ASME
FEDSM 2025

FLUIDS ENGINEERING DIVISION
SUMMER MEETING

DOUBLETREE BY HILTON PHILADELPHIA CENTER CITY
PHILADELPHIA, PA, USA

Wet Annular Seals in Multiphase Pumps: Leakage and Rotordynamic Force Coefficients and a Method to Promote Seal Direct Stiffness

Dr. Luis San Andrés is Professor Emeritus at Texas A&M University. He was the Mast-Childs Chair and Professor of Mechanical Engineering (2014-2023). Since 1990-2025, students and Luis conducted research in lubrication of bearings and seals, vibrations and rotordynamics.

TEES | Turbomachinery Laboratory
Texas A&M Engineering Experiment Station

Dr. San Andrés received **the ASME-IGTI 2022 Aircraft Engine Technology Award** for creative contributions to aircraft engine technology, the **2023 Mayo D. Hersey Award** for enduring contributions to foil bearing technology, and the **2025 ASME Henry R. Worthington Medal** for groundbreaking work on pump seals for cryogenic turbopumps and multiple-phase pump seals.

TURBOMACHINERY LABORATORY
TEXAS A&M ENGINEERING EXPERIMENT STATION

Making a Difference in Industrial & Aerospace Turbomachinery

http://turbolab.tamu.edu

Dr. Eric Petersen, Director
June 2025

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Facilities Turbomachinery Laboratory

37000 ft^2 total area with 12 test-cell suites, each 520 ft^2 sound-and seismically-isolated,

+

600VAC x 400 A bus = 240 kW= (330 HP motor) in each cell.

+ Air Compressors

1.1 m^3/s@8.6 bar

(2350 scfm @ 125 psi),

0.40 m^3/s @ 20 bar,

0.54 m^3/s @ 24 bar,

75 bar (1,100 psi)

New facilities in 2025 – Combustion & Propulsion – liquid and solid.

The Turbo Lab

Research Goals:

Conduct fundamental and applied research of the highest quality, meeting the needs of the turbomachinery industry for the 21st century (performance and reliability).

Thrust Areas:

(1) Turbine Combustion & Cooling Technologies.

(2) Aerodynamic Performance and Efficiency of Turbomachinery (Pumps).

(3) Rotordynamics of Turbomachinery including Fluid-Structure Interaction Forces (Compressors & Pumps)

(4) Oil-Free Micro-Turbomachinery (< 400 kW)

The Turbo Lab : Summary Outreach

Turbomachinery & Pump Symposia

- Annual event
- Houston, TX
- Approx. 4,500 attendees
- 350+ exhibitors
- Began:
  Turbo 1971
  Pump 1984

Asia Turbomachinery & Pump Symposium

- Biennial event since 2016
- Kuala Lumpur
- ~ 900+ attendees
- 50+ exhibitors

Continuing Education Stand-alone Short Courses

- 4-5 courses annually
- Houston, TX

Faculty & Graduate Student Involvement

- 6-8 ME faculty
- 2 Aero faculty
- 90+ graduate students

TYP $700k/faculty per year

Turbomachinery Research Consortium (TRC)

30+ companies (O&G OEMs and end users)

- $35k annual fee
- Approx. 13-17 research projects funded annually.

Global footprint!

Middle East Turbomachinery Symposium (METS)

Past (2011, 2013, 2015): 1000 attendees + 60+ exhibitors

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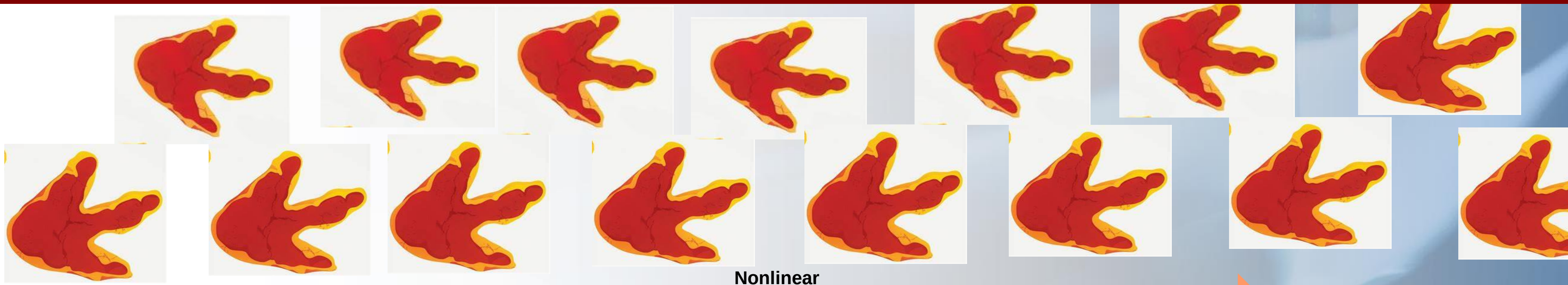
A dinosaur walk since last millennium



Funding Sources

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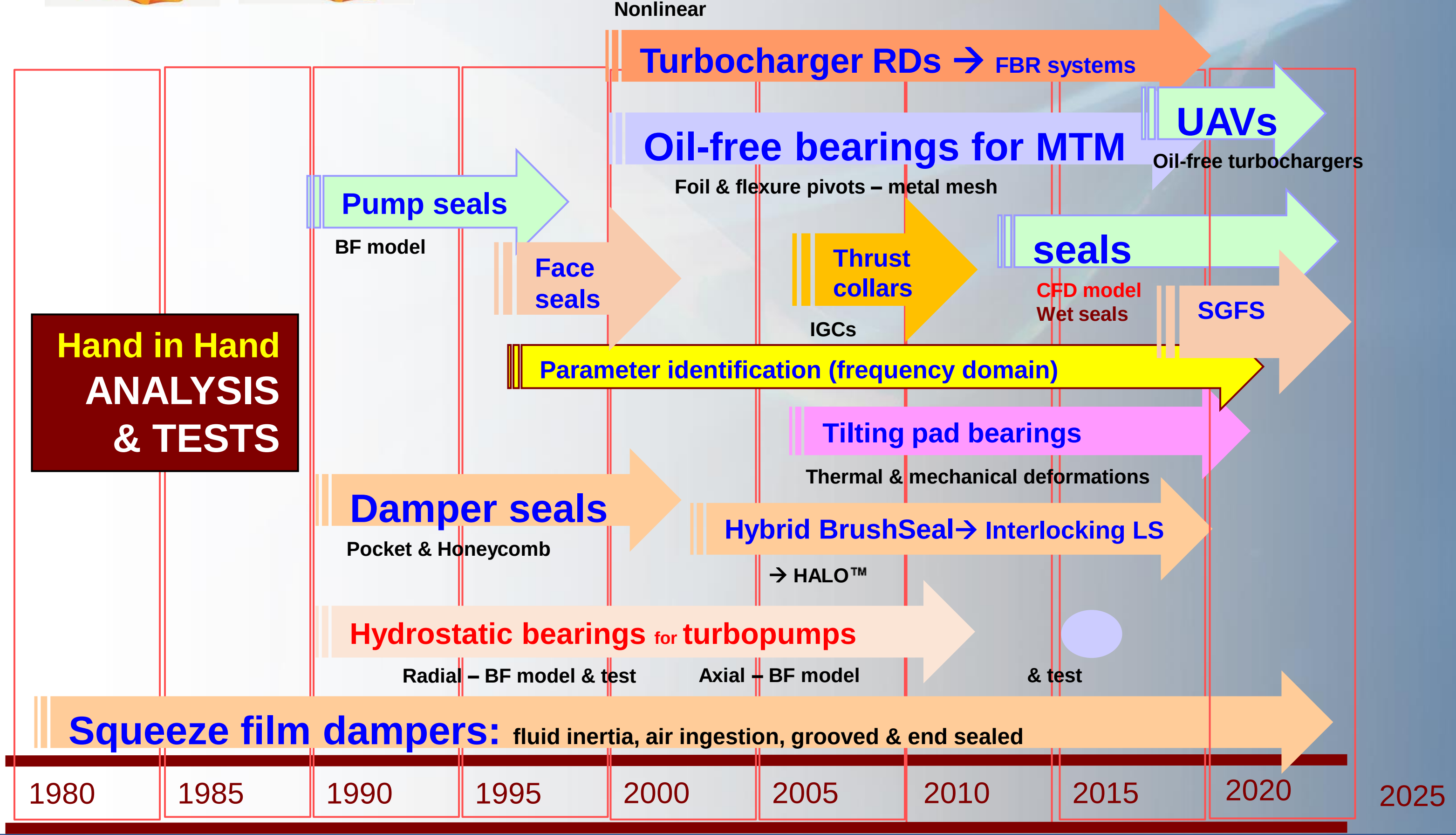
NSF, NASA GRC,
Pratt & Whitney,
Northrop Grumman,
Rocketdyne,
Honeywell TT,
Danfoss TurboCor
Borg-Warner TC,
Torishima Pumps,
MHI, Hitachi RL,
Samsung, Key Yang,
Hyundai HI, Capstone MT
Siemens, **TRC**



Hand in Hand
ANALYSIS
& TESTS

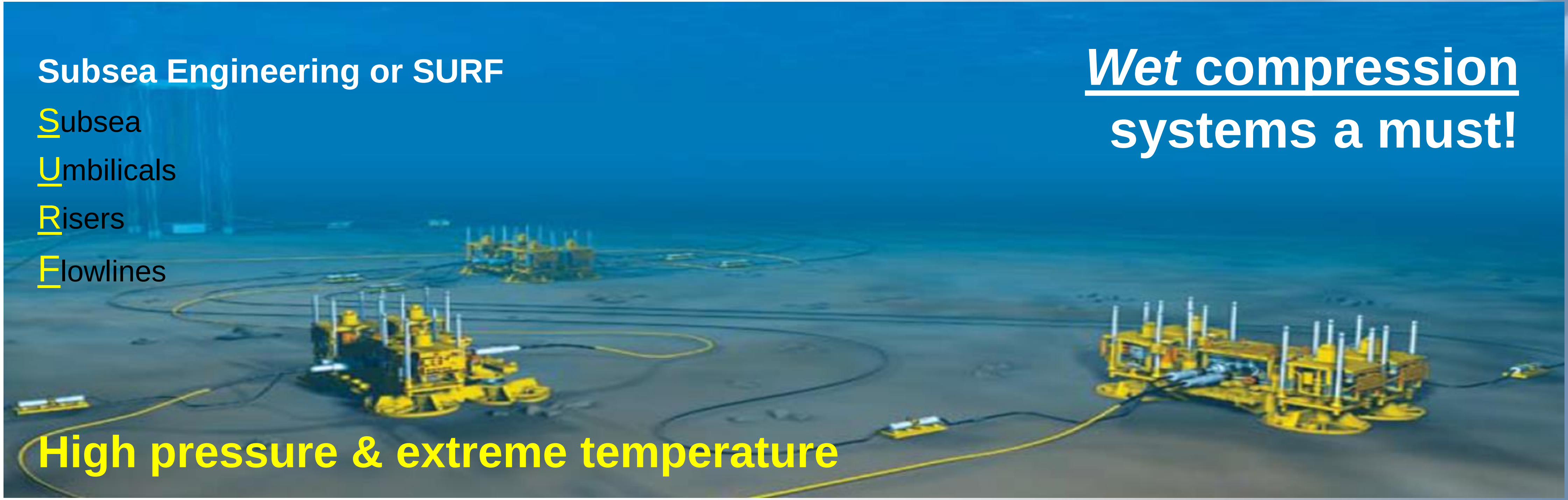
Nonlinear
Turbocharger RDs → FBR systems
UAVs
Oil-free turbochargers
Oil-free bearings for MTM
Foil & flexure pivots – metal mesh
Pump seals
BF model
Face seals
Thrust collars
IGCs
seals
CFD model
Wet seals
SGFS
Parameter identification (frequency domain)
Tilting pad bearings
Thermal & mechanical deformations
Damper seals
Pocket & Honeycomb
Hybrid BrushSeal→ Interlocking LS
→ HALO™
Hydrostatic bearings for turbopumps
Radial – BF model & test
Axial – BF model
& test
Squeeze film dampers: fluid inertia, air ingestion, grooved & end sealed

1980 1985 1990 1995 2000 2005 2010 2015 2020 2025

Today’s Outcome

1. Why wet (bubbly) seals? Where are they found?
2. How does the gas content affect seal leakage and drag?
3. How does gas content affect the stiffness and damping coefficients of a *wet* seal?
4. Why a wavy surface seal is a better option than a grooved seal in a two-phase viscous flow pump?
5. Low frequency self-excited vibrations & their source.
6. Why does gas injection increase the centering stiffness of seals in pumps & turbines?

A need: subsea pumping & compression



Subsea Engineering or SURF

Subsea
Umbilicals
Risers
Flowlines

***Wet* compression**
systems a must!

High pressure & extreme temperature

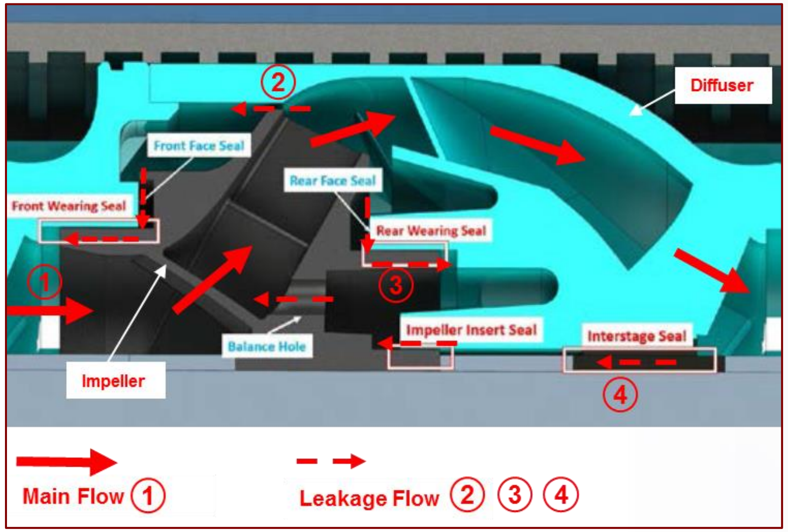
Bloomberg : Since 2018 offshore oil production tops shale oil on generation of jobs.

Extreme engineering enables five year or longer reliability for subsea production facilities (North Sea & Brazil → Gulf of Mexico → Arctic).

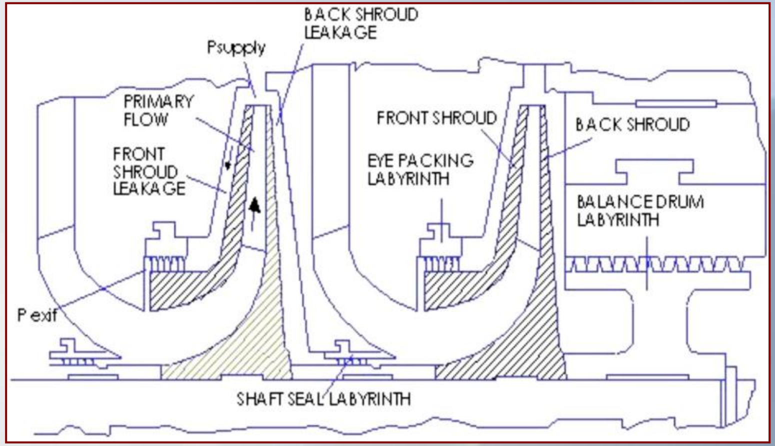
Subsea O&G Production Facilities

GVF: Gas volume fraction (pump)

LVF=1-GVF: Liquid volume fraction (compressor)



Electrical submersible pump stage (GVF: 0 to 100%)



Compressor stage (LVF: 0 to 5%)

Bubbly/Wet **seals affect pump/compressor performance (leakage and torque) and reduce system rotordynamic stability.**

Subsea O&G production facilities with improved reliability and reduced operational cost demand a concerted effort to quantify effect of two-phase flow in sealing components.

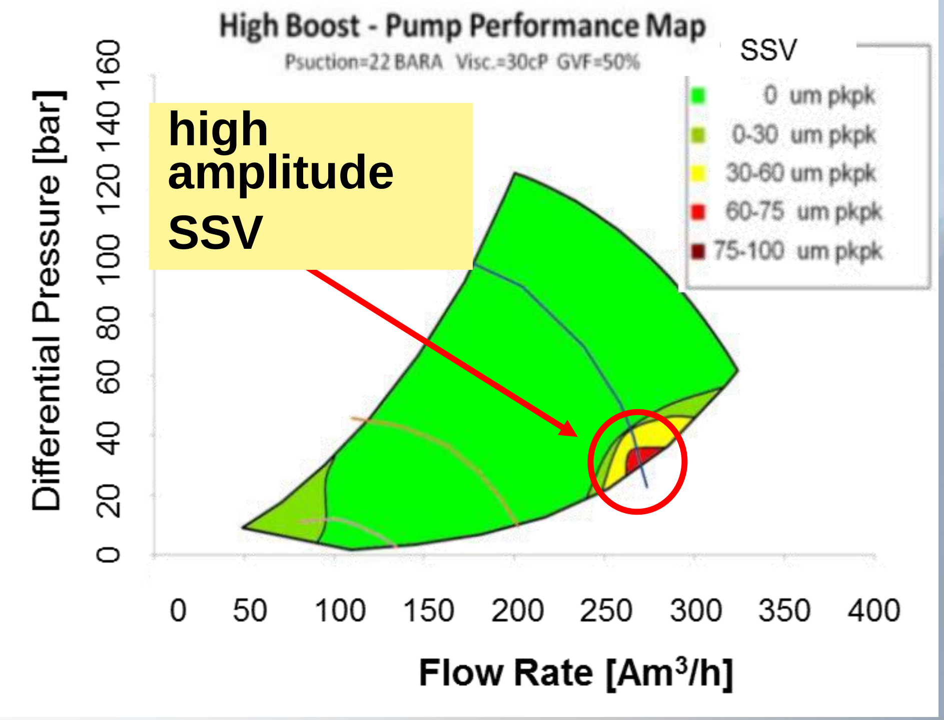
Prior art:
a shallow review

Balance drum seal in multiphase pump (2013)

Helico-axial pump
(1.5 to 4.6 krpm)

Pump operates stable with liquid. (ΔP 150 bar, μ_l =30 cP)

Rotor asynchronous vibrations (SSV) appear under some two-phase flow conditions: **low differential pressure with a high-viscosity mixture.**



Bibet et al. (2013)

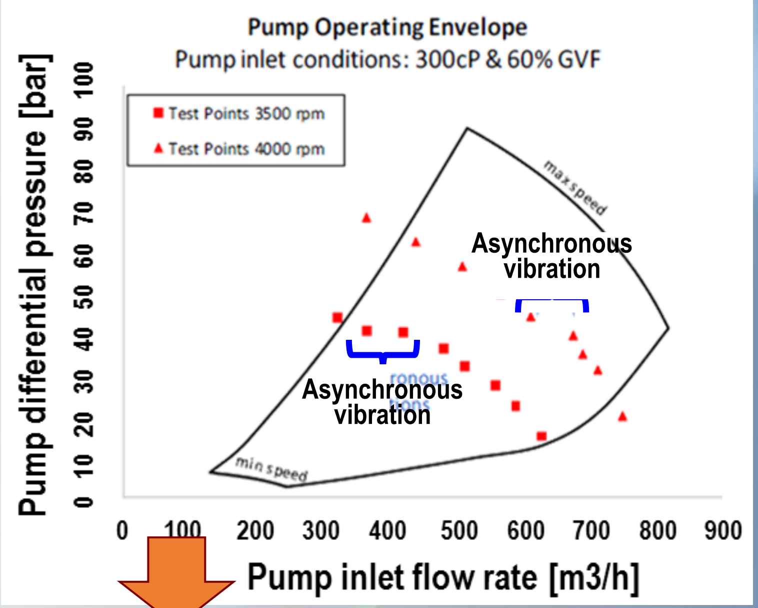
Bibet, P. J., et al., 2013, "Design and Verification Testing of a New Balance Piston for High Boost Multiphase Pumps," Proc. 29th International Pump User Symposium, Houston, TX.

