ON MULTIPLE PHASE PUMP SEALS: LEAKAGE AND GAS INJECTION TO CONTROL SEAL CENTERING STIFFNESS

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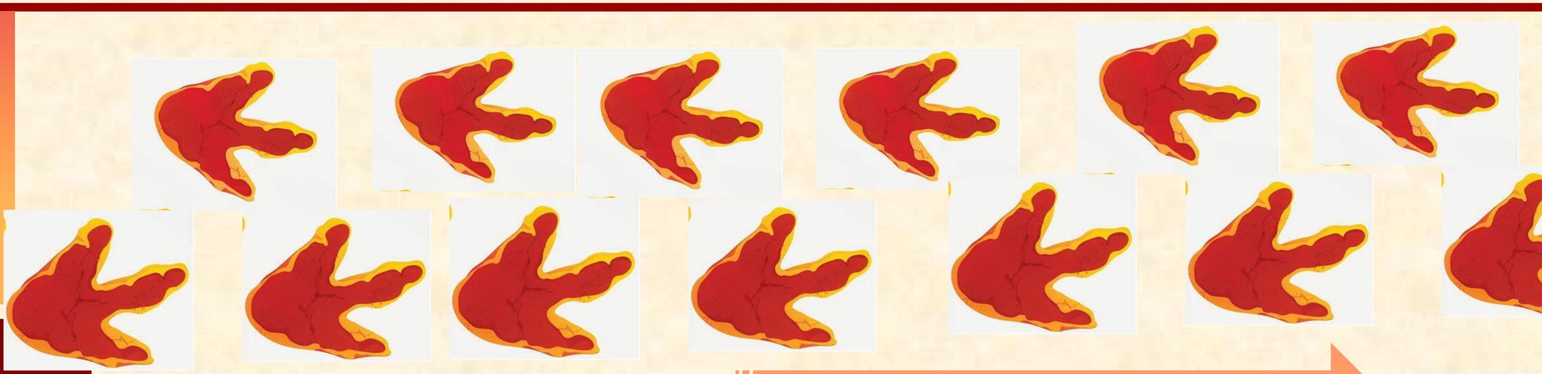
Luis San Andrés is the Mast-Childs Chair Professor at Texas A&M University. Since 1990, Luis performed research in lubrication and rotordynamics and produced advanced technologies of hydrostatic bearings for cryogenic turbo pumps, squeeze film dampers for aircraft jet engines, and gas foil bearings for oil-free micro turbomachinery..

Dr. San Andrés received the ASME-IGTI 2022 Aircraft Engine Technology Award for sustained personal creative contributions to aircraft engine technology.

Dr. Xueliang Lu is a Vice Chief Engineer at Hunan Sund Technological Corporation (Xiangtan) developing advanced rotor-bearing systems and sliding bearings for wind turbine gear boxes and main shaft bearings. Xueliang received B.S. and M.S. degrees in ME from Xiangtan University in China, and a PhD in ME at Texas A&M University – Turbomachinery Laboratory. After graduation, Xueliang worked for Atlas Co.

A dinosaur walk since last millennium



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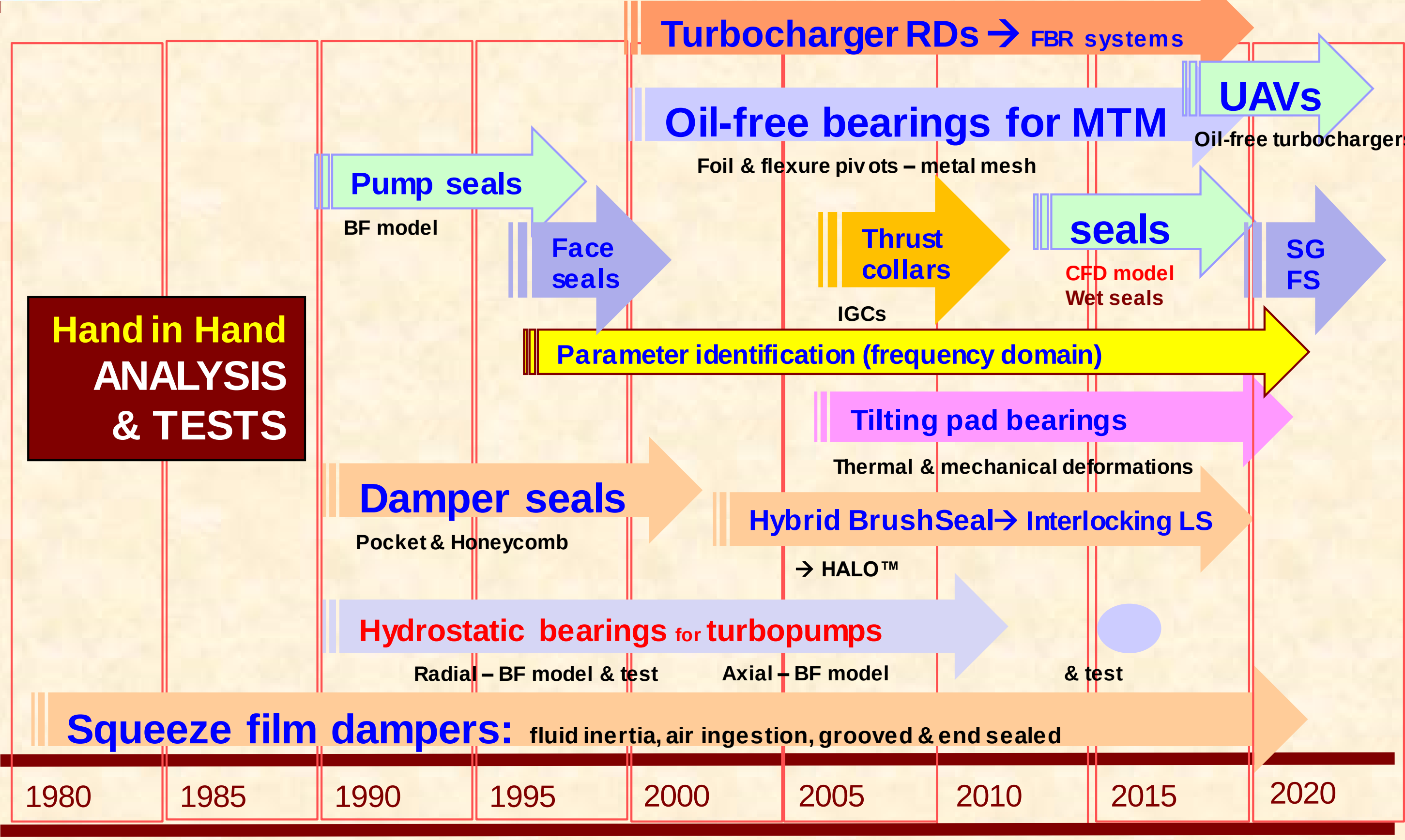
Borg-Warner TC,
Torishima Pumps
MHI, Hitachi RL,
Samsung, Key Yang,
Hyundai HI, Capstone MT
Siemens TRC

