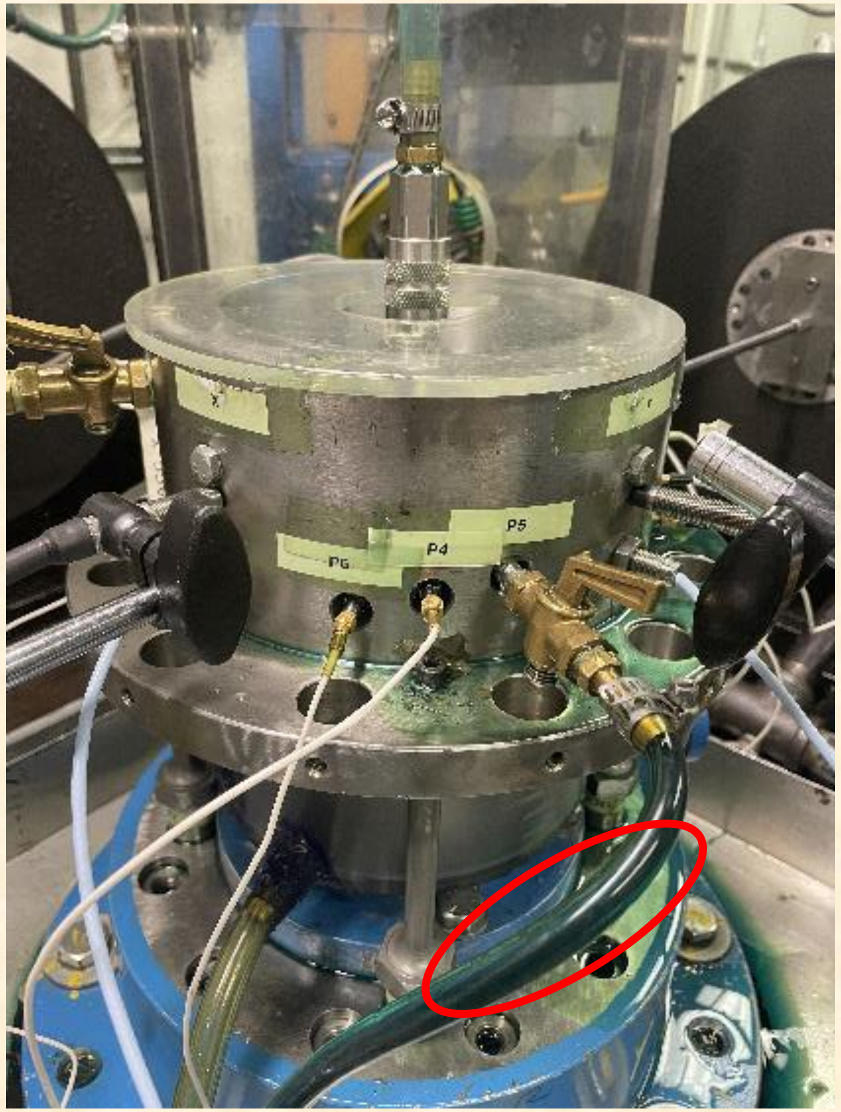
Quantifying air ingestion could aid selecting adequate operating conditions and to produce accurate predictions.