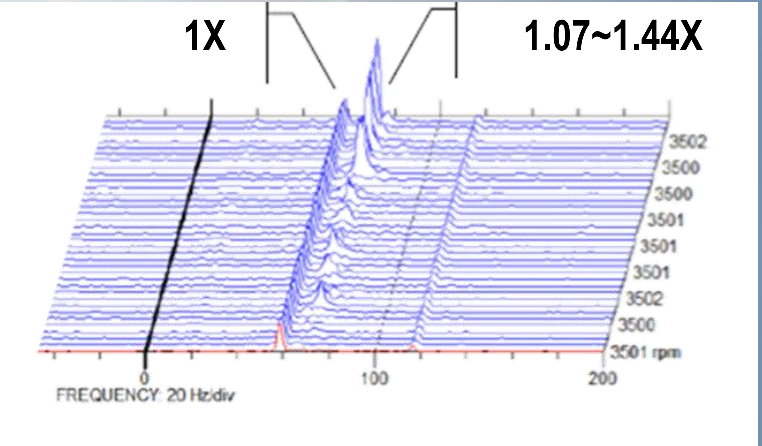
Balance drum seal in multiphase pump (2018)

Helico-axial pump (1.5 to 4.6 krpm)

ΔP 110 bar, μ_l=800 cP, GVF= 0→1.

For pump ΔP ~ 40 bar, inlet GVF ~ 0.6
Rotor super-synchronous vibration (1.07 ~1.44 X).
Seal mass flow 'jumps' when *ΔP* is held constant.
Negative seal direct stiffness predicted.





REMEDY: Changed seal clearance profile affects axial flow velocity development and removes super synchronous vibration.

Ekeberg, I., Bibet, P., Knudsen, H., Torbergsen, E., Kjellnes, H.F., Angeltveit, R., and Klepsvik, K., 2018, "Design and Verification Testing of Balance Piston for High-Viscosity Multiphase Pumps," *Proc.* 47th *Turbomachinery* & 34th International Pump Symposium, Houston, TX.

Research on Wet Gas / Bubbly Seals at the Turbo Lab

Childs and students (2012-2018)

J. Eng. Gas Turb. Power, 2017

J. Tribol., 2018

Measured leakage and rotordynamic force coefficients of wet seals with an air in silicon oil mixture. GVF: 0 → 10%, LVF: 0 → 8%.
Max pressure: 70 bar, shaft speed 20 krpm (*R*Ω=96 m/s)

San Andrés and students (2014-2022)

Tribol. Trans., 2016

J. Eng. Gas Turb. Power: 2017, 2018, 2019,2022

2018 ATPS/TPS

AJKFluids 2019

WTC 2022

Quantify leakage and dynamic force coefficients of wet seals [five types] with air in ISO VG10 oil mixture. GVF: 0 → 1,
Max supply pressure: 5 bar, shaft speed: 5.2 krpm (*R*Ω=35 m/s).

Application: subsea multiphase pumps and wet gas compressors

In sum … and …… this lecture

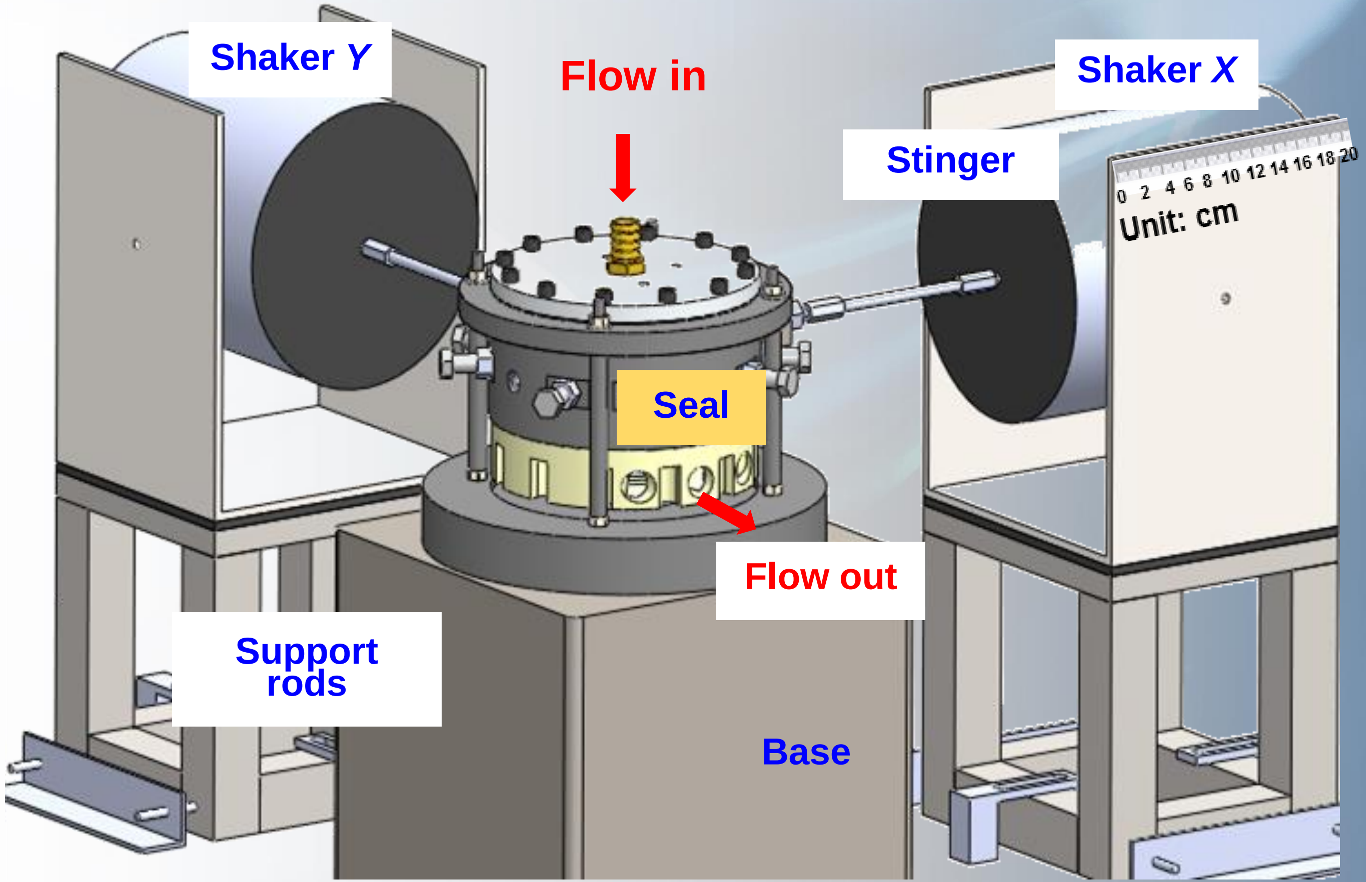
In the subsea oil and gas industry, multiphase pumps and wet gas compressors enable long distance tie back system and <u>eliminate</u> oil and gas separation stations.

Seals must be able to operate without affecting the system efficiency and dynamic stability.

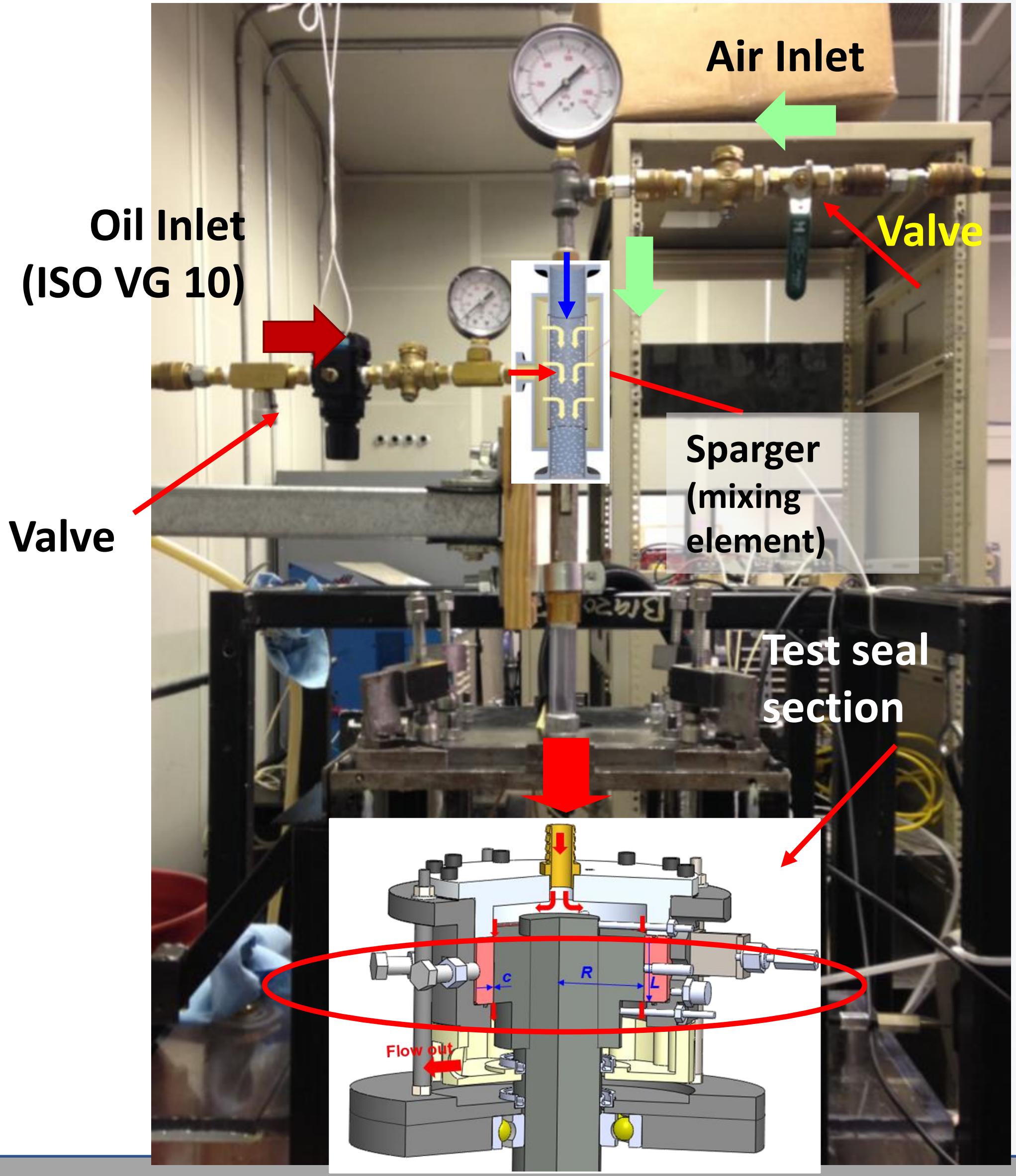
The lecture presents measurements of leakage and force coefficients for two annular seals operating with an air in oil mixture ranging from pure liquid to mostly air. Some interesting features reported!

Wet Gas Test Rig

- Controlled motion test rig with "floating" seal housing and spinning rigid shaft.
- Shakers exert frequency-dependent loads to excite system toward obtaining seal force coefficients.



Wet Gas / Bubbly Liquid Seal Rig



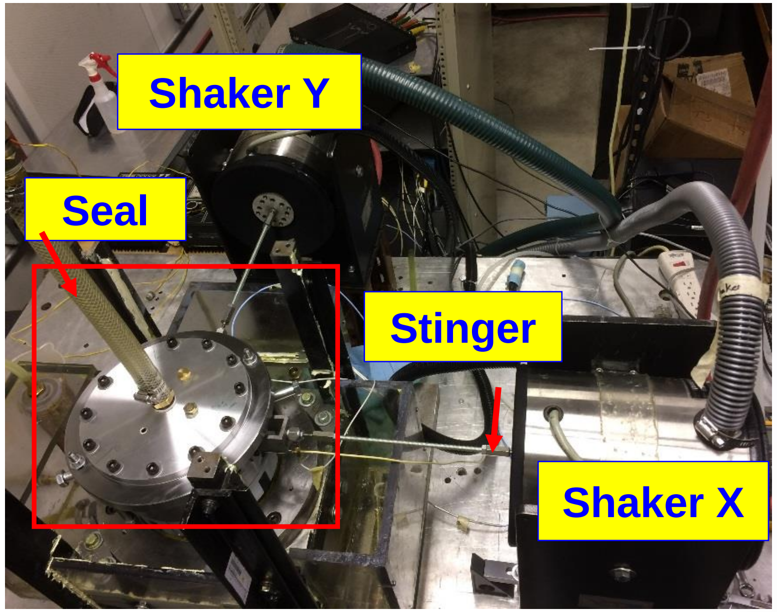
Rotor Diameter (*D*)	127 mm
Rotor Length (*L*)	46 mm
Supply pressure (P_s)	1.0~4.5 bar (abs)
Oil ISO VG 10 density(ρ_l)	**830 kg/m^3**
viscosity (μ_l)	**10.6 cP** at 34 ºC
Air density (ρ_{ga})	**1.2 kg/m^3** at 1bar
viscosity(μ_{ga})	**0.02 cP** at 20 ºC
Sparger pore size	2 µm
Air bubble size	Up to 4 mm

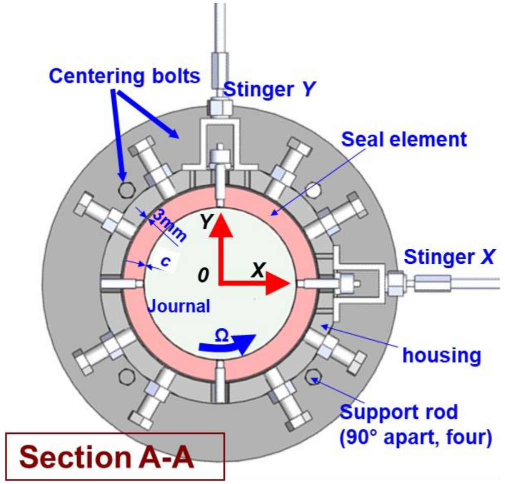
***L/D*=0.36 short length**

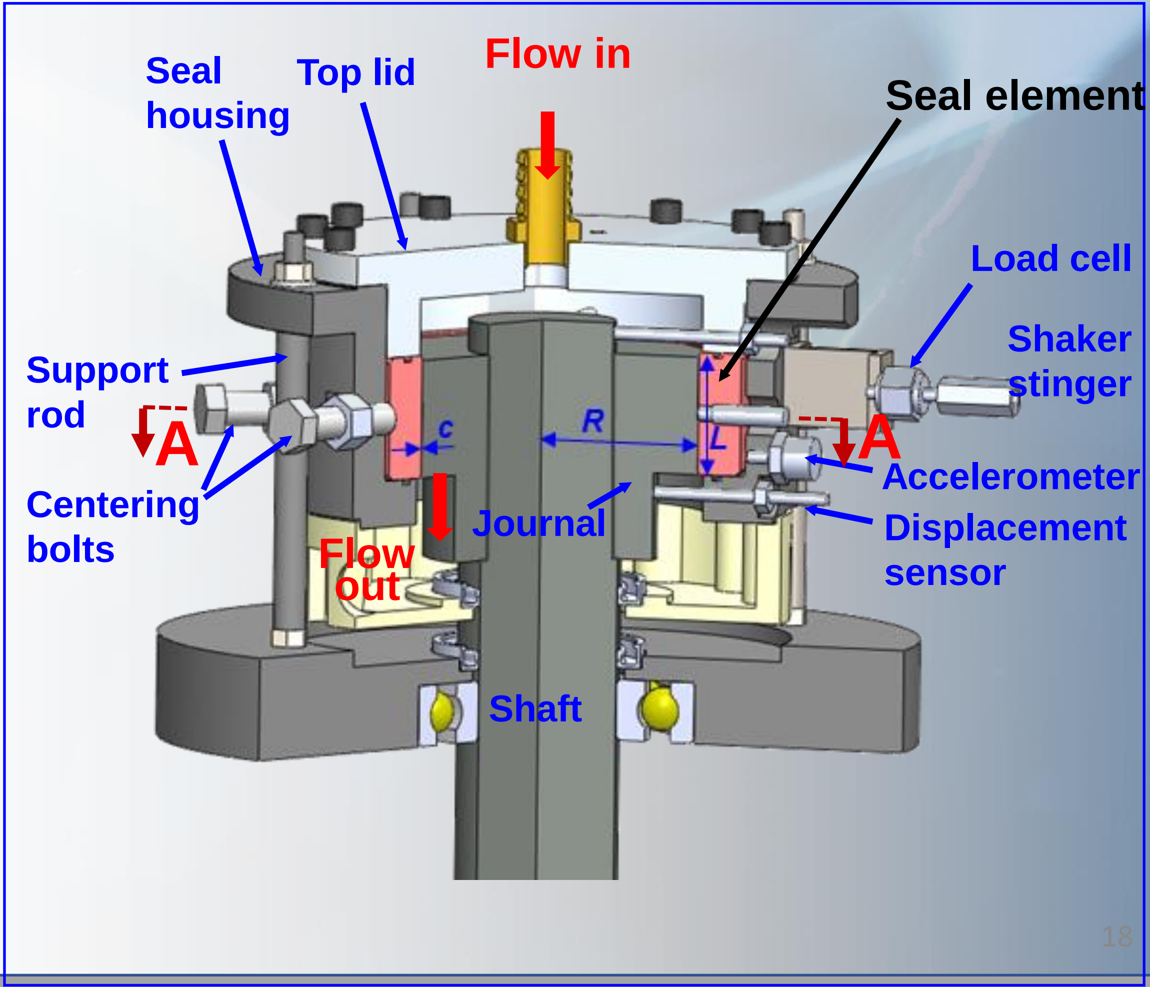
APPLICABLE to impeller interstage seals and/or neck ring seals.