Hand in Hand
ANALYSIS
& TESTS



Turbocharger RDs → FBR systems

Oil-free bearings for MTM

Foil & flexure pivots – metal mesh

UAVs

Oil-free turbocharger

Pump seals

BF model

Face seals

Thrust collars

IGCs

seals

CFD model
Wet seals

SG FS

Parameter identification (frequency domain)

Tilting pad bearings

Thermal & mechanical deformations

Damper seals

Pocket & Honeycomb

Hybrid BrushSeal→ Interlocking LS

→ HALO™

Hydrostatic bearings for turbopumps

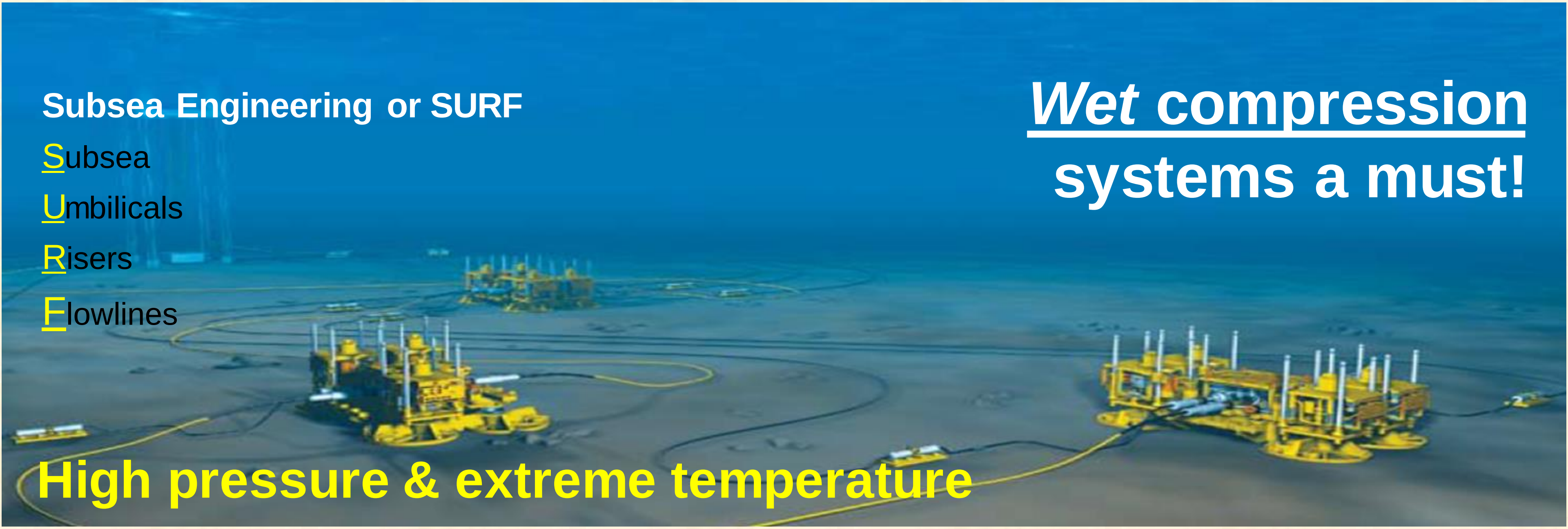
Radial – BF model & test

Axial – BF model

& test

Squeeze film dampers: fluid inertia, air ingestion, grooved & end sealed

A need: subsea pumping & compression



Subsea Engineering or SURF

Subsea
Umbilicals
Risers
Flowlines

Wet compression systems a must!

High pressure & extreme temperature

Bloomberg 7/30/19: Offshore oil production tops shale oil on generation of jobs.

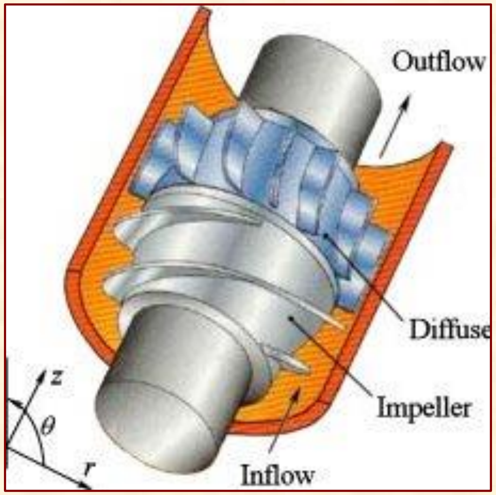
Extreme engineering enables five year or longer reliability for subsea production facilities (North Sea & Brazil → Gulf of Mexico → Artic).