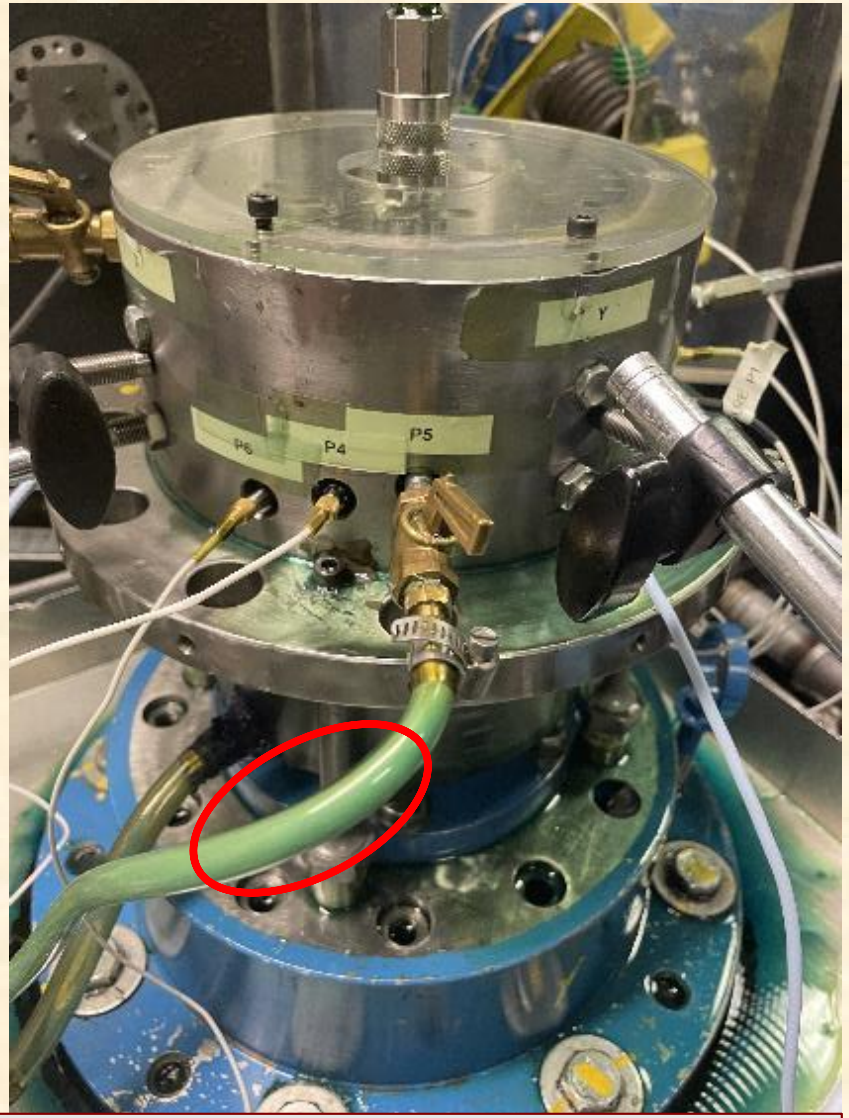
Estimation of gas content in film

OR-SFD



Rodriguez, L., San Andrés, L., **2012**, **TRC-SFD-01-2022.**

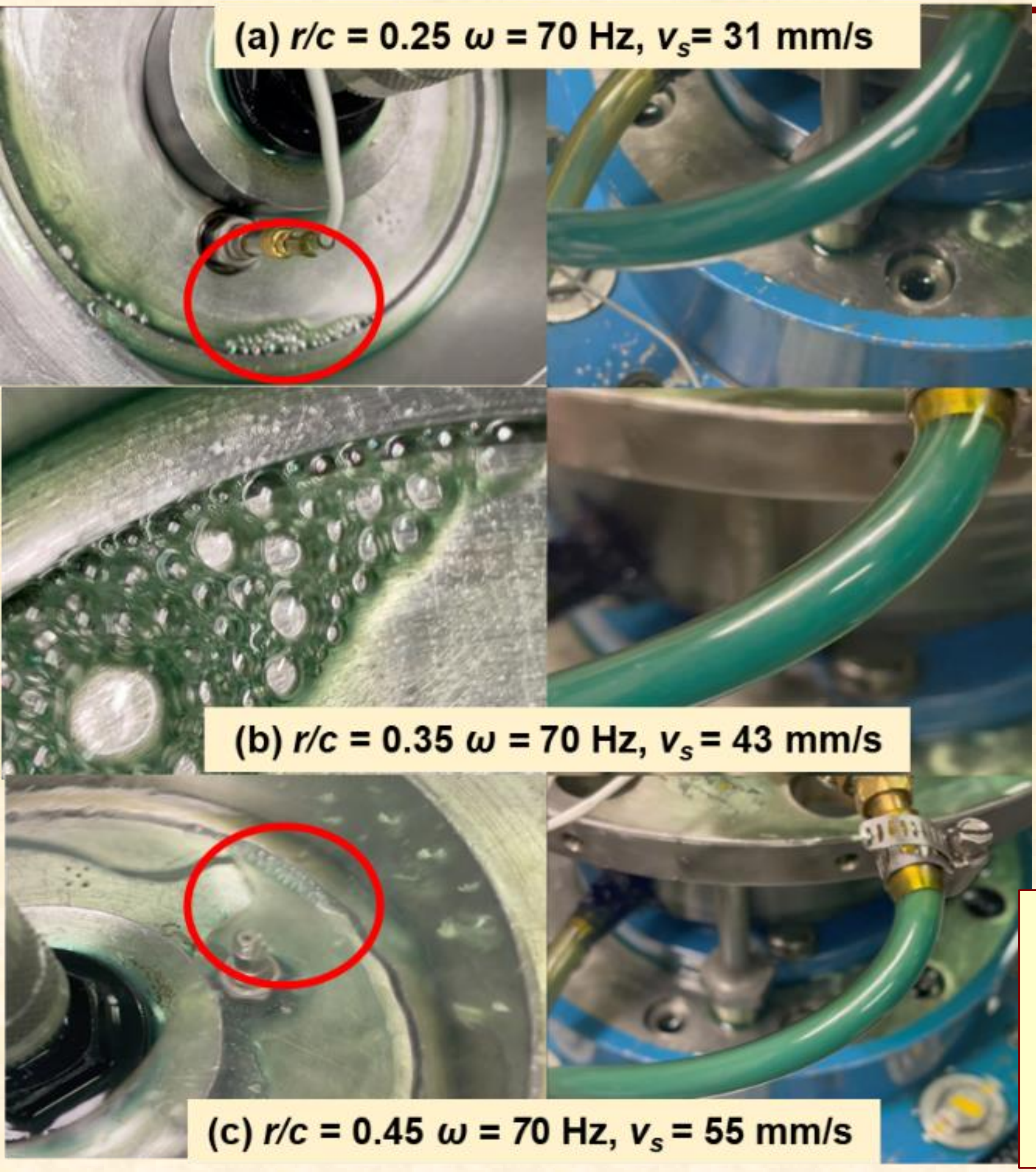