Cartridge Housing Seal

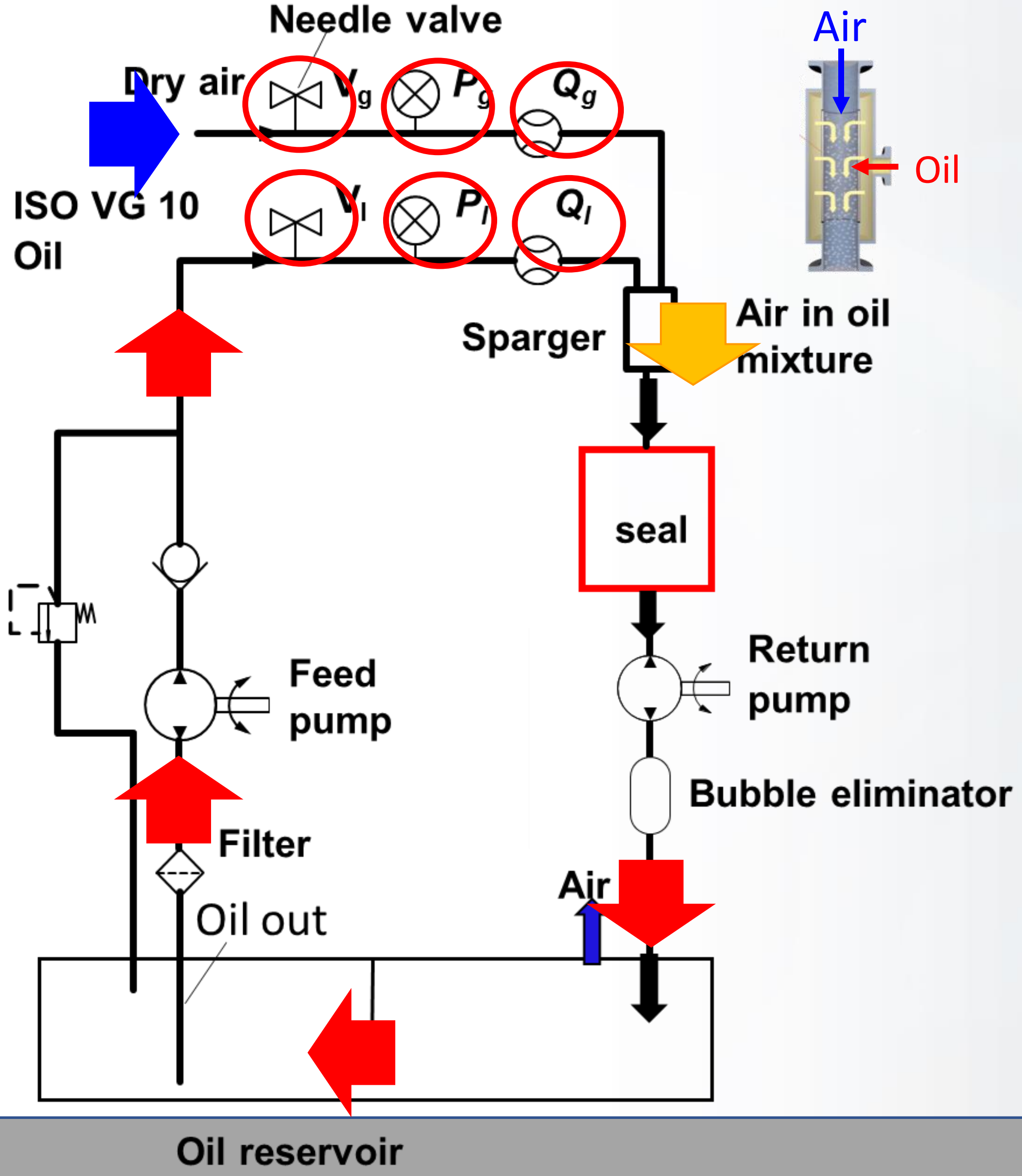
Rotor speed, Ω_{max}	3.5 krpm
surface speed, $½D\Omega_{max}$	23.3 m/s







Air & Oil Systems and their Mixing



α : Gas volume fraction (GVF)

P_s: pressure at seal inlet plane

P_a: ambient pressure= 1 bar(a)

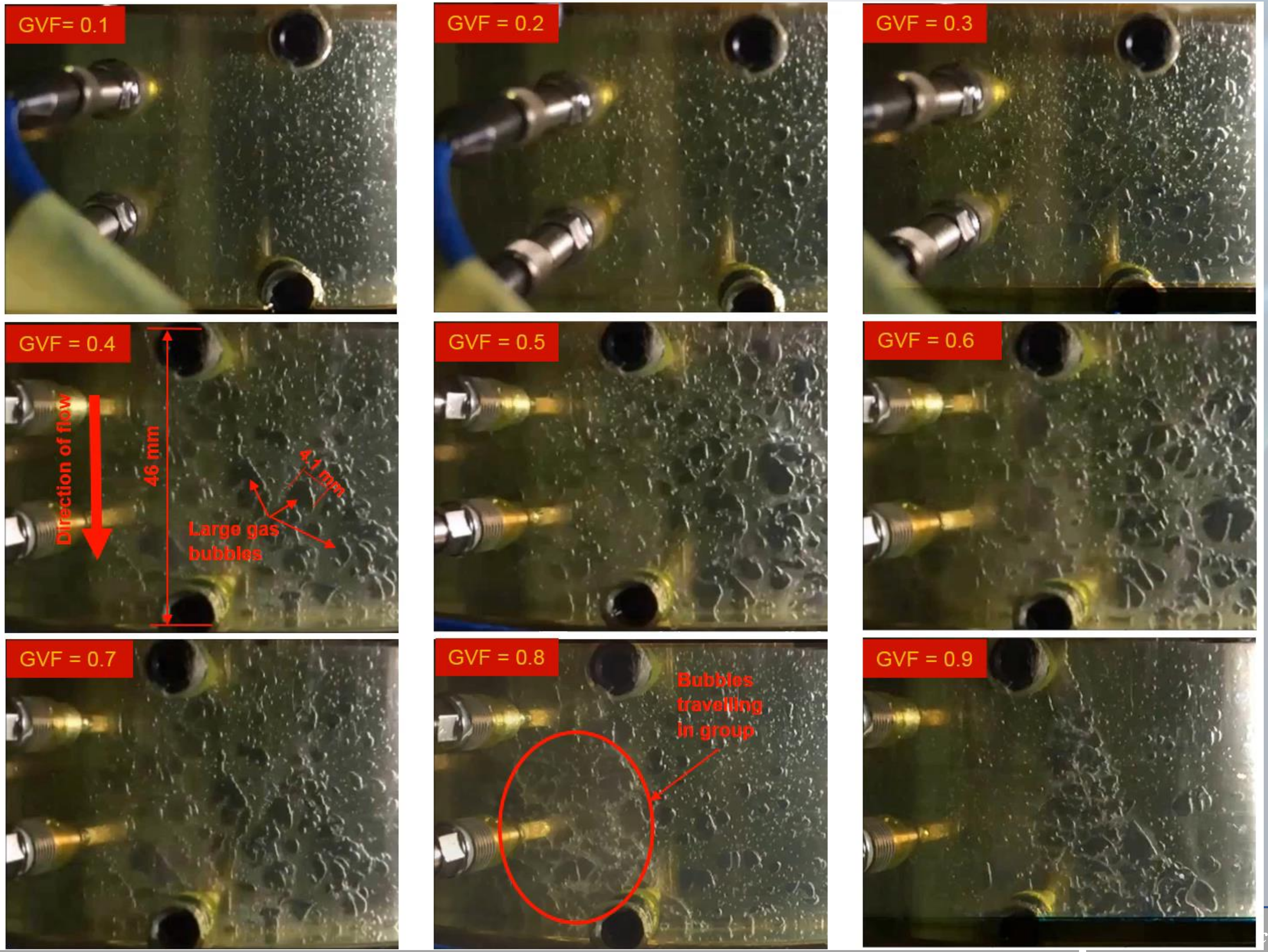
Q_g: gas flow rate at P_s

Q_l: liquid flow rate

GVF at inlet:

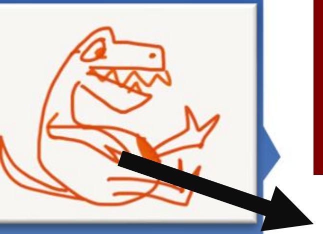
$$\alpha_{in} = \frac{Q_g\left(P_a/P_s\right)}{Q_l+Q_g\left(P_a/P_s\right)}$$

Visualization → GVF = 0-0.9. *Ps*/Pa=2.5. Speed 0 rpm (pain seal)

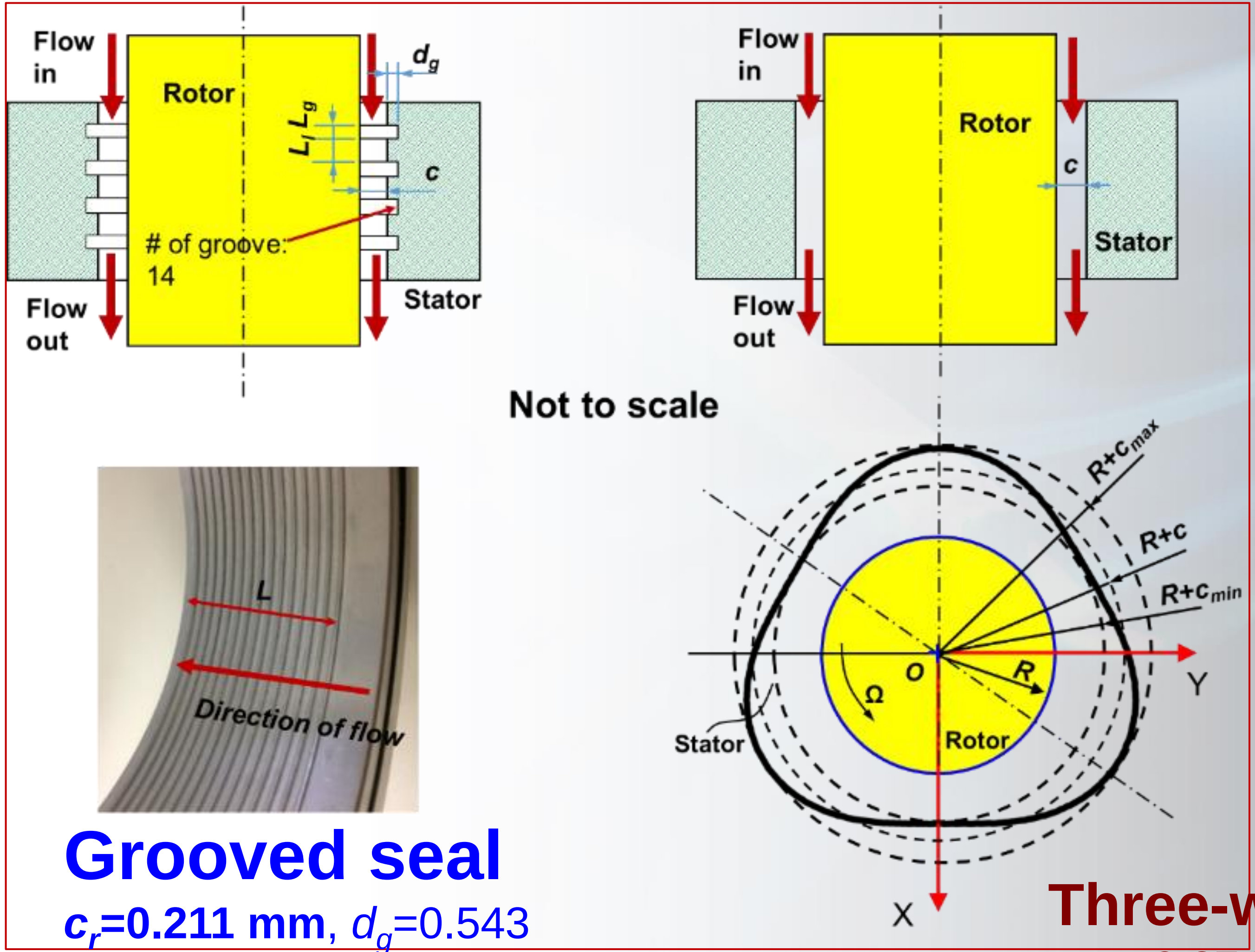


Gas content increases

ASME 2019



Two test seals



Grooved seal

c_r=0.211 mm, d_g=0.543 mm, l_g=1.5 mm, l_l=0.904 mm, N_g=14

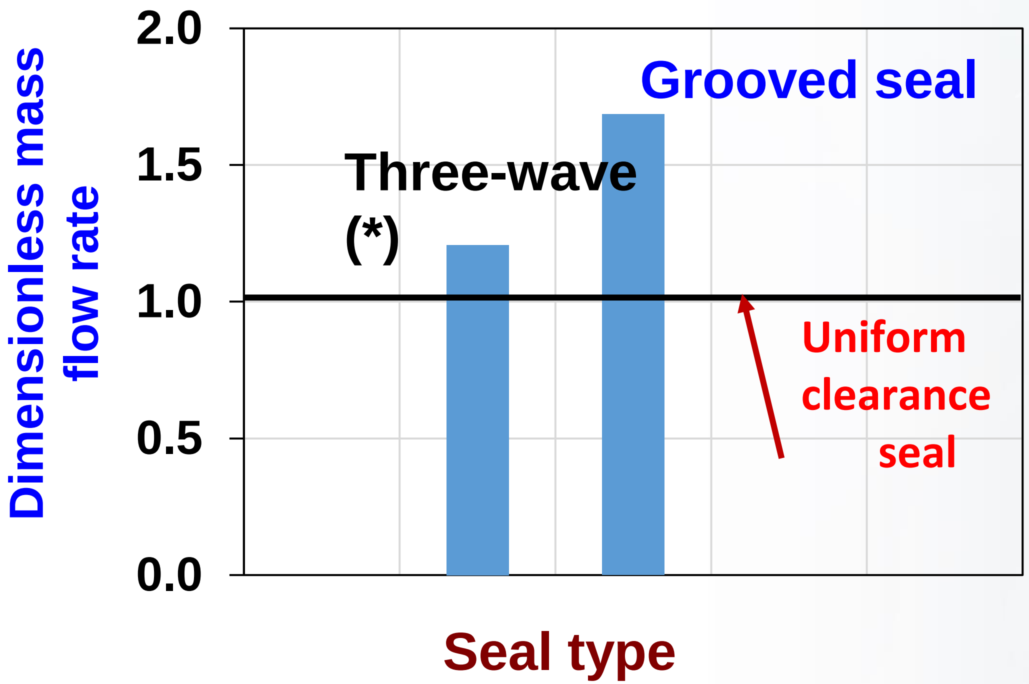
Three-wave seal

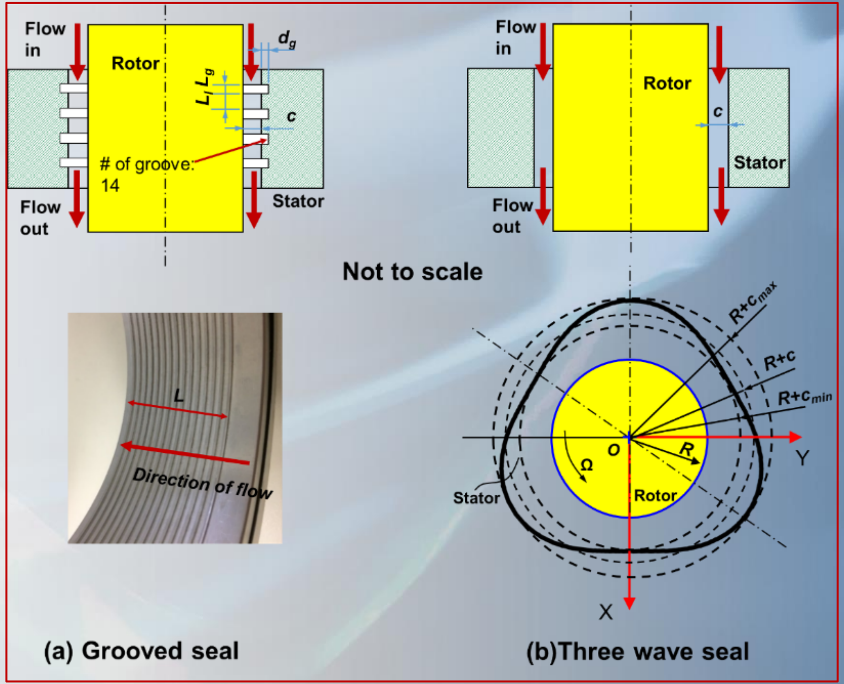
c_{max}=0.274 mm, c_{min} = 0.108 mm, **c_{av}=0.191 mm**

Measured Leakage: Liquid Only LVF=1

Normalized to

$$\dot{m}_{pl} = \frac{1}{12}\frac{\rho_l}{\mu_l}\pi D c^3 \frac{\Delta P}{L}$$





(a) Grooved seal (b)Three wave seal

Grooved seal not as effective as three wave seal (*) since flow is laminar.