Pros/cons of two-phase flow operation

	Multiphase pumping	**Wet gas compression**	**Hydraulic turbine/pumps**
Applications	Onshore, offshore, subsea and downhole **GVF 0 -100%** [1]	Subsea and downhole **LVF 0 – 5%** [2]	Power generation
Benefits	Add pressure to process fluids, enabling long distance tie back system to reduce O&G separation stations. **Cost drops ~ 30%**		Clean energy
Challenges	**Rotor sub-synchronous vibrations**		Often suffer from non-synchronous vibration even at null speed [3]

[1] **Gong, H., et al**., **2012**, "Comparison of Multiphase Pumping Technologies for Subsea and Downhole Applications." Oil and Gas Facilities, **1**(01), pp. 36-46.
[2] **Vannini, G., et al., 2014**, "Centrifugal Compressor Rotordynamics in Wet Gas Conditions." *Proc. of the* 43th *Turbomachinery &* 30th *Pump Users Symposia*, Houston, TX, September 23-25.
[3] **Smith, et al., 1996**, "Centrifugal Pump Vibration Caused by Supersynchronous Shaft Instability Use of Pumpout Vanes to Increase Pump Shaft Stability." *Proc.* 13*th International Pump Users Symposium*, Houston, TX, Mar. 5-7.

Current knowledge

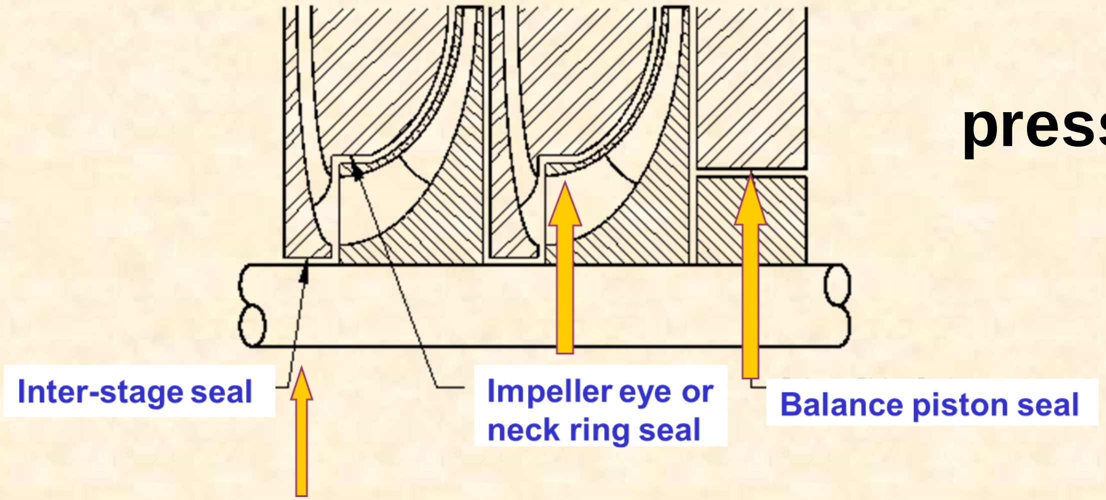


Cost efficient subsea factories must rely on multiple-phase flow compression and pump systems that reduce tieback systems and perform full flow separation on the sea floor,

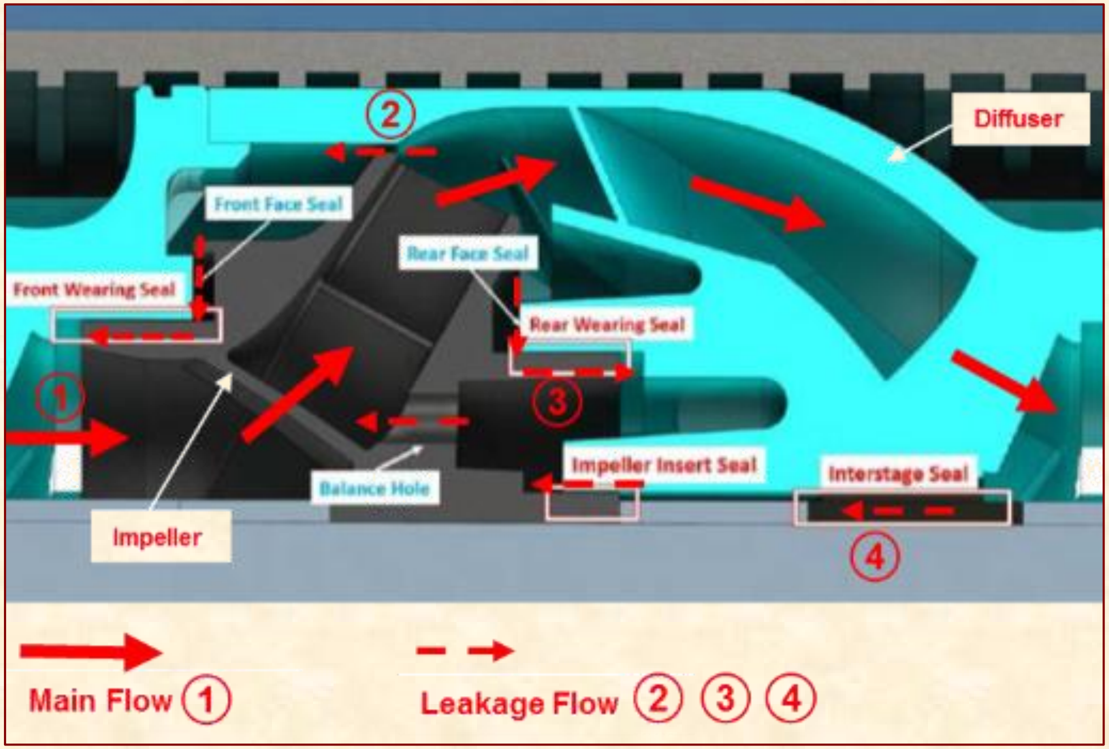
but rotor dynamic stability is an issue.

Annular Pressure Seals

separate regions of high pressure and low pressure to minimize the leakage (secondary flow).