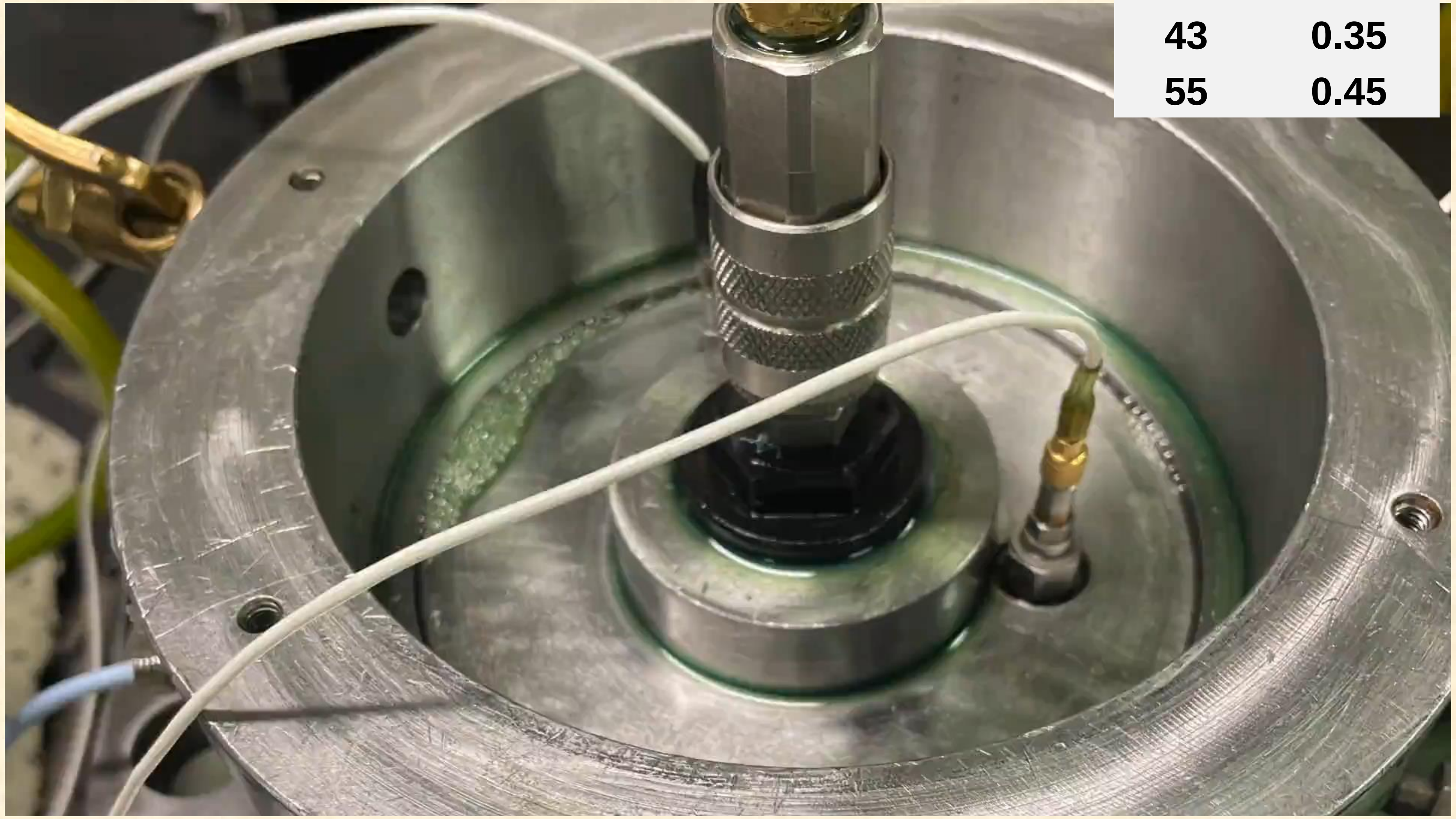


Blue oil (no gas) → **Bubbly oil (foam)**

Visual records showing air ingestion

Squeeze Velocity [mm/s]	Orbit radius (r/c) [-]
31	0.25
43	0.35
55	0.45