(*) Lu, X., and San Andrés, L., **2019**," ASME J. Eng. Gas Turb. Power, **141**.

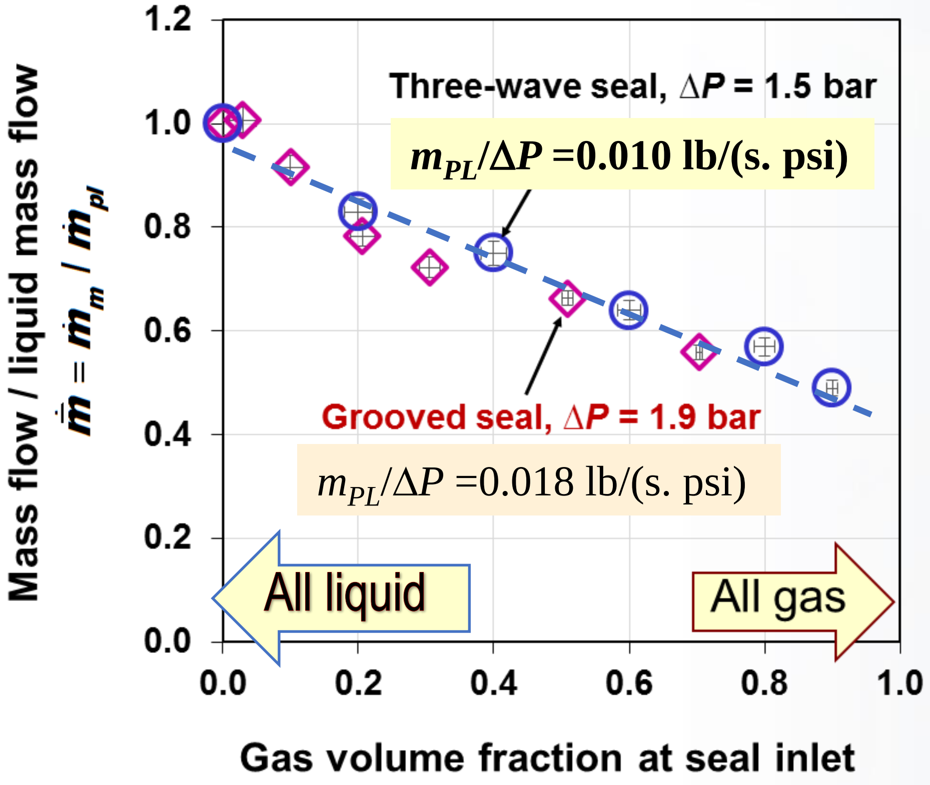
Leakage vs. Gas Content

Grooved seal vs Three-wave seal

$$\dot{\bar{m}} = \frac{\dot{m}_{mixture}}{\dot{m}_{liquid}}$$

Normalized with respect to liquid (GFV=0)

Rotor speed, Ω_{max}	3.5 krpm
Surface speed	23.3 m/s



Leakage for two seals drops as GVF increases.

For LVF=1 (GVF=0), grooved seal leaks (117 g/s) more than three wave seal (65 g/s).

Inlet GVF	0	0.10	0.5	0.7
GMF	0	0.05%	0.43%	1%

Grooved Seal Drag Torque vs. GVF

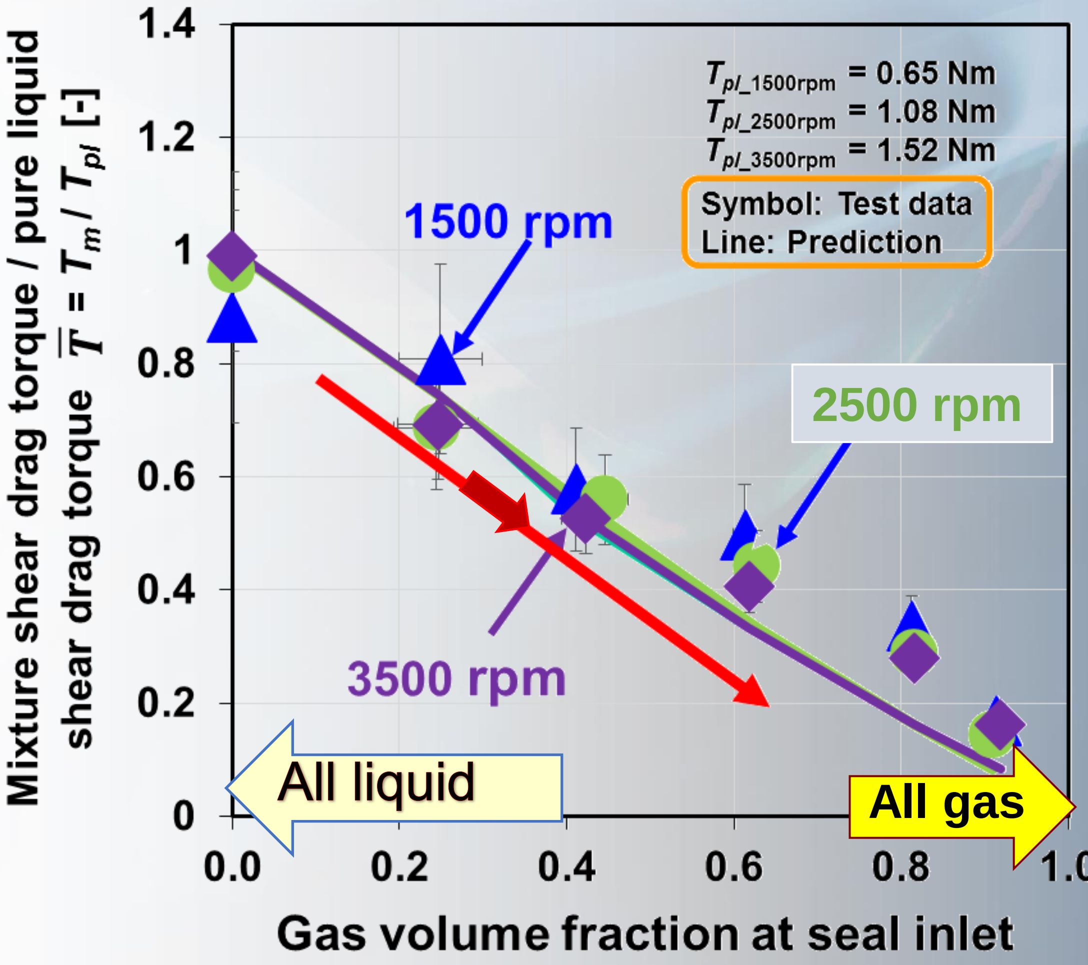
normalized to all liquid condition

prediction $T_{seal} \sim \dfrac{2\pi\ \mu_{(\mathrm{GVF})}\ \Omega R^3 L}{c}$

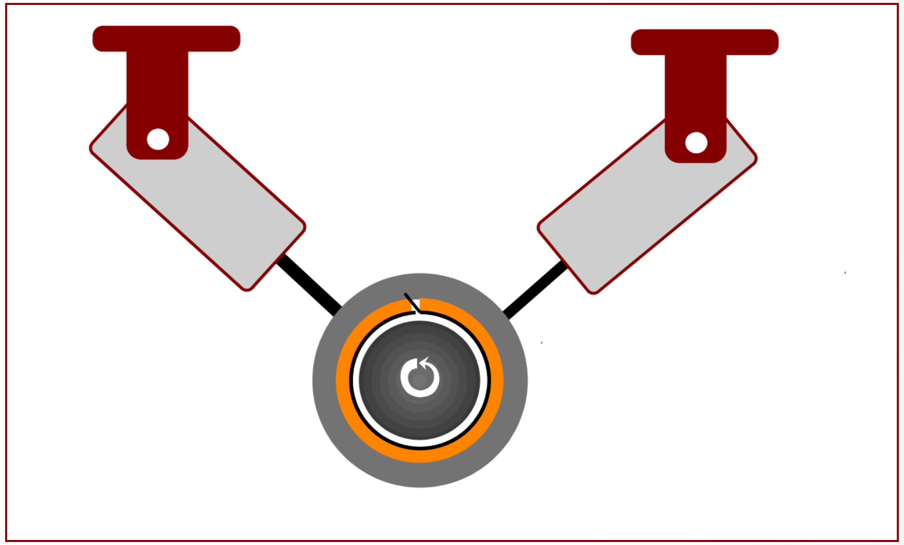
Torque linearly decreases with GVF.

GVF = 0 to 0.9 → 85% reduction in drag torque.

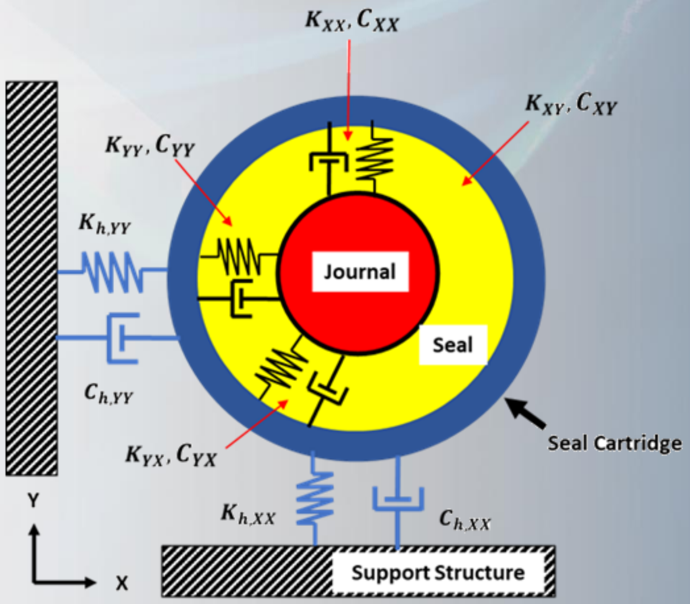
Torque change → Sudden overspeed will affect the pump and motor reliability!



Experimental Identification of Seal Rotordynamic Force Coefficients



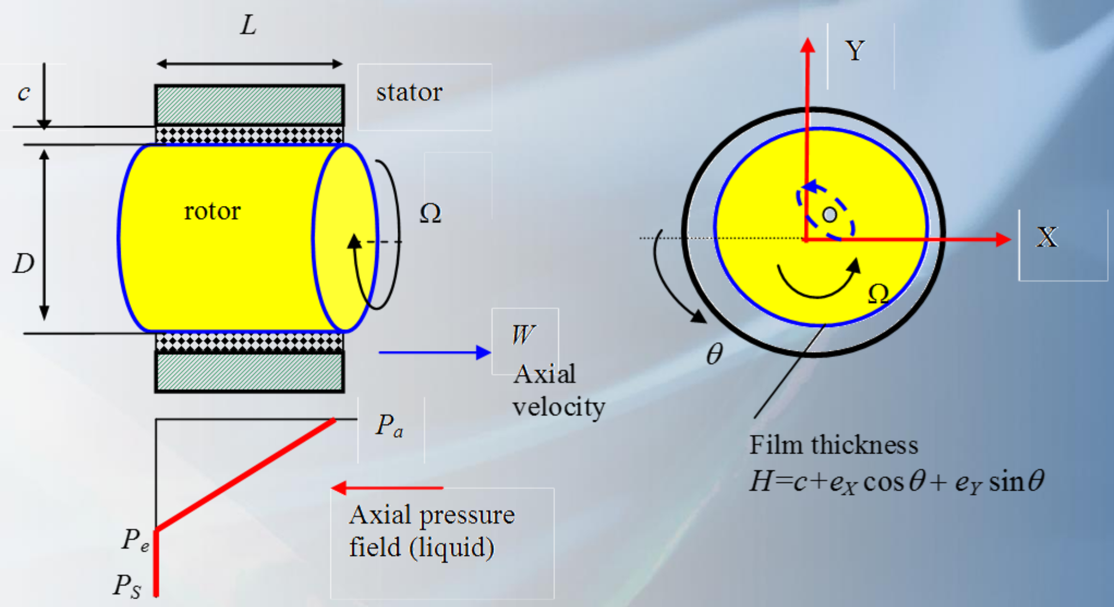
2-DOF system for seal and support structure



Dynamic Force Coefficients

For small amplitudes of rotor motion, a seal force is represented with **stiffness (*K*), damping (*C*)** and **added mass (*M*)** force coefficients.

However, wet seals have frequency dependent *K* & *C*:



$$\begin{Bmatrix} F_X \\ F_Y \end{Bmatrix} = -\begin{bmatrix} K_{(\omega)} & k_{(\omega)} \\ -k_{(\omega)} & K_{(\omega)} \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} - \begin{bmatrix} C_{(\omega)} & c_{(\omega)} \\ -c_{(\omega)} & C_{(\omega)} \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix}$$

For two-phase flow or a gas

Identification of Force Coefficients

1) Apply Load $\mathbf{F=F_o \sin(\omega t)}$ →

Measure vectors of displacements $\mathbf{z} = \{x, y\}^T$, & accelerations $\mathbf{a} = \{a_x, a_y\}^T$

2) $\bar{\mathbf{F}}, \bar{\mathbf{A}}, \bar{\mathbf{Z}}$ = Discrete Fourier Transform of **F, a, z**

3) $\bar{\mathbf{F}} - \mathbf{M}_h \bar{\mathbf{A}} - [\mathbf{K}_h + i\omega \mathbf{C}_h]\bar{\mathbf{Z}}' \rightarrow \mathbf{H}_{(\omega)} \bar{\mathbf{Z}}$

$[M, K, C]_h$ = mass, stiffness, damping of support structure

Components of seal complex stiffness H

$\mathrm{Re}(\mathbf{H}_{(\omega)}) \rightarrow \mathbf{K}_{(\omega)}$ **Dynamic Stiffness**

$\mathrm{Im}(\mathbf{H}_{(\omega)}) \rightarrow \omega\, \mathbf{C}_{(\omega)}$ **Proportional to Damping**

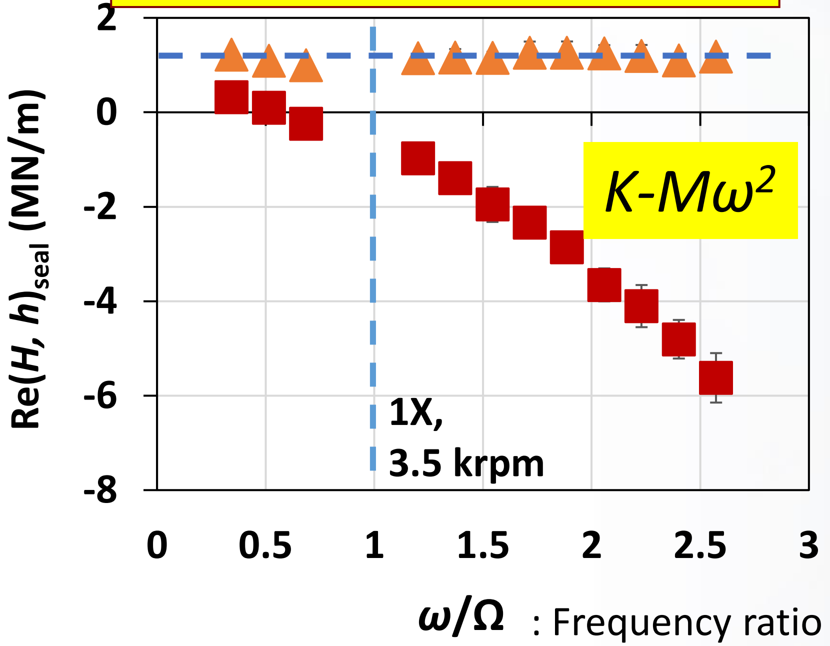
$C_{eff} = C - k/\omega = [\mathrm{Im}(H_{xx}) - \mathrm{Re}(H_{xy})]/\omega$ **Effective Damping**

Grooved Seal **H**

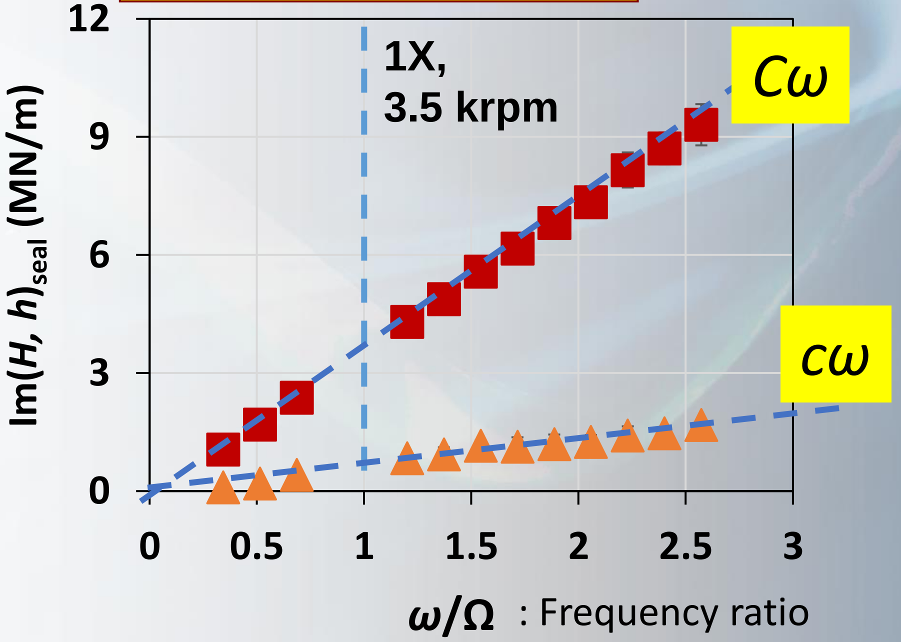
Unidirectional single frequency load,
motion amplitude = 0.05