Multiple phase pump



Two-phase flow in a *wet gas* compressor

Rotor lateral vibration

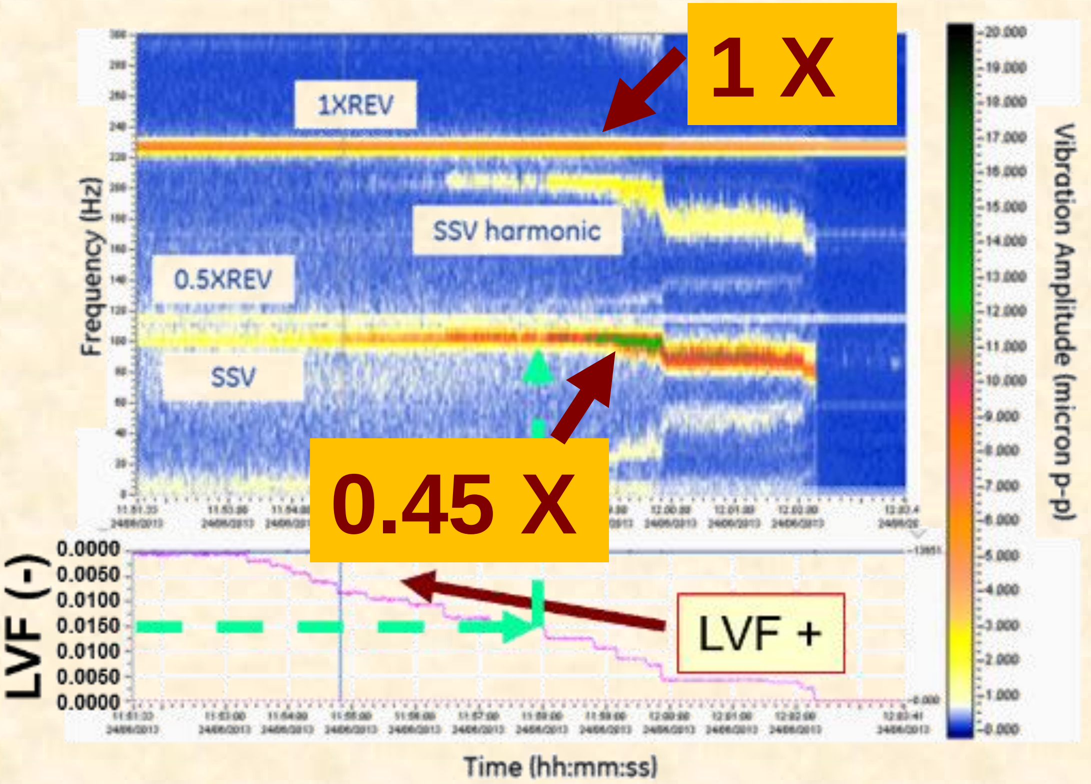
13.5 krpm, 10 bar

Balance piston:
Labyrinth seal

Fluids:
Air and water

LVF: 0~3%

0.45 X SSV increases in magnitude with LVF



Trapped liquid in seal rotates and causes SSV

Vannini et al., **2016**, "Experimental Results and CFD Simulations of Labyrinth and Pocket Damper Seals for Wet Gas Compression," ASME J. Eng. Gas Turb. Power, **138**, p. 052501.

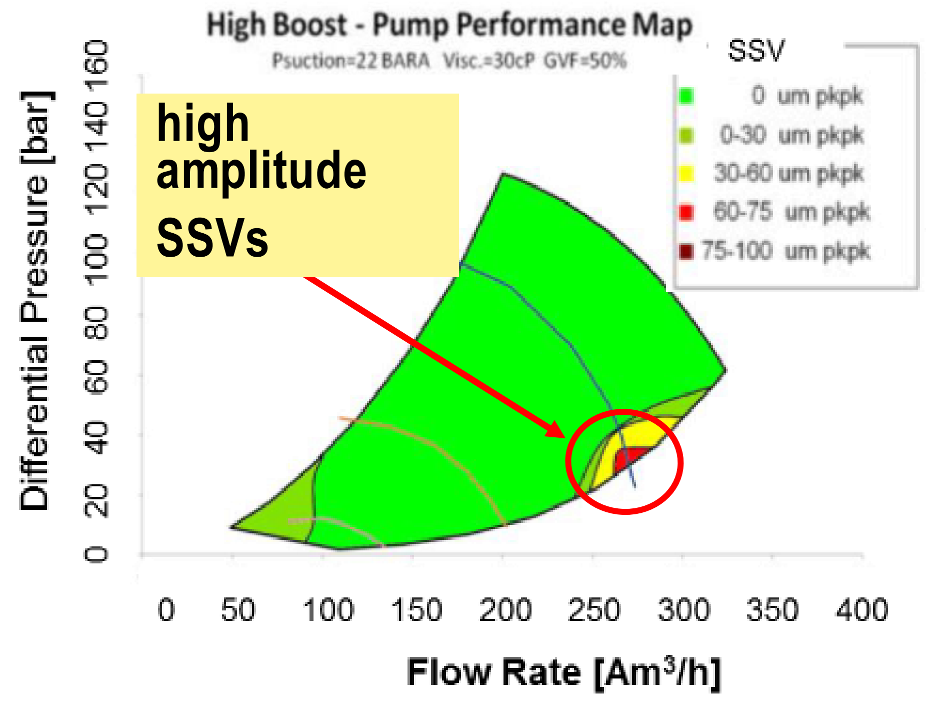
Two-phase flow in a pump

Helico-axial pump
(1.5 to 4.6 krpm)

Pump operates stable with liquid.
(600 cPoise)

Rotor SSV appears under some two-phase flow conditions : **low differential pressure with a high-viscosity mixture.**

When SSV occurs, rotor **whirl frequency ratio > 1.0.**



high amplitude SSVs

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.

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 eliminate oil and gas separation stations.