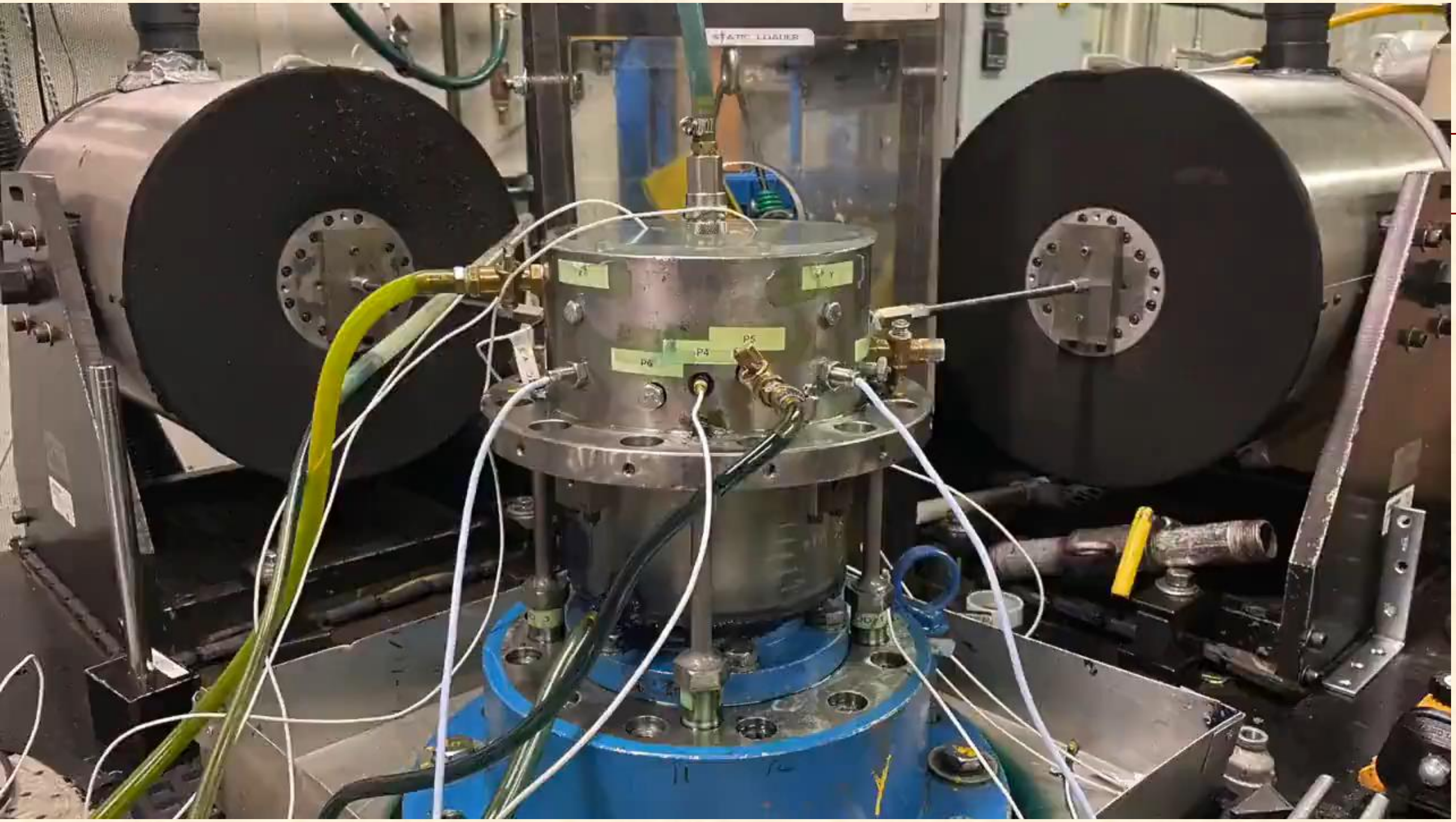


ORs perfectly seal land for v_s< 25 mm/s. For larger v_s , lubricant leaks through damper top end.

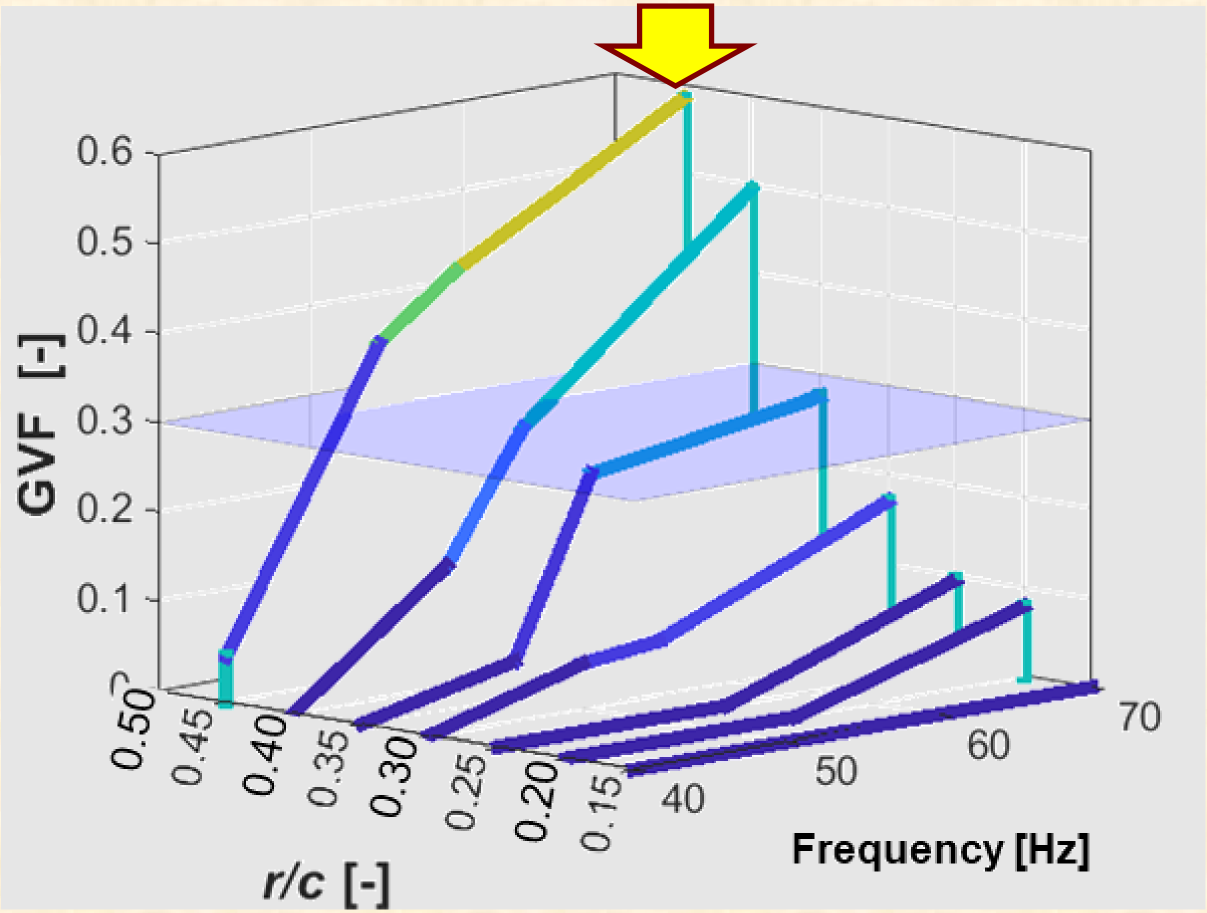
Exit lubricant line shows gas content.