P_s/P_a=3.9 **LVF=1 : liquid only**

$$\mathrm{Re}\left(\mathbf{H}_{\mathbf{seal}}\right) \rightarrow \mathbf{K}_{\mathbf{seal}} - \omega^2 \mathbf{M}_{\mathbf{seal}}$$

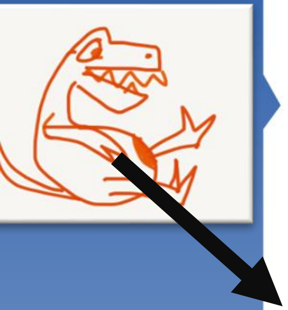


$$\mathrm{Im}\left(\mathbf{H}_{\mathbf{seal}}\right) \rightarrow \mathbf{C}_{\mathbf{seal}}\,\omega$$



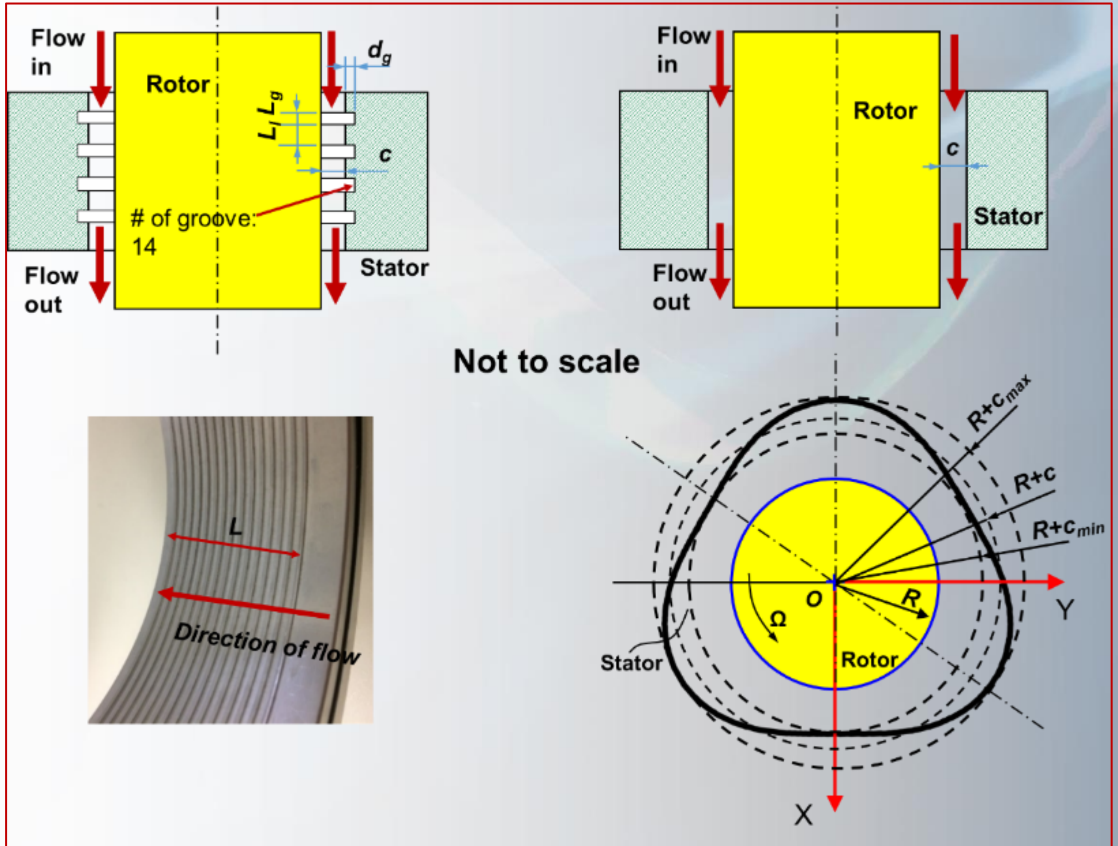
Curve Fit → force coefficients

***K*= 0.3 MN/m, *k*= 1.2 MN/m, *C*= 10 kNs/m, *c*= 1.8 kNs/m, *M*=6.6 kg, *m*=0 kg**



Compare force coefficients:

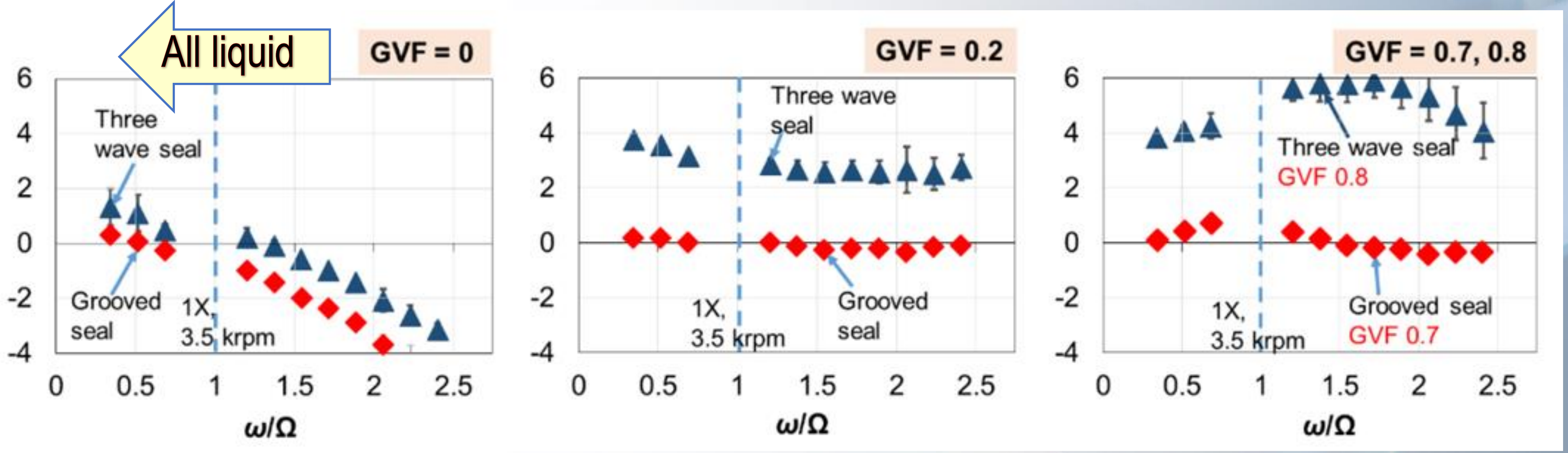
grooved seal vs. wave seal



(a) Grooved seal

(b)Three wave seal

Direct stiffness K (MN/m)



GVF = 0 (liquid): Both seals : K reduces with frequency → large added mass.

GVF 0.7 & 0.8:

Grooved seal shows small change in $K \sim 0$ for all frequencies.

Three-wave seal produces significant stiffening ($K >> 0$)

$K > 0$ desirable for system static stability and to increase its natural frequency. → Most useful in vertical pumps.

3.5 kpm & P_s/P_a = GroovedS=2.9, WaveS= 2.5

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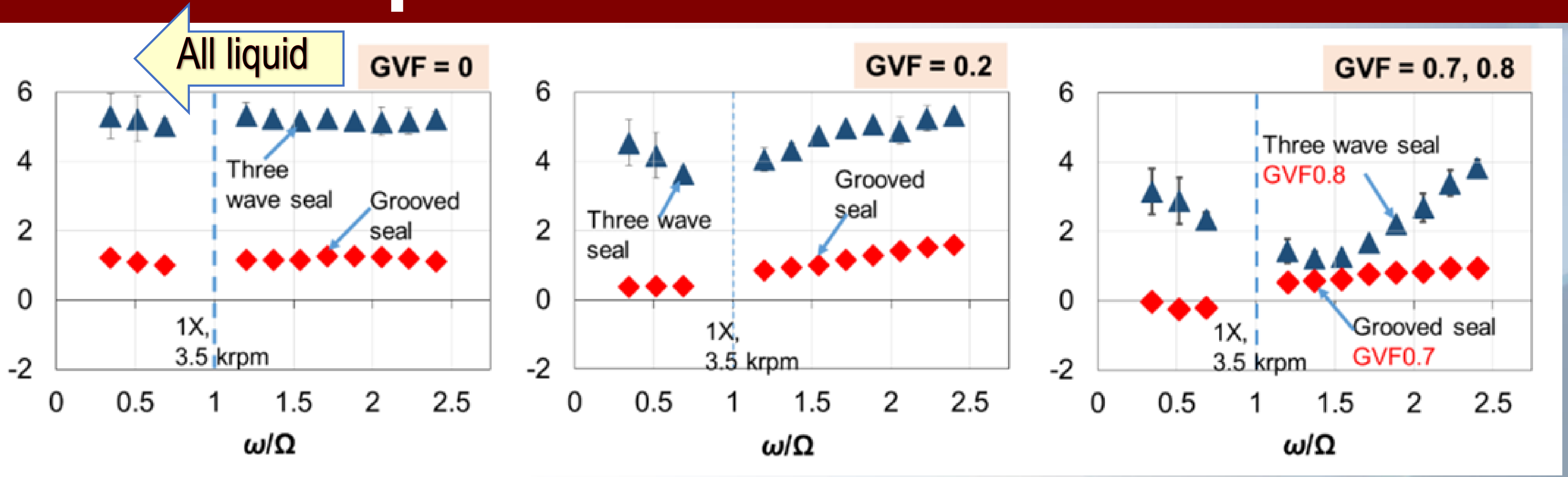
Cross-coupled stiffness *k* (MN/m)



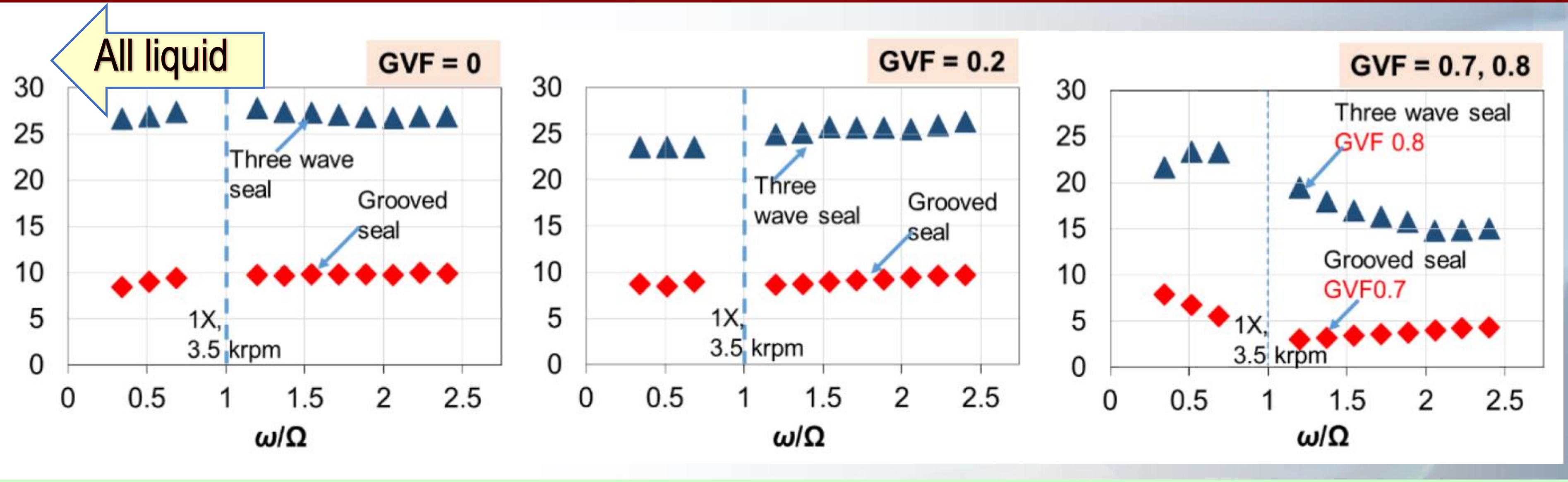
GVF = 0 (liquid) & 0.2: Wave seal shows larger cross-stiffness → affects stability.

GVF > 0.7:
Grooved seal shows small changes in k ~ 0 for all frequencies.
Three-wave seal has *k lowest* for synchronous frequency.

***k* → 0 desirable for system dynamic stability.**

3.5 kpm & P_s/P_a = GroovedS=2.9, WaveS= 2.5

Direct damping *C* (kNs/m)



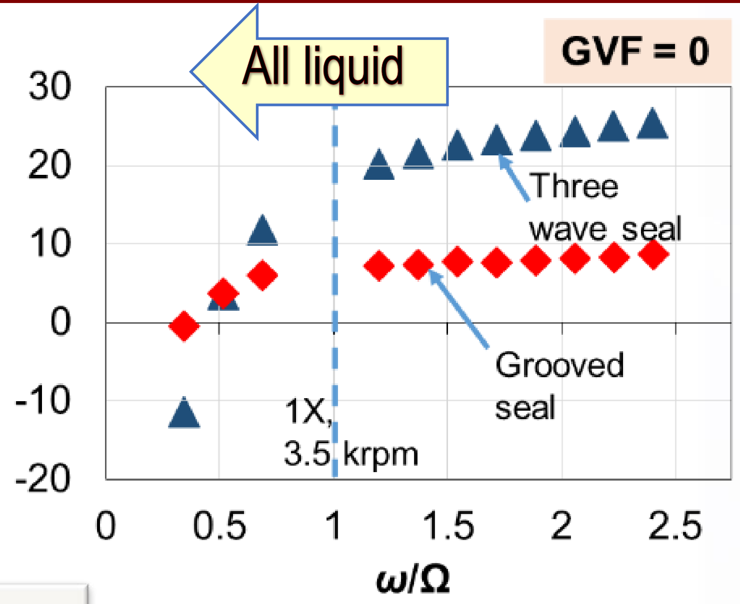
GVF = 0 (liquid) and 0.2:
Wave seal has ~ 2.5 more damping & not affected by frequency.

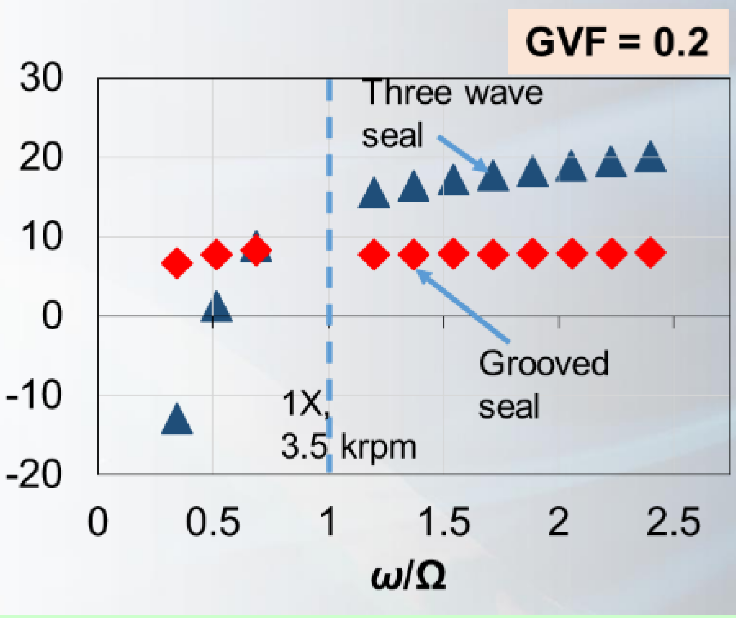
GVF > 0.7, 0.8
Wave seal still shows large *C*, albeit decreasing with frequency above shaft speed.