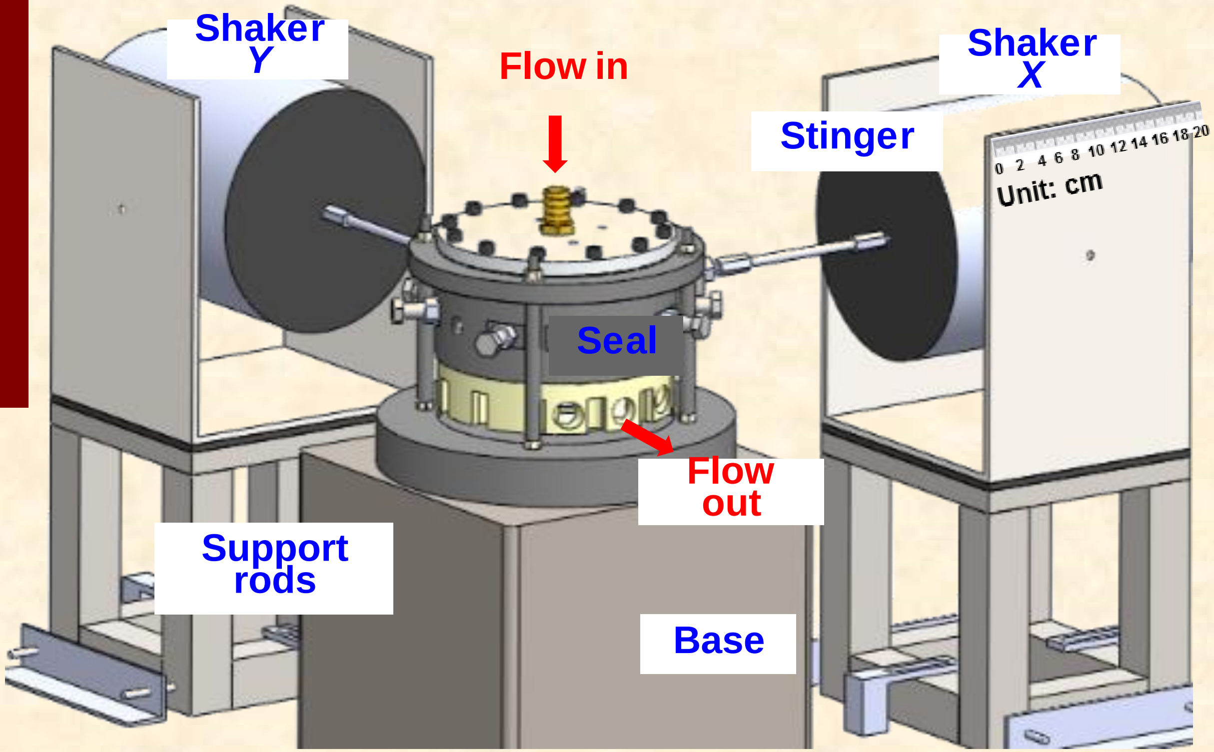
Seals must be able to operate without affecting the system efficiency and dynamic stability.

The lecture presents measurements of leakage and force coefficients for several annular clearance seals operating with an air in oil mixture ranging from pure liquid to mostly air.

Knowledge (learning) today

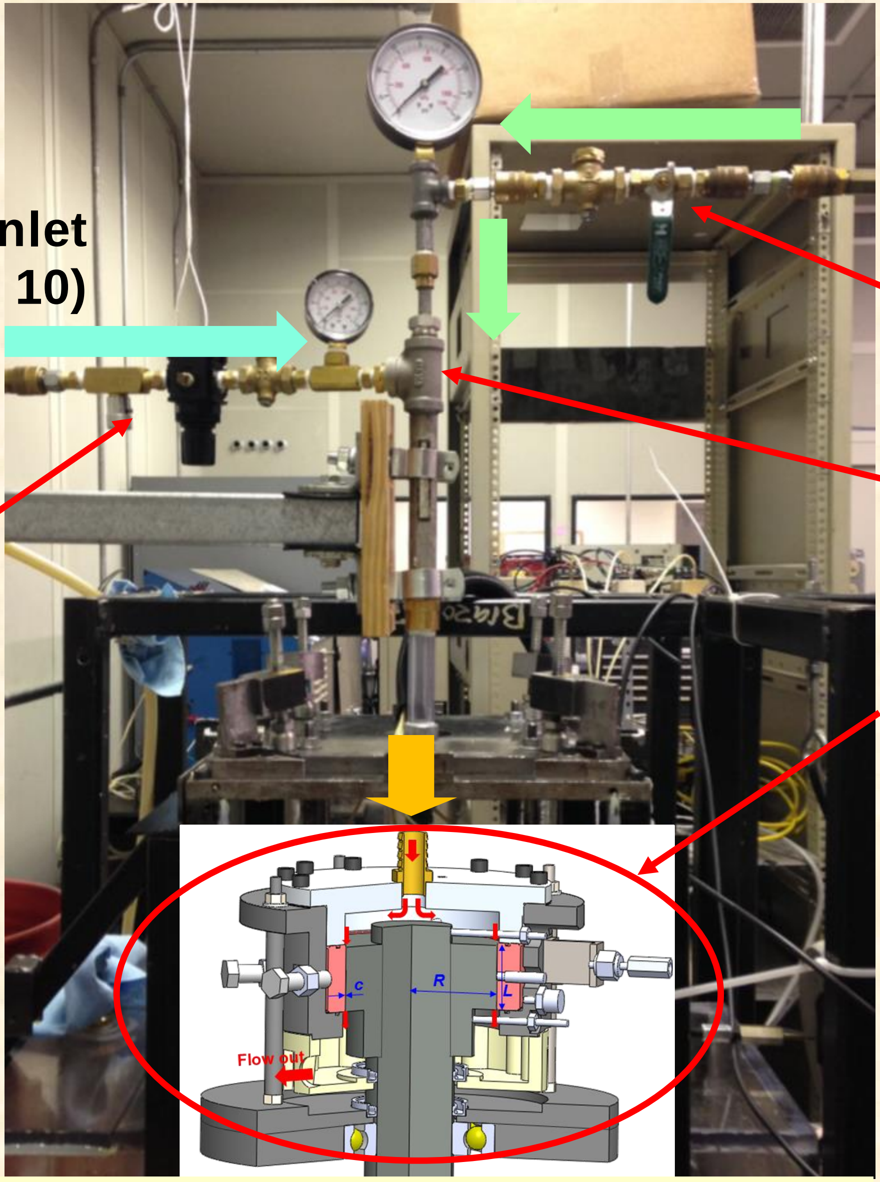
1. Why *wet* (bubbly) seals? Where are they found?
2. How does gas content affect seal leakage and drag?
3. How does gas content affect the stiffness and damping coefficients of a *wet* seal?
4. Why a wavy surface seal is a better option than a plain seal for a two phase flow pump?
5. Why gas injection increases the centering stiffness of seals in pumps & hydraulic turbines?