Estimation of gas volume fraction (GVF)

Squeeze Velocity [mm/s]	Orbit radius (r/c) [-]
24	0.20
31	0.25
37	0.30
55	0.45



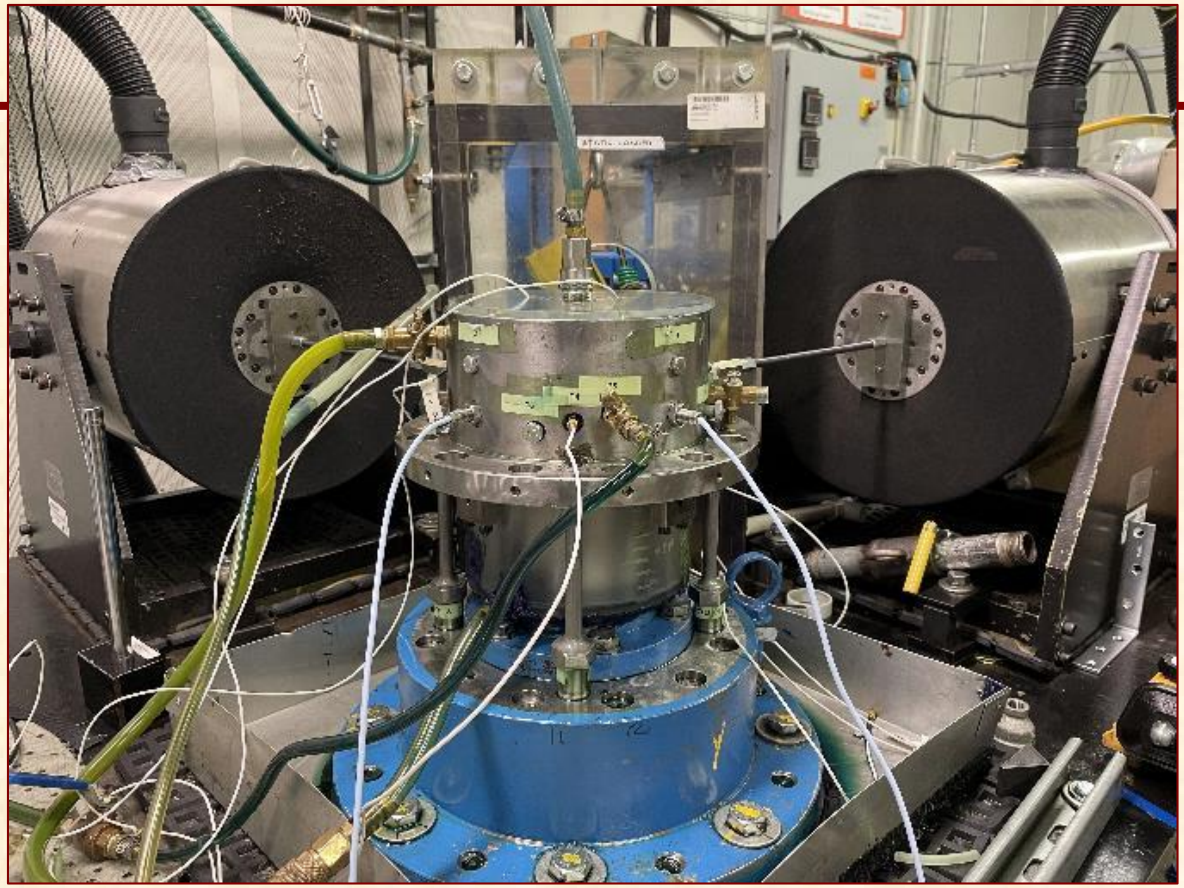
r=0.45 c x ω=70Hz = v_s=55 mm/s → **GVF~58%**



Method *catches* air ingestion, but (clearly) not oil vapor cavitation

Closure

MEASUREMENTS AND MODELS OF SQUEEZE FILM DAMPERS' FORCED RESPONSE AND A BIRD'S EYE VIEW TO AIR INGESTION & ENTRAPMENT



After 25+ years of work,

what is the learning?

Closure (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) Classical lubrication theory does a poor job in producing physically accurate results for test SFDs with feed groove.

→ Advanced accurate model includes fluid inertia, two-phase flow transport and correct boundary conditions.