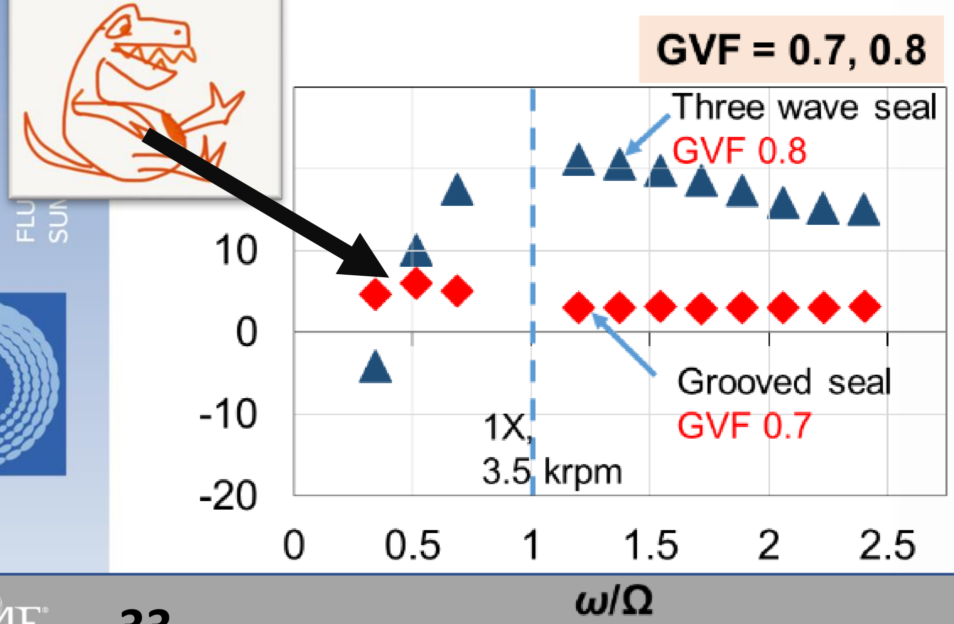
***C* > 0 desirable for system dynamic stability.**

3.5 kpm & P_s/P_a : GroovedS=2.9, WaveS= 2.5

Effective damping $C_{eff} = (C - k/\omega)$ (kN.s/m)



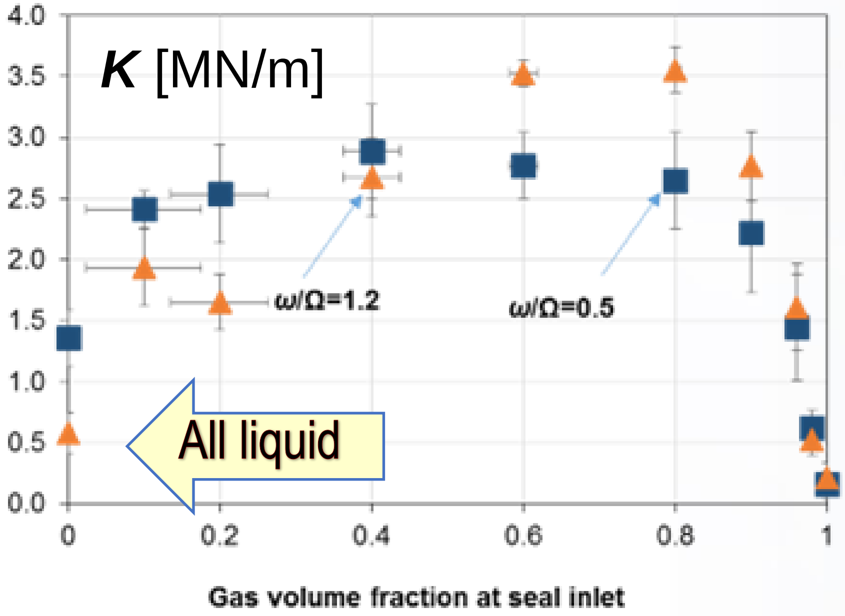


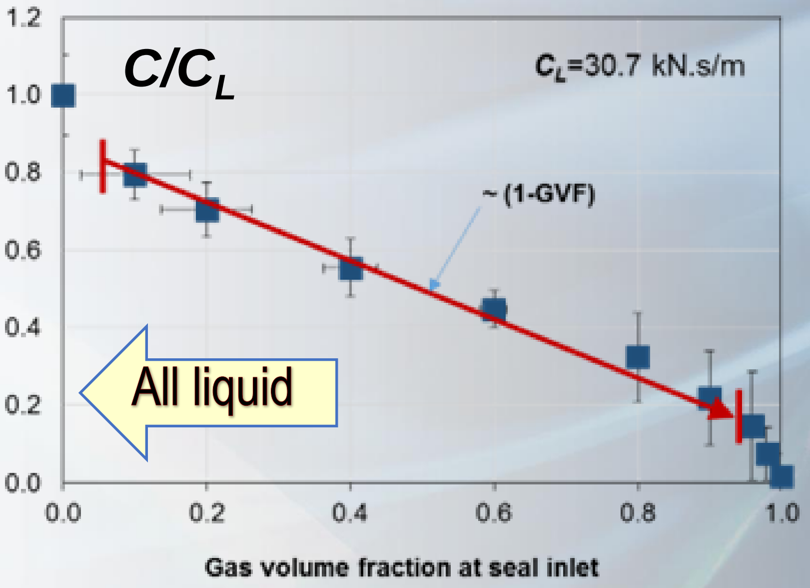


Wave seal shows much larger effective damping for most frequencies above break frequency.

Whirl frequency ratio WFR=
Three wave seal: 0.47
Grooved seal: 0.33

Three-wave seal stiffness and damping





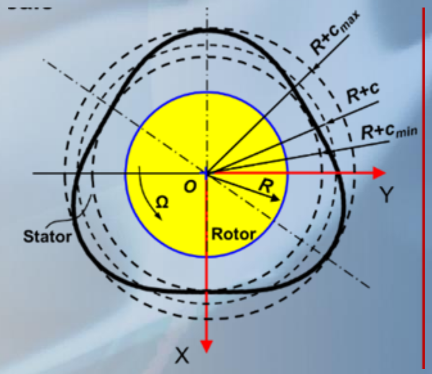
Wave seal quickly stiffens *(K >> 0)* when supplied with a small amount of gas.

GVF	0	0.10	0.5	0.7
GMF	0	0.05%	0.43%	1.00%

Damping decreases in proportion to GVF=1-LVF

3.5 kpm & P_s/P_a = 2.0

Conclusion Grooved vs Three Wave Seals



(a) Grooved seal leaks more compared to other seals ← laminar flow. NOT a sound choice for a seal!

(b) Leakage drops continuously with an increase in gas volume fraction (GVF).

(c) Force coefficients are frequency dependent ← gas/oil mixture

(d) For pure liquid (GVF=0), seals produce a small centering stiffness (*K*) and a large added mass (*M*).

(e) Wave seal produces large direct stiffness *K* (>0) even for small GVFs (> 0.05).

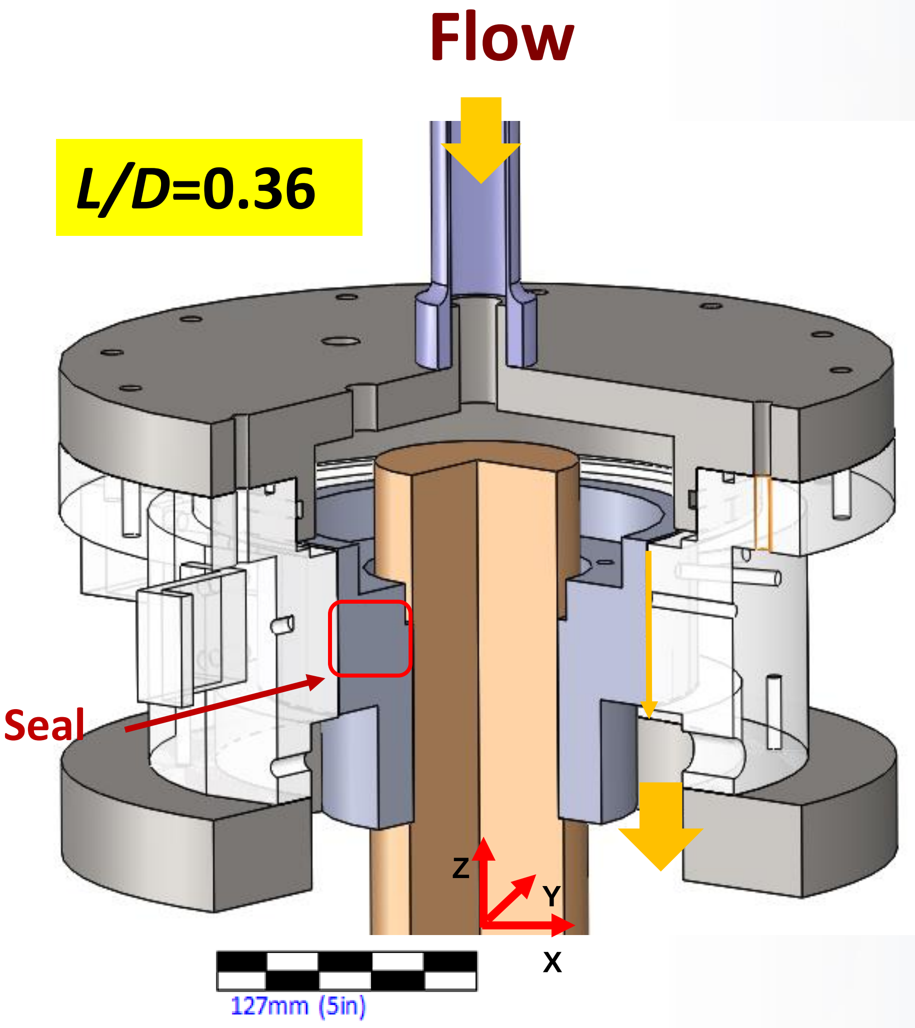
(f) Wave seal offers 2.5 to 3 x more effective damping than grooved seal!

Unexpected phenomenon

Observed (and quantified) low frequency seal cartridge motions exacerbated by GVF.

ASME GT2017-63254

Seal Geometry and Fluid Properties



Rotor and seal

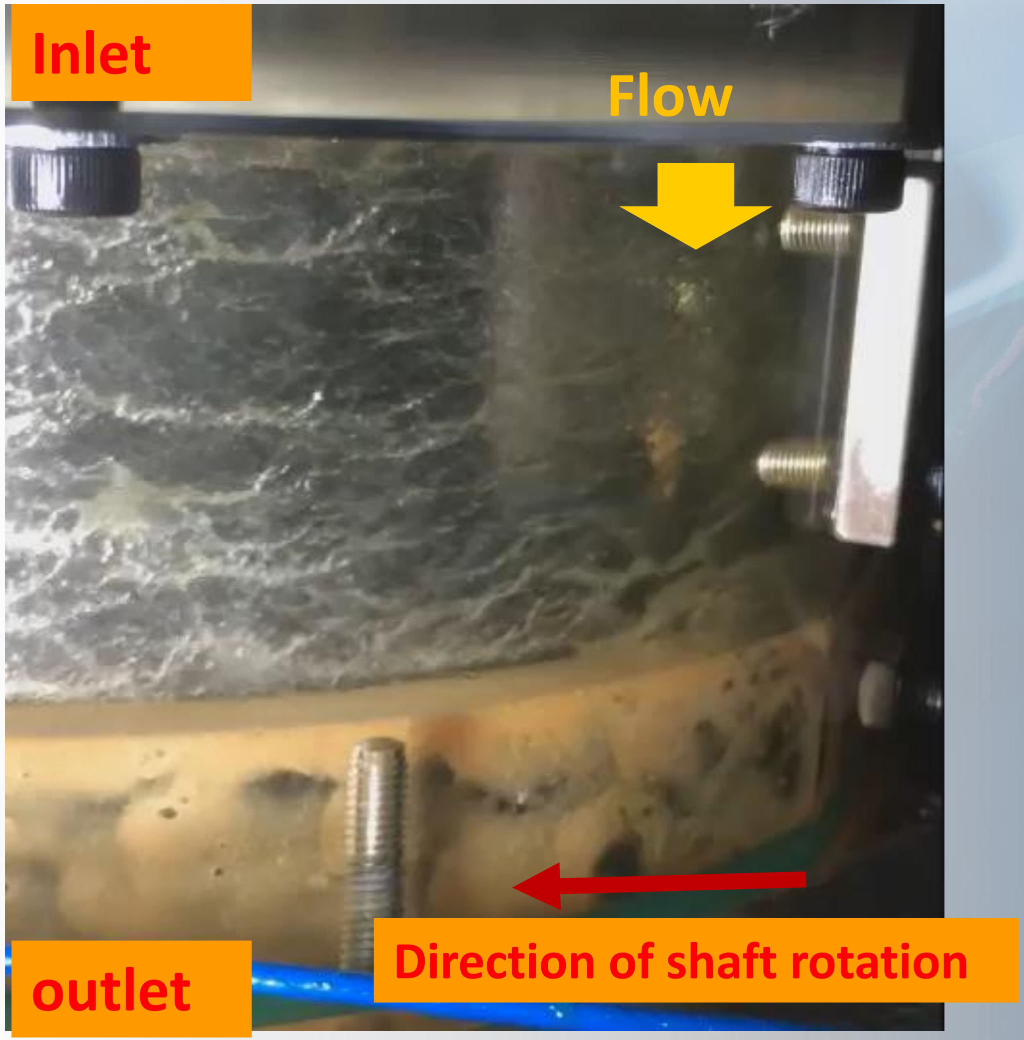
UNIFORM CLEARANCE Seal	
Diameter (*D*)	**127 mm** (5 in)
Length (*L*)	**46 mm** (1.8 in)
Clearance (*c*)	**0.203 mm** (8 mil) at 34 ºC
Supply pressure (P_s)	1.0~3.5 bar (abs)
Oil ISO VG 10 density(ρ_l)	**830 kg/m³**
viscosity (μ_l)	**10.6 cP** at 34 ºC
Air density (ρ_{ga})	**1.2 kg/m³** at 1bar
viscosity(μ_{ga})	**0.02 cP** at 20 ºC, 1 bar (abs)
Shaft speed (Ω_{max})	**3.5 krpm**
Rotor surface speed ½ $D\Omega_{max}$	**23 m/s**

Visualization of Mixture

$P_s/P_a = 2$, speed 1.8 krpm

Stroboscope light with frequency 30 Hz freezes shaft motion

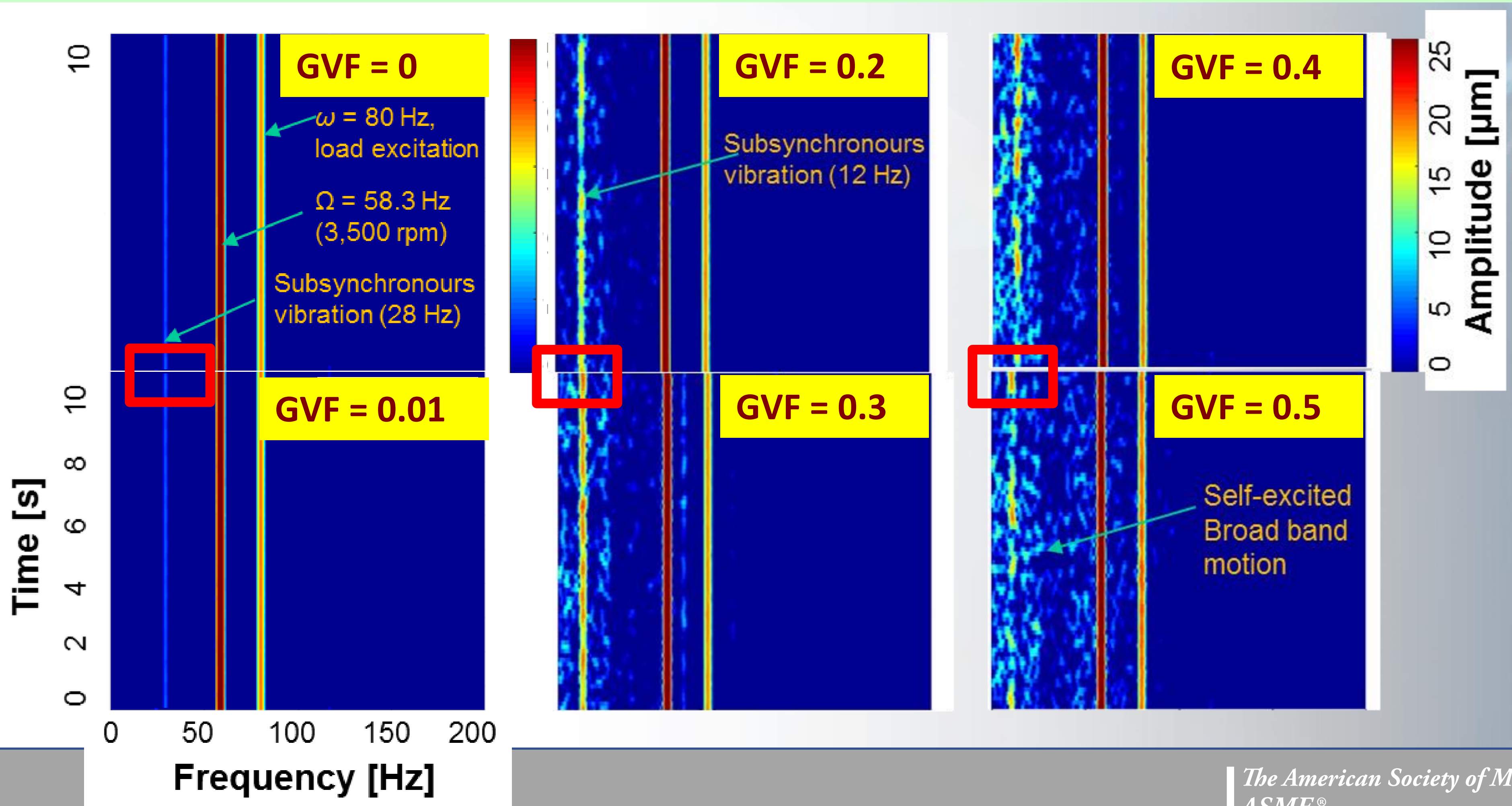
Air bubbles coalesce and liquid merges to make streamlets



Low Frequency Self-Excited Vibrations

Small amplitude SSV at ½X for GVF = 0 and 0.01

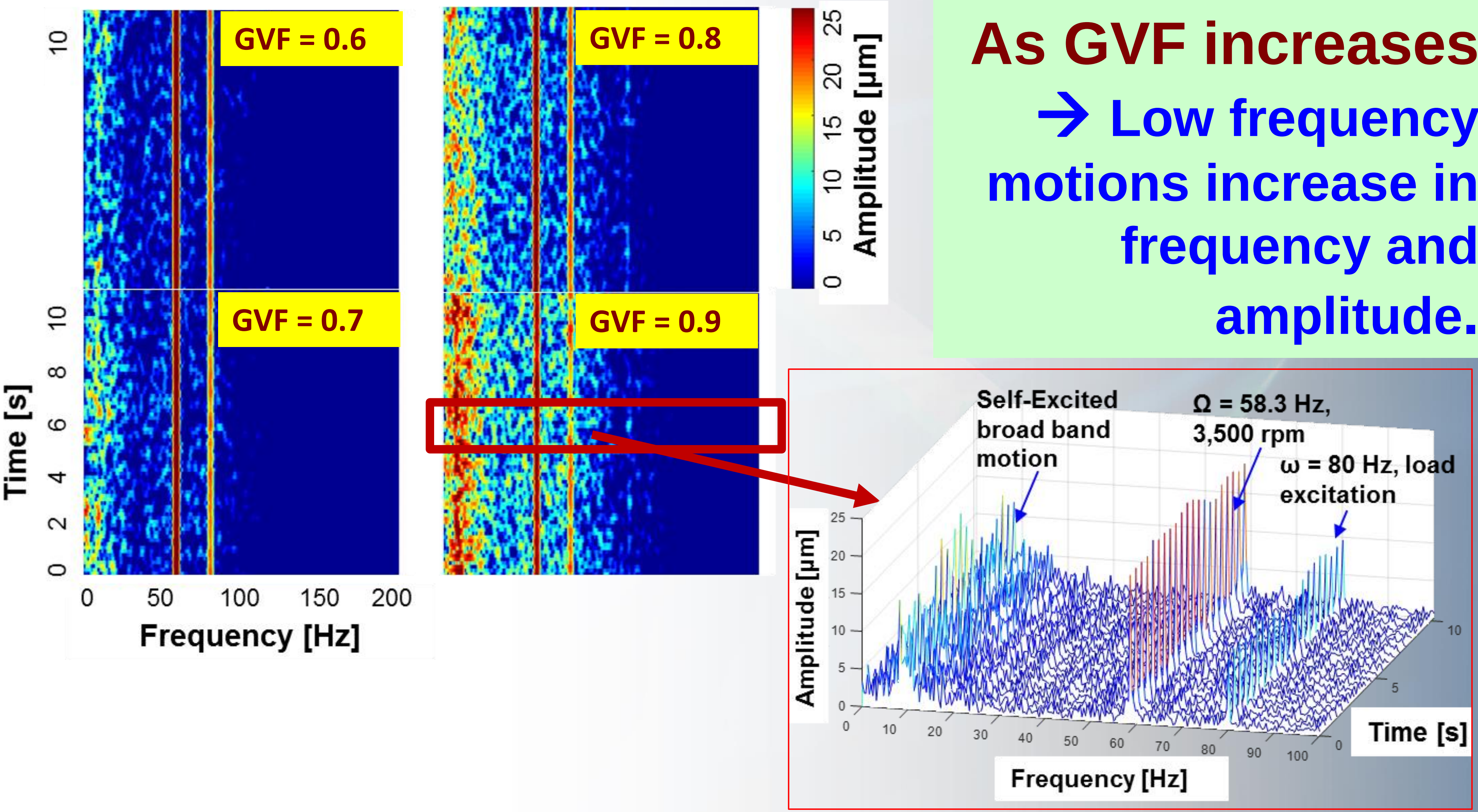
Low frequency at ~12 Hz for GVF > 0.1. Amplitude → increases



Low Frequency Self-Excited Vibrations

As GVF increases

→ Low frequency motions increase in frequency and amplitude.