Wet Gas Test Rig



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

Wet seal test rig



Oil Inlet
(ISO VG 10)

Valve

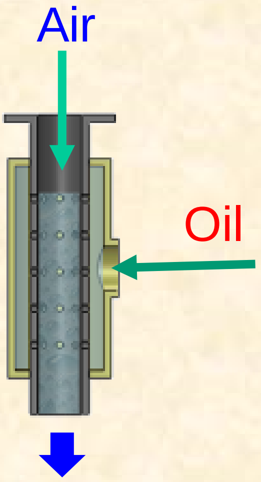
Air Inlet

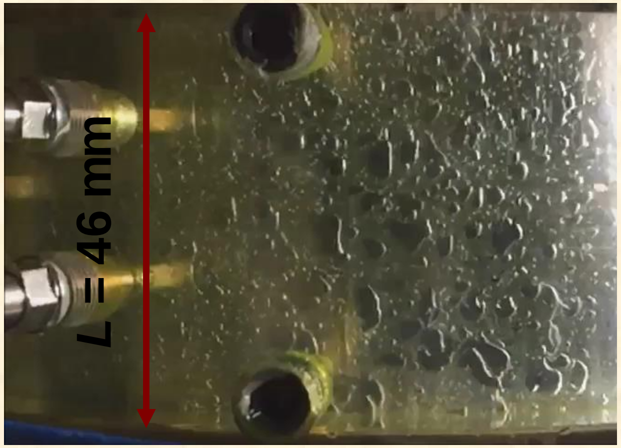
Valve

Sparger
(mixing)
element

Test seal
section

journal speed: 3.5 krpm (23.3 m/s)

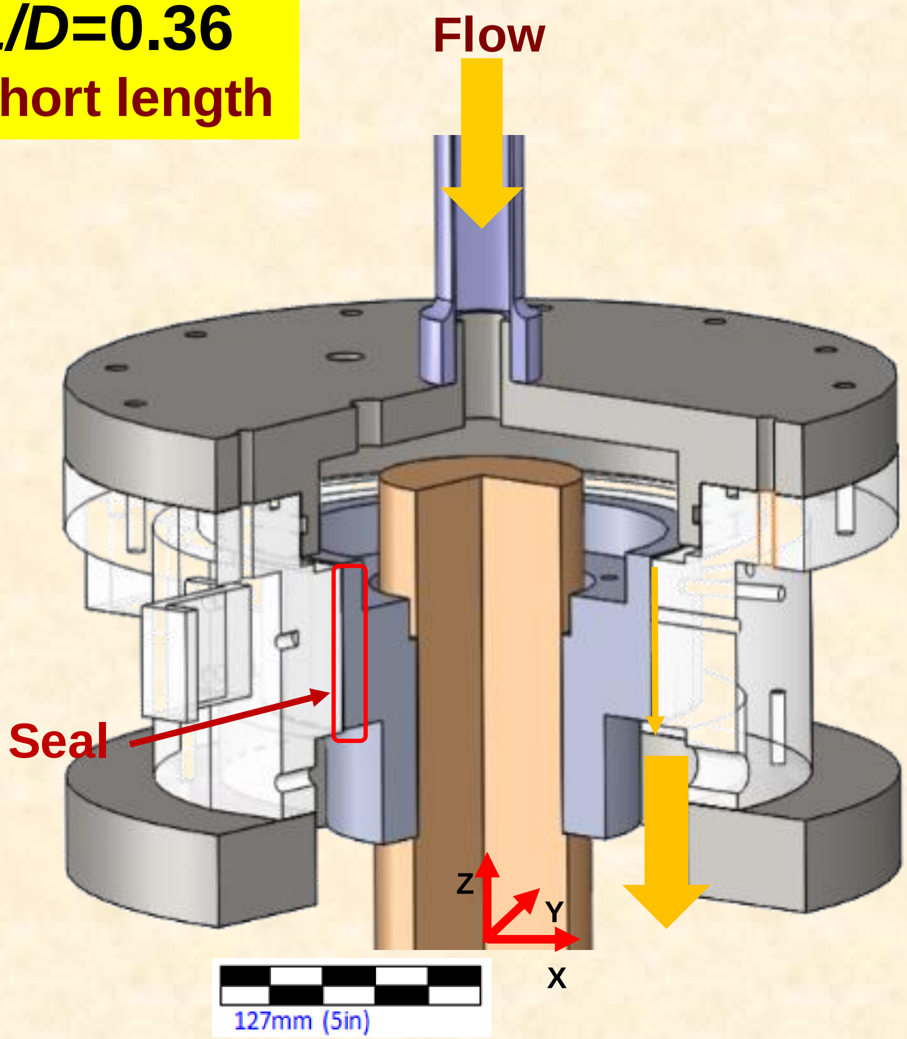




P_s/P_a=2.5, inlet GVF=50%,
stationary shaft

Seal geometry and fluids

L/D=0.36
short length



Seals	
Diameter (D)	**127 mm** (5 in)
Length (L)	**46 mm** (1.8 in)
Clearance (c) 34 °C	**0.203 mm** (8 mil)
Supply pressure (P_s)	1.0~3.5 bar (abs)
Oil ISO VG 10 density(ρ_l)	**830 kg/m^3**
viscosity (μ_l)	**10.6 cP** at 34 °C
Air density (ρ_{ga})	**1.2 kg/m^3** at 1bar
viscosity(μ_{ga})	**0.02 cP** at 20 °C, 1 bar (abs)
Shaft speed (Ω_{max})	3.5 krpm
surface speed ½ $D\Omega_{max}$	**23 m/s**