(c) SFDs generate large added mass coefficients, in particular for configurations with feed and discharge grooves.

→ Grooves generate dynamic pressures → affect force coefficients

Closure (2):

(d) A damper with one feed hole is more effective than other with three feed holes.

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

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

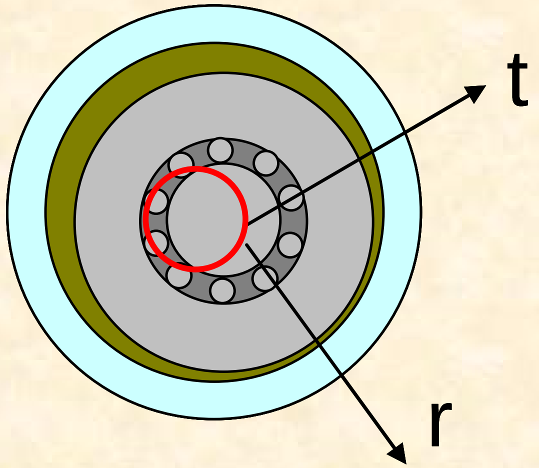
(g) Air ingestion impairs the growth of film pressures for increasing squeeze velocities = (orbit amplitude x whirl frequency) → damping coefficients decrease.

➡ **The experimental record shows SFDs perform as a linear mechanical element.**

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.

Thanks to students

Bryan Rodriguez (2022) LA Turbine

Bonjin Koo (2020) Daikin Applied

Sung-Hwa Jeung (2013, 2017) Ingersoll-Rand, ResMed Motor Technologies

Sean Den (2015) Formosa Plastics Co.

Gary Bradley (2013) TAMU APT ENTC

Paola Mahecha (2011)

Sanjeev Seshagiri (2011) Calnetix

Adolfo Delgado (2005, 2008) GERC, Texas A&M University

Oscar de Santiago (1998, 2002) Dresser-Rand, ETU, MX

Sergio Diaz (1999) Universidad Simon Bolivar, Venezuela

Daniel Lubell (2000) Sundstrand, Capstone MT, Calnetix, Oil Free TM

Miller Robison (1995) Rocketdyne, Solar Turbines

Hector Laos (1993) GE Transp., Oakridge NL

Grigory Arauz (1993) Reta Pumps

UGs: **Israel Silva, Aquiles Lopez, CW Kartens, Scott Tran, Will Radke,** and many, many more.

Acknowledgements

Thanks to

- Pratt & Whitney
- TAMU Turbomachinery Research Consortium

TURBOMACHINERY LABORATORY
TEXAS A&M ENGINEERING EXPERIMENT STATION

Profs. John Vance & Dara Childs (TAMU),

Prof. John Tichy (RPI),

Greg Savela, Phil Varney, Chris Zuck, Zeping Yu, David Hibner (PWUTC)

Tod Steen, Al Storace (GE Aviation)

Walter Meacham (Honeywell)