Low frequency self-excited motions due to acoustic resonance from sound speed in the mixture.

Sound speed in mixture:

$$V_s = \frac{1}{\sqrt{\rho_m\left(\frac{\alpha}{\rho_g V_{sg}^2}+\frac{1-\alpha}{\rho_l V_{sl}^2}\right)}}$$

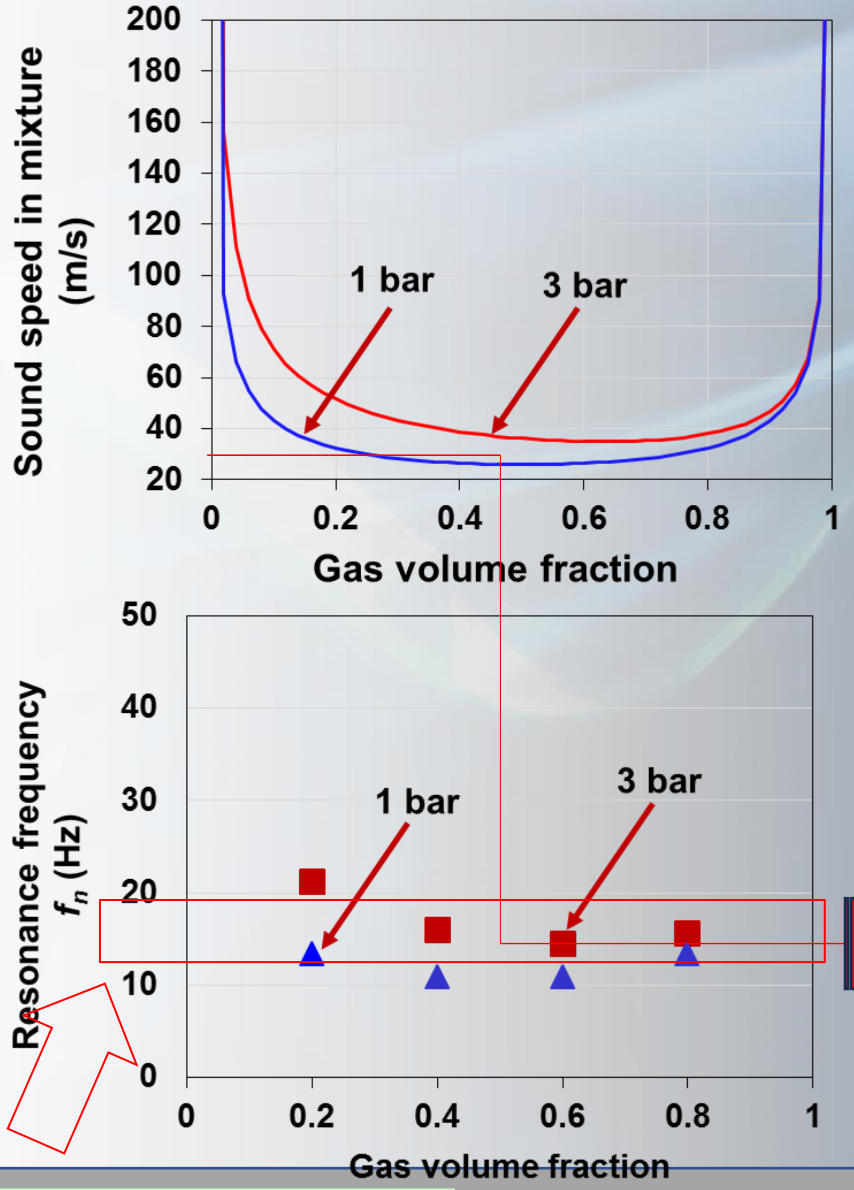
V_{sg}: sound speed in air, **351 m/s** at 34 °C & 1 bar.
V_{sl} : sound speed in oil, **1,432 m/s.**

Acoustic resonance in a circular duct with one end open (*):

$$f_n = \frac{nV_s}{8\pi\cdot(L+\zeta D)}\ \text{(Hz)}$$

n = 1, 2, 3…

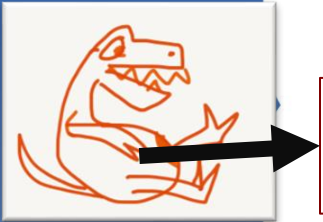
n = 1 for fundamental frequency, ζ =0.4 correction factor for tube with one end open.



Mixture low sound speed produces resonance → tests show it!

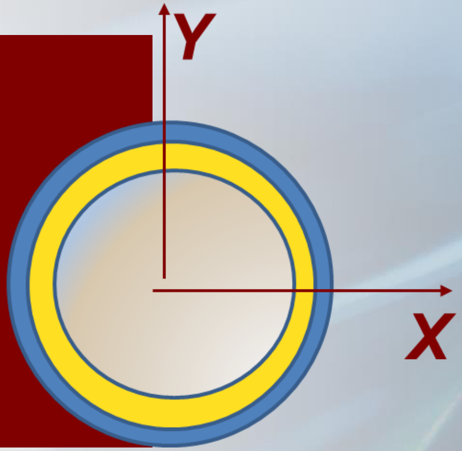
12.3 Hz

(*) Kuttruff, H., 2007, Acoustics: An Introduction, Taylor & Francis, NY, Chap. 9.



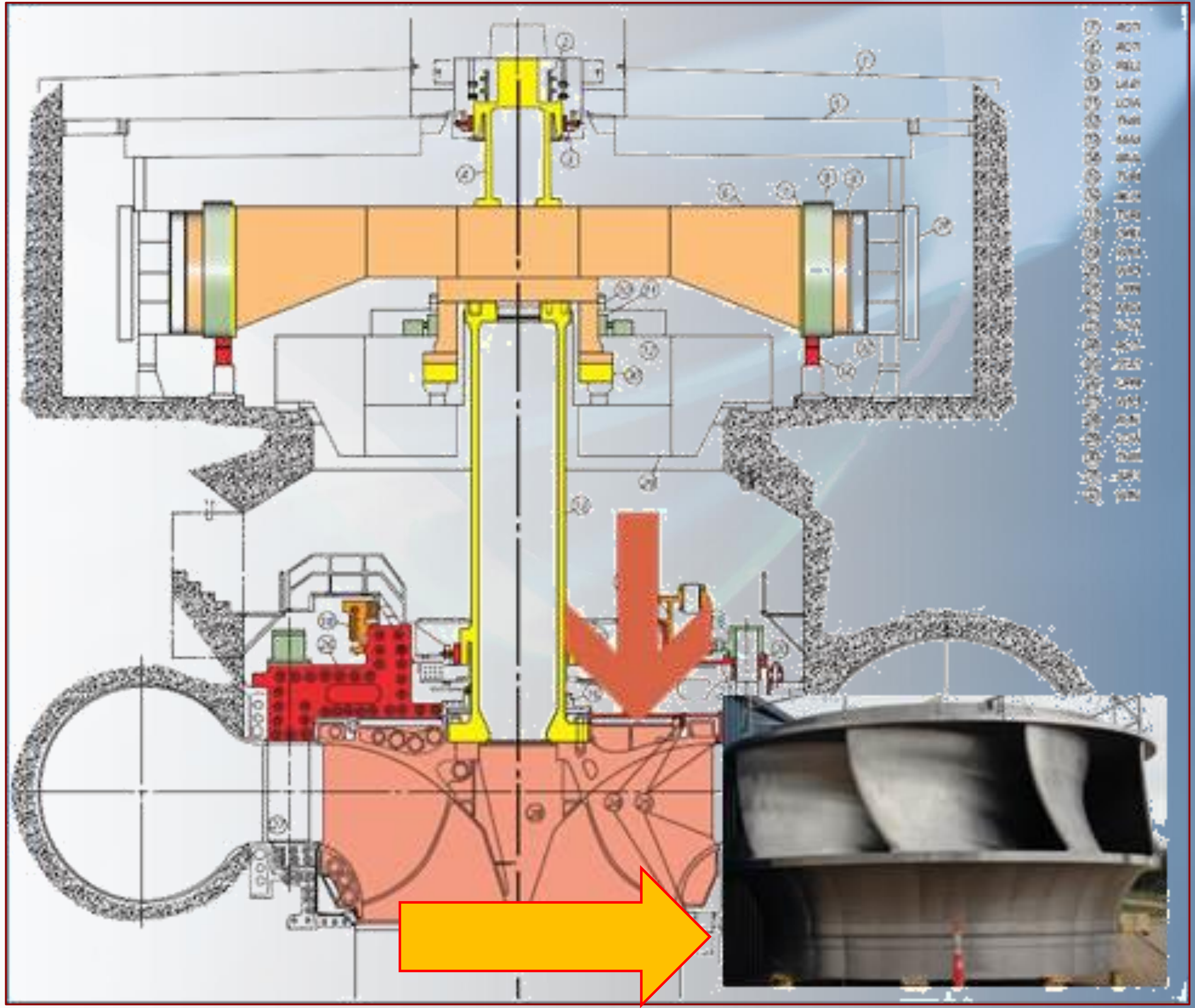
Useful for the practicing engineer

Air injection to increase stiffness *K*



Often large vertical turbines/pumps show SSV (→ a resonance)

A common practice is to inject air into the band seal to reduce rotor motion amplitudes.

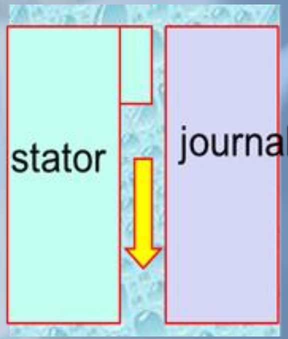


Air injection increases *K*

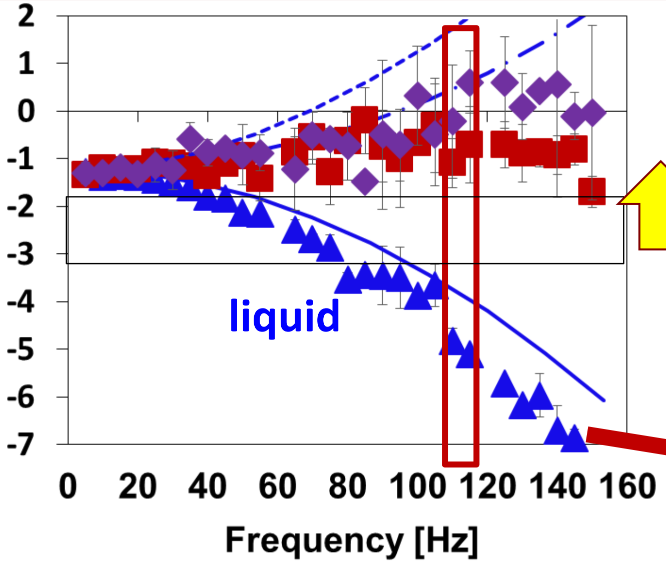
Step clearance seal

Air injection turns a negative stiffness into a positive one (*K* > 0 is centering).

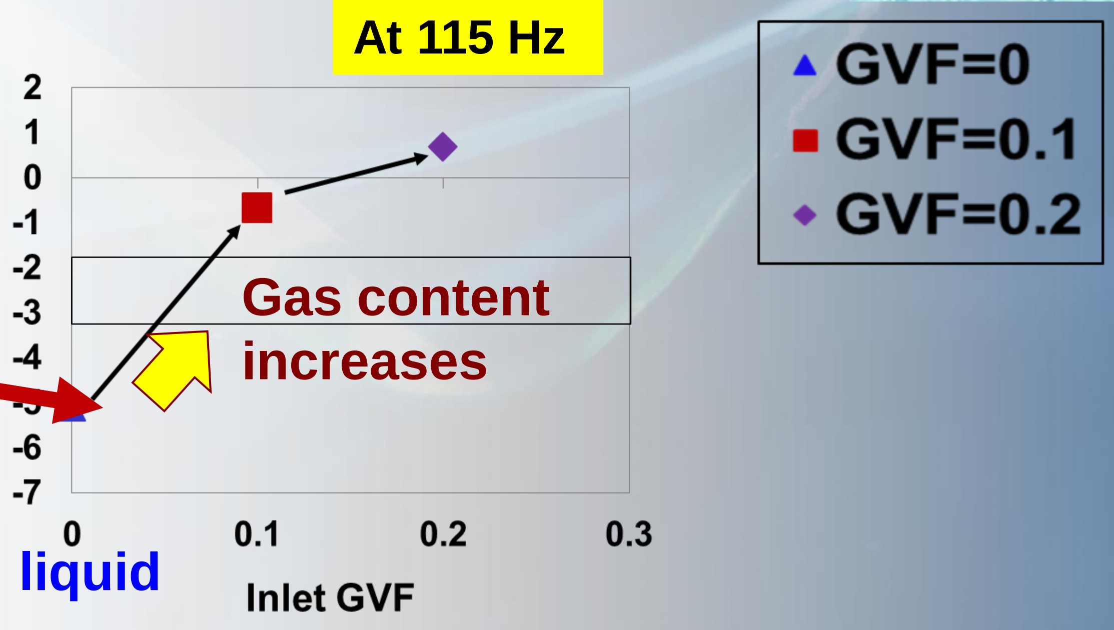
0 rpm



Q_l = 11.4 L/min, P_s=2.9 bara



Gas content increases



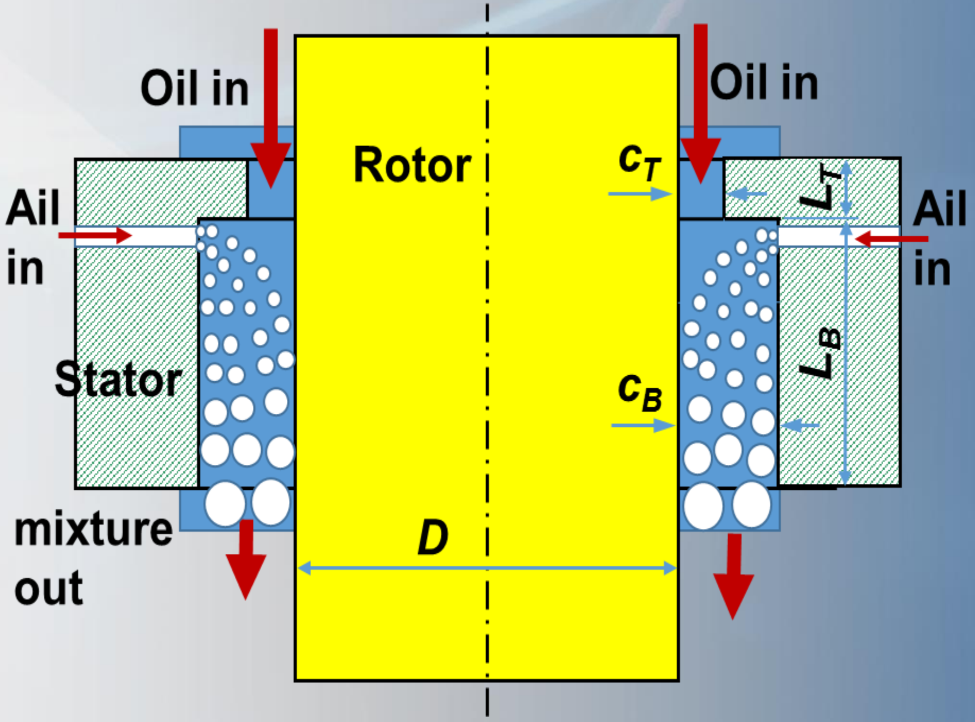
***K* > 0 brings static stability to vertical turbines.**

Seal stiffness hardens due to quick drop in sound speed brought in by the small amount of gas and exacerbated by excitation frequency.

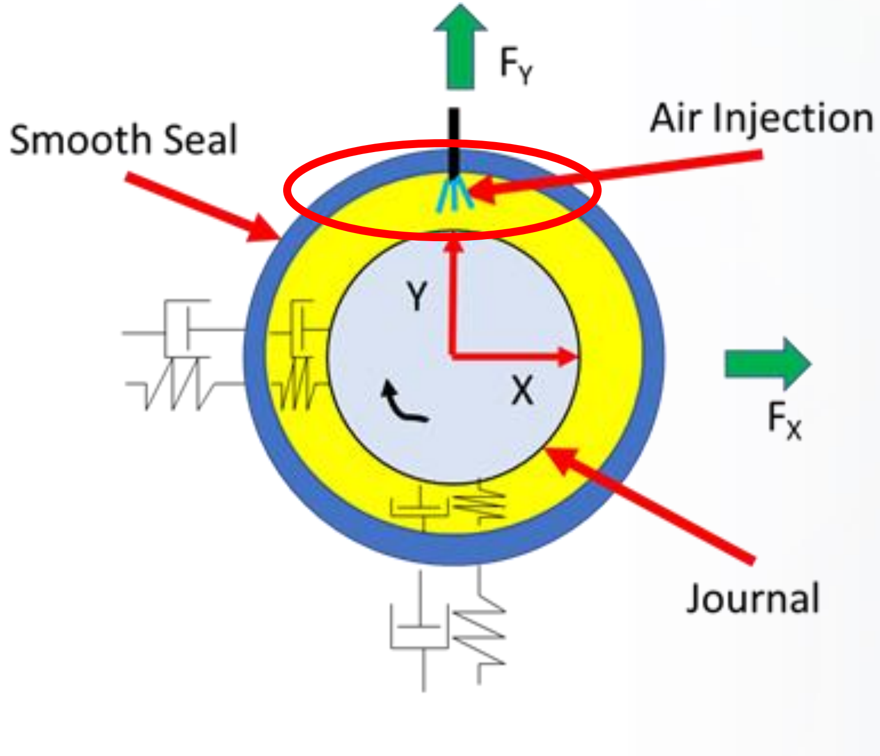
Idea!