Sparger pore size	2 µm
Air bubble size	Up to 4 mm

Five test seals

$D/c \sim 640$

Smooth surface plain seal

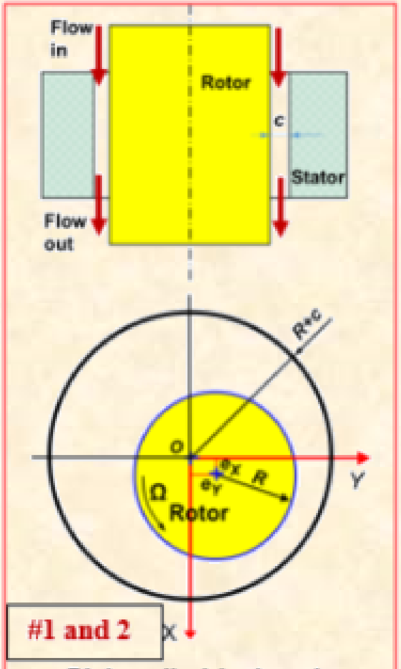
Nominal ***c*** and **worn** (>*c*)

Three-wave seal:

Large dynamic stiffness

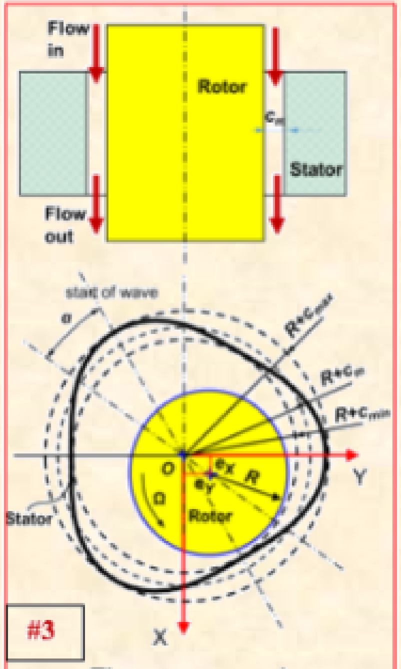
(Rim) step clearance seals:

Used in hydraulic turbines/pumps.



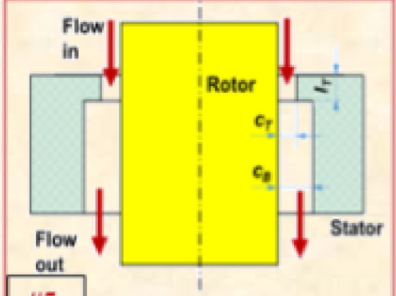
#1 and 2

Plain cylindrical seal



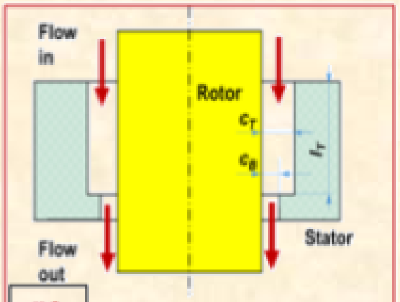
#3

Three-wave seal



#5

Upstream step clearance seal



#6

Downstream step clearance seal

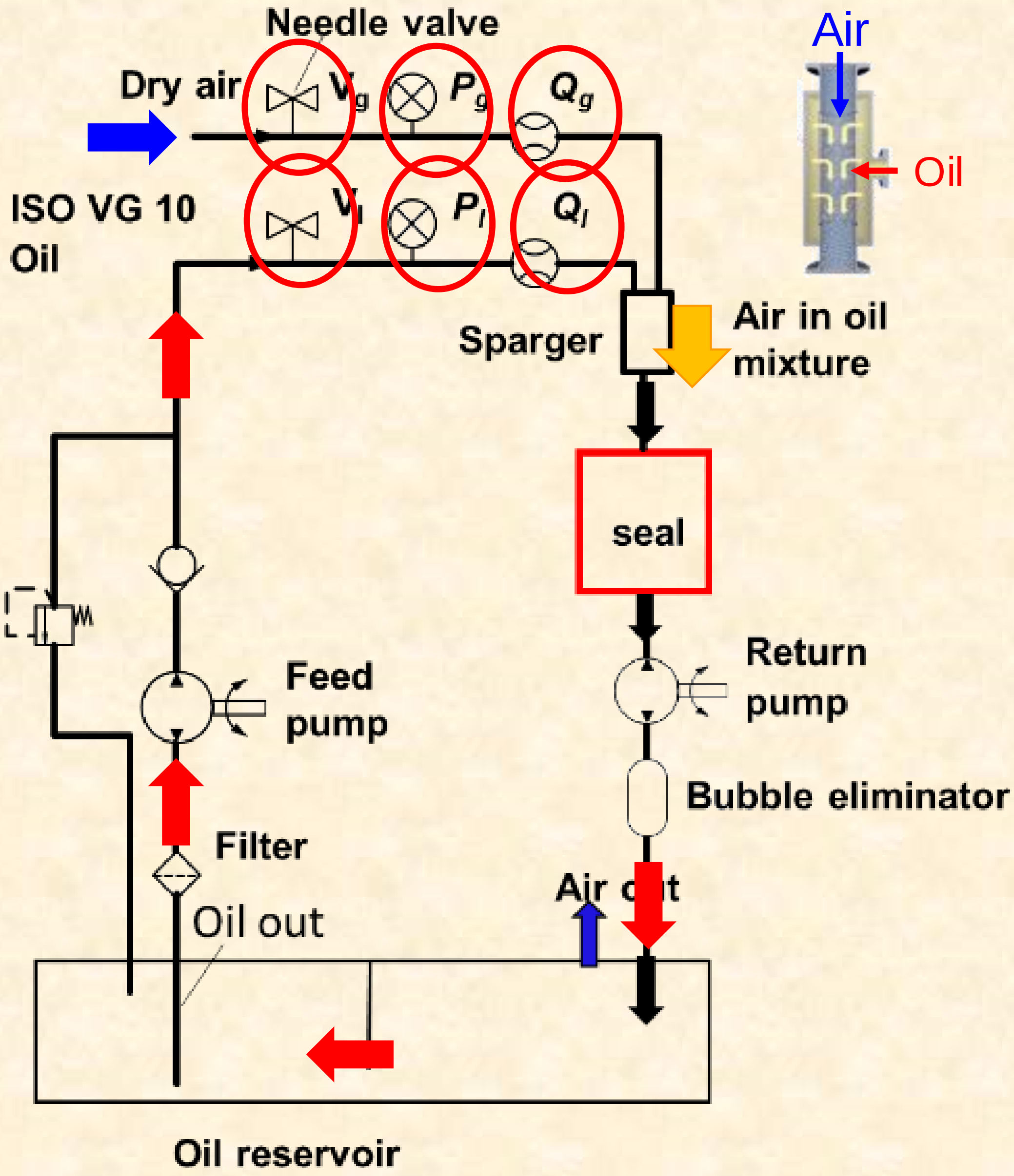
Plain seals #1 & 2:
c_1= 0.203 mm,
c_2 = 0.274 mm
(worn clearance)