The end of the quest

When the music's over,

When the music's over, …

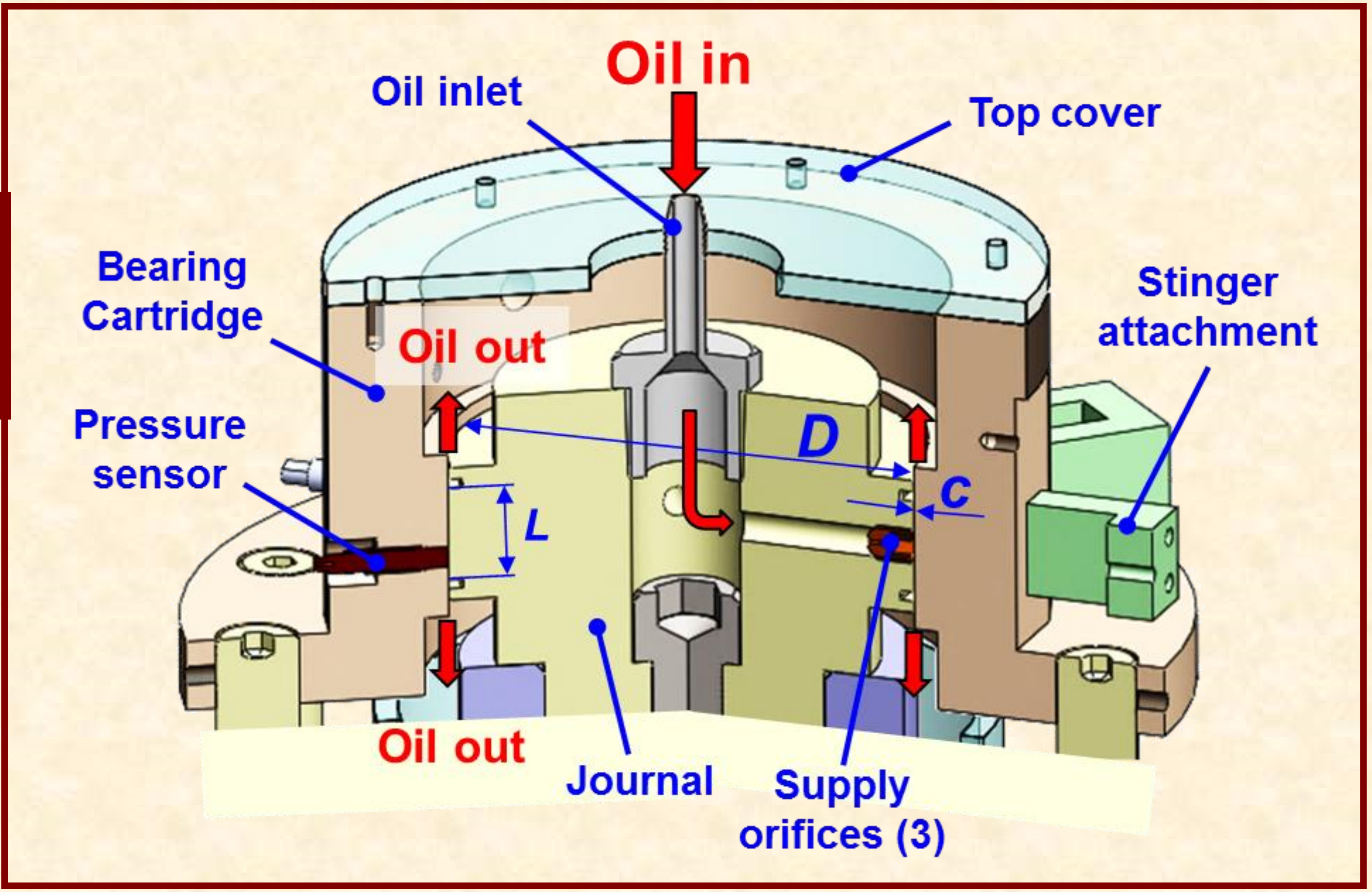
turn out the lights,

turn out the lights...

Jim Morrison (The Doors)

Questions (?)

References



Oil in
Oil inlet
Top cover
Bearing Cartridge
Stinger attachment
Oil out
Pressure sensor
D
C
L
Oil out
Journal
Supply orifices (3)

References SFDs

2022	Rodriguez, L., and San Andres, L., 2022, MEASUREMENTS OF THE DYNAMIC FORCED RESPONSE OF AN O-RINGS SEALED SQUEEZE FILM DAMPER SUPPLIED WITH A LOW PRESSURE, Technical Report TRC-SFD-01-22
2022	Lu, X., San Andrés, L,, and Koo, B., 2022,"On the Influence of Lubricant Feedhole Size and End Plate Seals' Clearance on the Dynamic Performance of Integral Squeeze Film Dampers," (Lecture), Asia Turbomachinery Pump Symposium, Kuala Lumpur, Malaysia, May
2022	Koo, B., and San Andrés, L., 2022, "A Model and Experimental Validation for a Piston Rings-Squeeze Film Damper: a Step toward Quantifying Air Ingestion," **ASME GT2022-8199**0.
2021	San Andrés, L., and Rodriguez, B., 2021, "On the Experimental Dynamic Force Performance of a Squeeze Film Damper Supplied Through a Check Valve and Sealed With O-Rings," ASME J. Eng. Gas Turbines Power, Vol. 143(11): 111011 **(ASME GT2021-58627)**
2021	San Andrés, L., Lu, X., Koo, B., and Tran, S., 2021, "On the Effect of the Gap of End Seals on the Force Coefficients of a Test Integral Squeeze Film Damper: Experiments and Predictions," ASME J. Eng. Gas Turbines Power, Vol. 143(1): 011014, https://doi.org/10.1115/1.4048700 (**ASME GT2020-14182**).
2020	San Andrés, L., and Koo, B., 2020, "Model and Experimental Verification of the Dynamic Forced Performance of a Tightly Sealed Squeeze Film Damper Supplied with a Bubbly Mixture," ASME J. Eng. Gas Turbines Power, Vol. 142(1): 011023, **ASME 2019-90330**
2019	San Andrés, L., Koo, B.J., and Seung, J.-H., 2019, "Experimental Force Coefficients for Two Sealed Ends Squeeze Film Dampers(Piston Rings and O-rings): An Assessment of Their Similarities and Differences," ASME J. Eng. Gas Turbines Power, Vol. **141**(2), 02104, DOI: 10.1115/1.4040902 (**ASME GT20018-76224)**
2018	San Andrés, L., and Koo, B., 2018, "Effect of Lubricant Supply Pressure on SFD Performance: Ends Sealed with O-rings and Piston Rings" IFTOMM2018-018, 10TH IFToMM INTERNATIONAL CONFERENCE ON ROTOR DYNAMICS, Sept 24-27, 2018, Rio de Janeiro, Brazil