LOCAL AIR INJECTION TO PROMOTE THE STATIC STABILITY OF VERTICAL PUMPS: TESTS & ANALYSIS

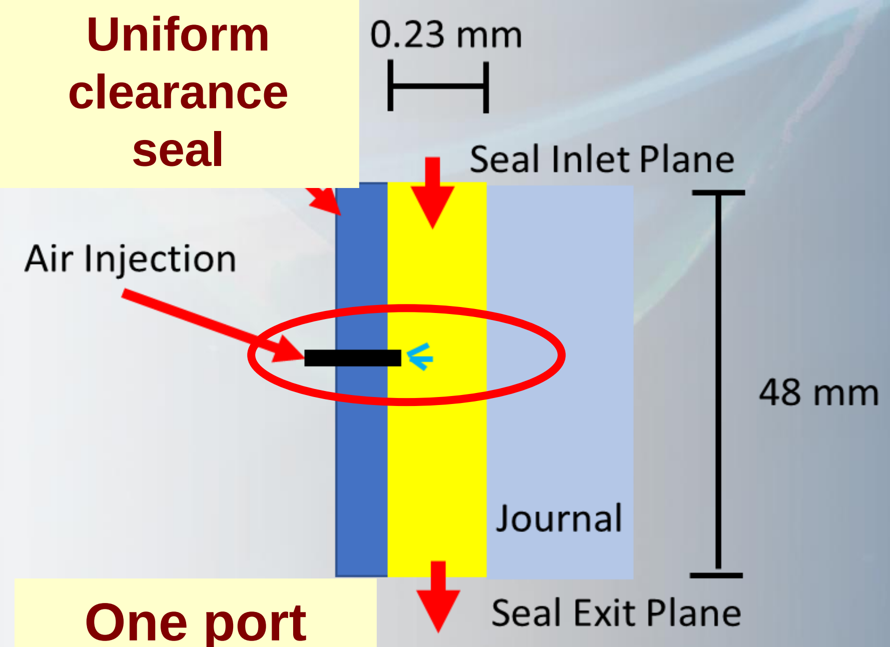
1. Design and manufacture a uniform clearance seal.
2. Measure leakage at various conditions of GVF, supply pressure, and shaft speed.
3. Conduct dynamic load tests and identify force coefficients vs. frequency and GVF.



Local injection of bubbles to increase seal stiffness



Top View



Uniform clearance seal

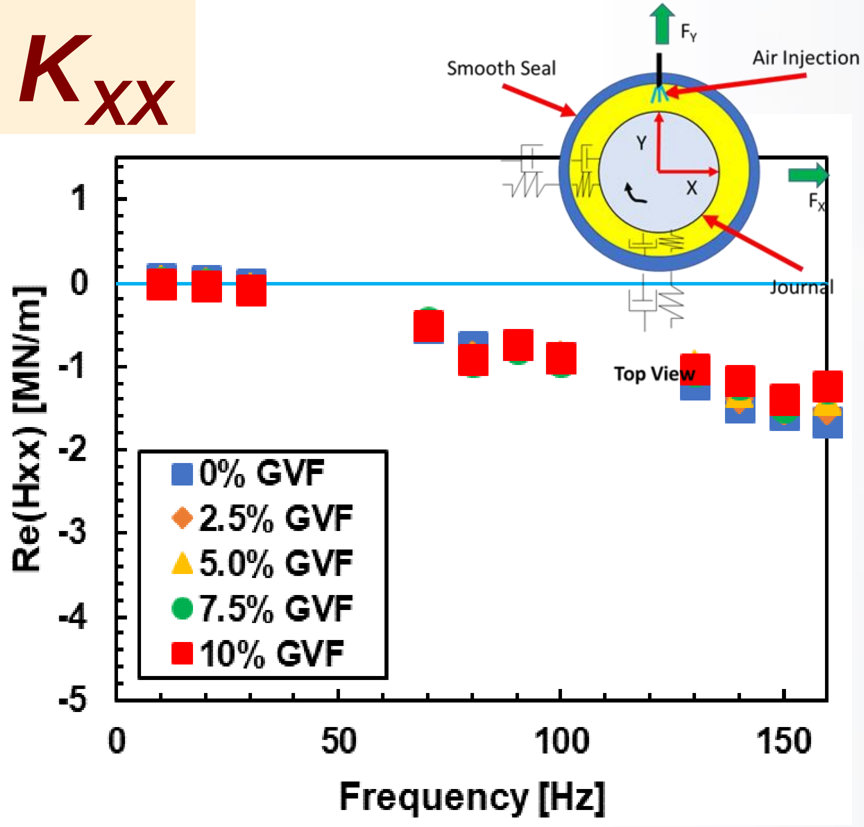
One port (feed orifice) along *Y*

Side View

Injection of bubbles increases *K*

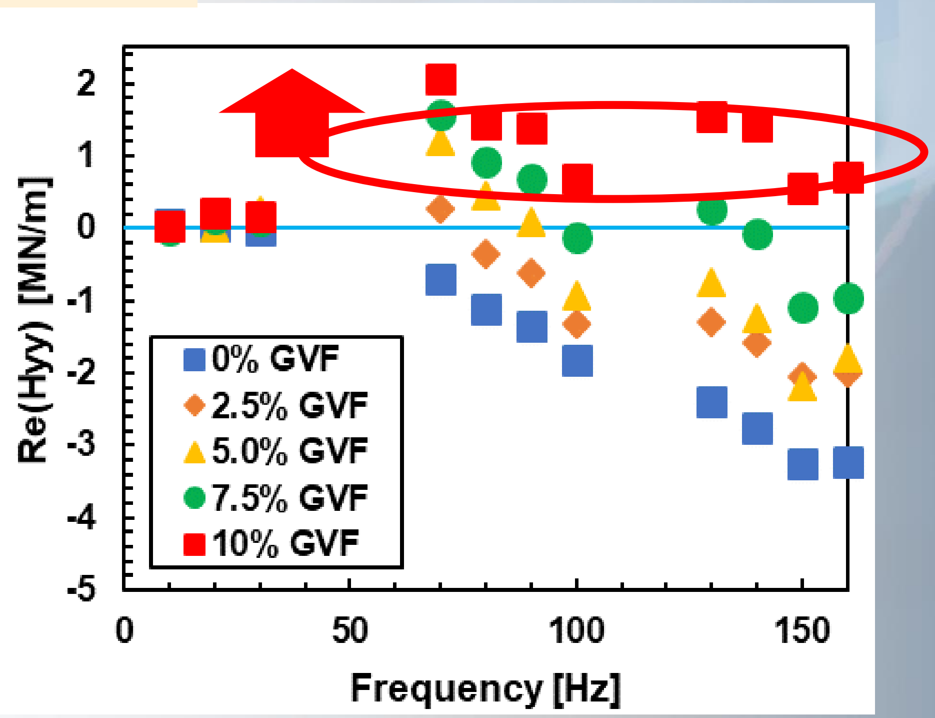
0 rpm, *Ps/Pa*=2.5

K_{XX}



K_{YY}

***Y*- direction of bubbles injection**



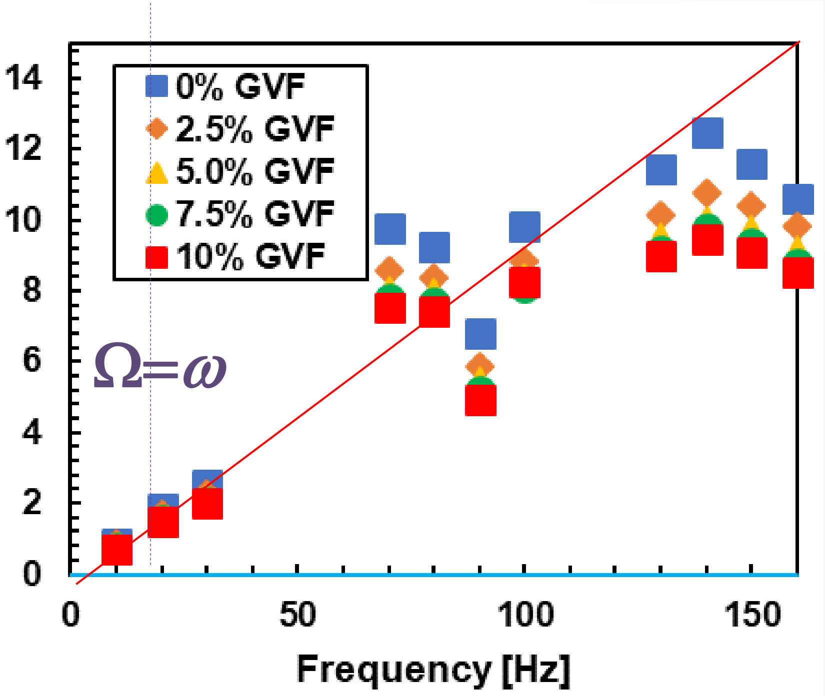
$K_{XX} < 0$, $K_{YY} > 0$ as GVF increases

Stiffness asymmetry promotes rotor stability!

Injection of bubbles reduces damping

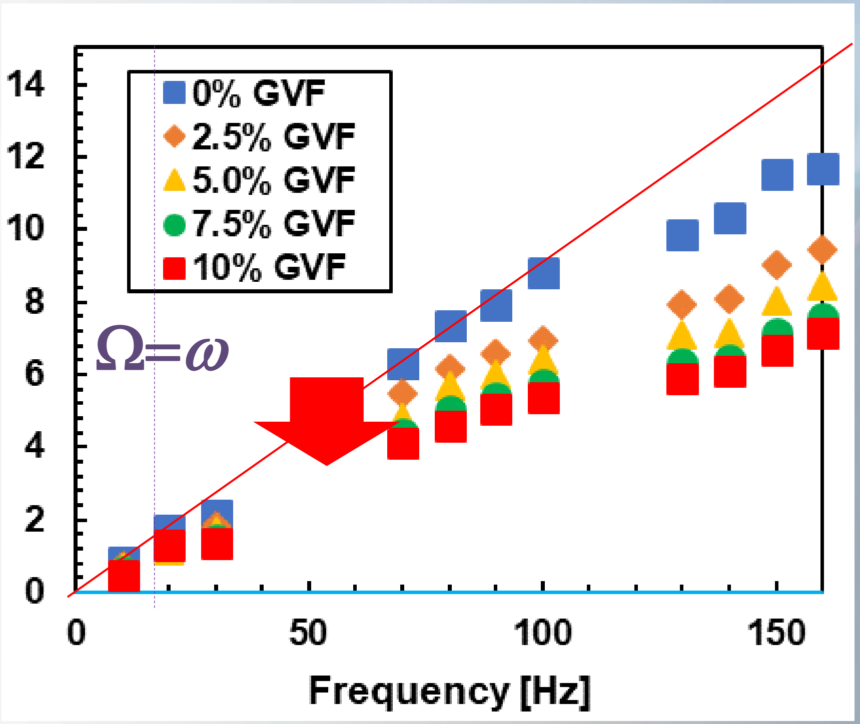
1 krpm, *Ps/Pa*=2.5

$\sim\omega C_{XX}$



$\sim\omega C_{YY}$

***Y*- direction of bubbles injection**

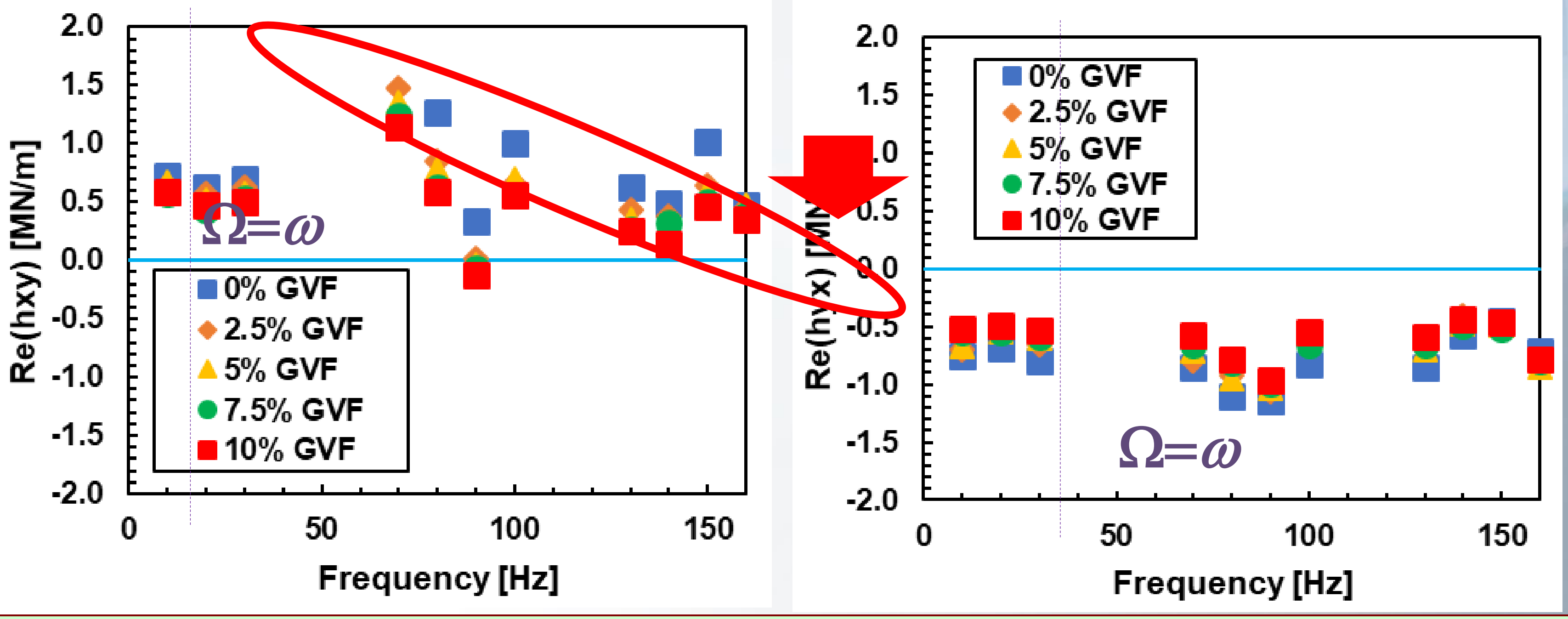


C_{XX} & C_{YY} > 0 decrease as GVF increases.

Speed Ω = 1 krpm (16.6 Hz)

Injection of bubbles reduces k_{XY}

1 krpm
Ps/Pa=2.5



k_{XY}

k_{YX}

Y- direction of bubbles injection

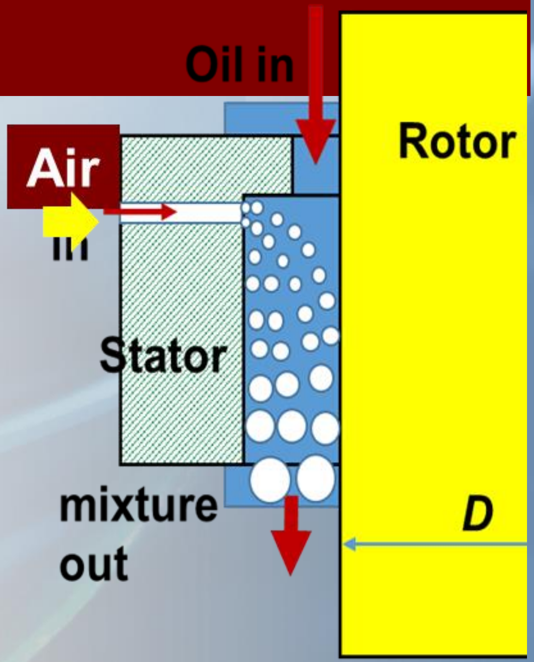
k_{XY} ~ - k_{YX} for $\omega < 2\,\Omega$. $|k_{XY} - k_{YX}|$ decreases as both GVF & frequency increase → + stable.

$C_{eff\_,XX}=(C_{XX} - k_{XY}/\omega)$ ~ C_{XX} $\qquad$ $C_{eff,\_YY}=(C_{YY} + k_{XY}/\omega)$

Speed Ω = 1 krpm (16.6 Hz)

Closure

A SIMPLE WAY TO CONTROL SEAL STIFFNESS IN VERTICAL PUMP CONFIGURATIONS



Local gas injection produces seal stiffness hardening & orthotropy → increases static stability (+ good for vertical systems). Even a small amount of gas injection helps!

The road ahead

To seal or not to seal….

As O&G demands increase, industry will continue to advance more efficient, reliable and cost-effective sealing solutions.

Questions?

Acknowledgments

The multiple contributions of the late Dr. Dara Childs and Dr. John Vance to the rotordynamics of turbomachinery are acknowledged with gratitude.

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Learn more: http://rotorlab.tamu.edu

53th Turbomachinery & Pump Symposium, Aug 20-22, 2024, Houston (https://turbolab.tamu.edu/proceedings)

“Wet (Bubbly Liquid) Seals for Multiphase Pumps: Leakage and Dynamic Force Coefficients of Two seals and a Simple Way to Control Seal Stiffness in Vertical Pump Configurations,” https://hdl.handle.net/1969.1/1582500

Questions to lsanandres@tamu.edu

and/or get pdf of this presentation

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