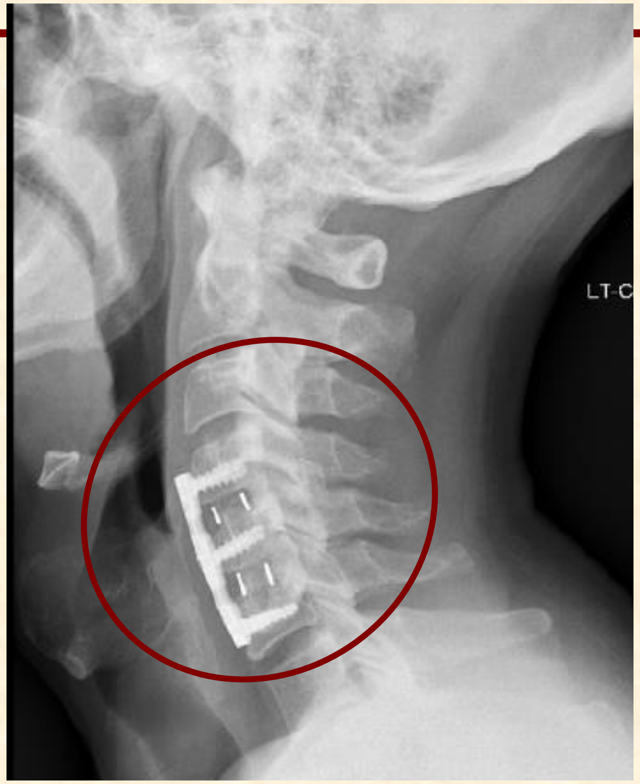
References SFDs

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

Being a pain in the neck



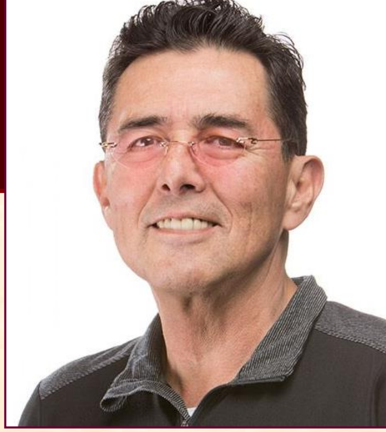
… to being screwed

… to being totally screwed

Luis San Andres

June 6, 2022

Mast-Childs Chair Professor



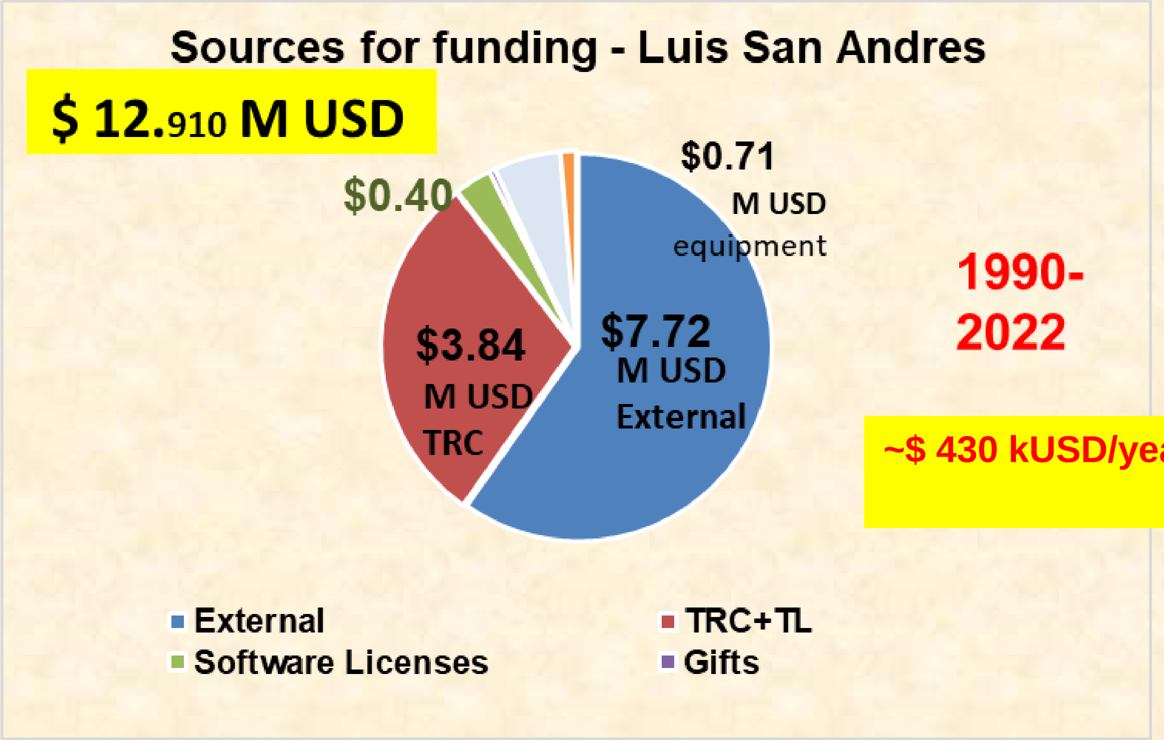
1991-2022

2017-22	Career		Publications
45	196		Journal (peer reviewed)
52	138		Conference (peer reviewed)
7	42		Conference (NOT peer reviewed)
			Accepted/awaiting publication
104	376		

rotorlab.tamu.edu

BEST PAPER AWARDS

2022 IGTI S&D – Bearings & Seals
2019 IGTI S&D – Bearings & Seals
2018 IGTI S&D – Bearings & Seals
2017 IGTI S&D – Bearings & Seals
2016 IGTI S&D – Bearings & Seals
2012 IGTI S&D – Bearings & Seals
2011 IJTC J. Tribology (with KIST)
2010 IGTI S&D – Microturbomachinery
2010 IFToMM Best Paper
2008 IGTI S&D - Rotordynamics
2005 IGTI S&D - Rotordynamics
2003 IGTI S&D - Rotordynamics



~$ 430 kUSD/year