#3
Three-wave seal
(c_m=0.191 mm)

#4
Upstream step clearance
(c_T=0.164mm, c_B=0.274 mm, L_T=0.11L).

#5
Downstream step clearance
(c_T=0.274 mm, c_B=0.164 mm, L_T=0.82L).

Air and oil circulation systems



α : Gas volume fraction

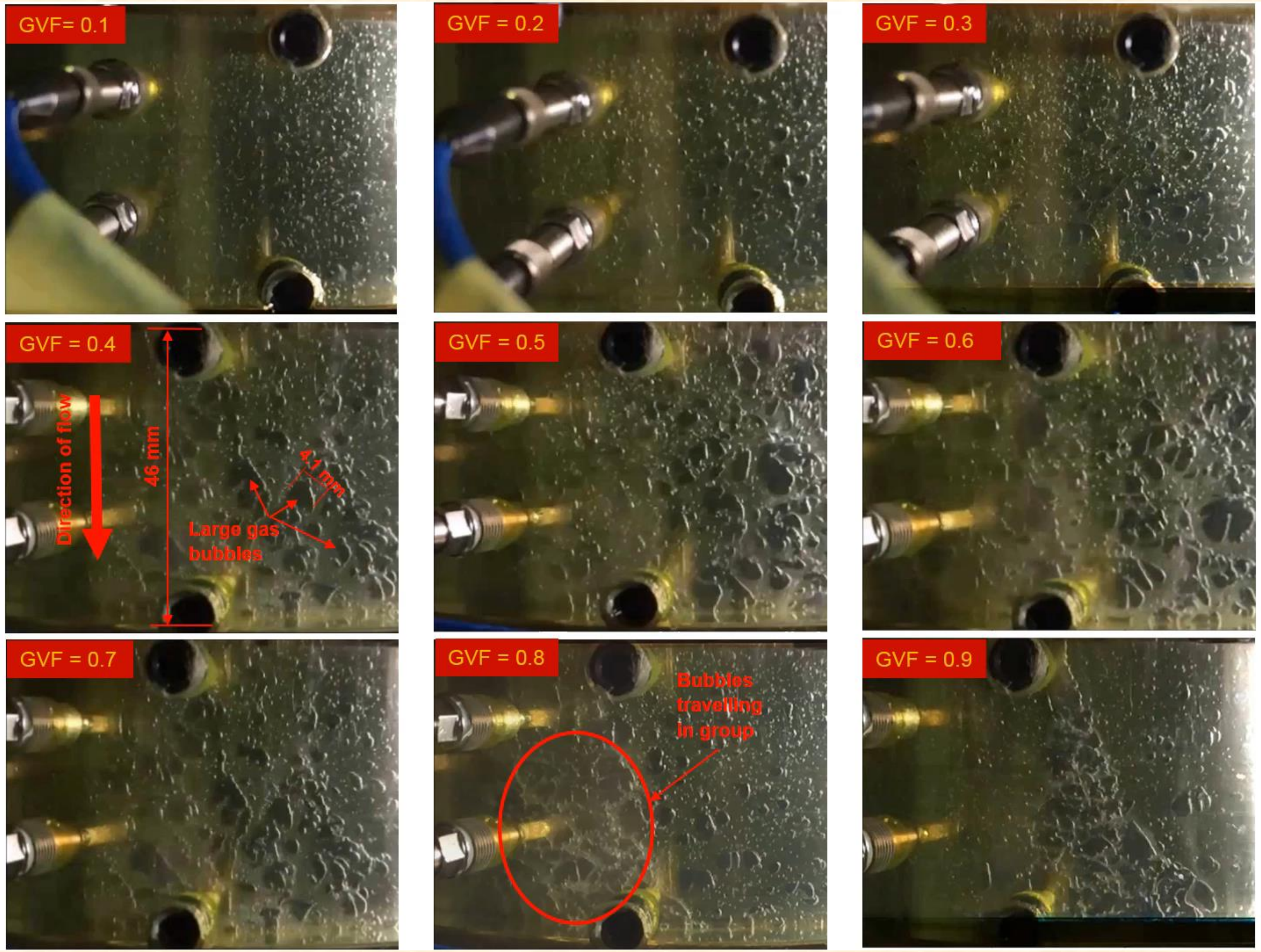
P_s: pressure at seal inlet plane
P_a: ambient pressure= 1 bar(a)

Q_g: gas flow rate at P_s
Q_l: liquid flow rate

GVF at inlet:

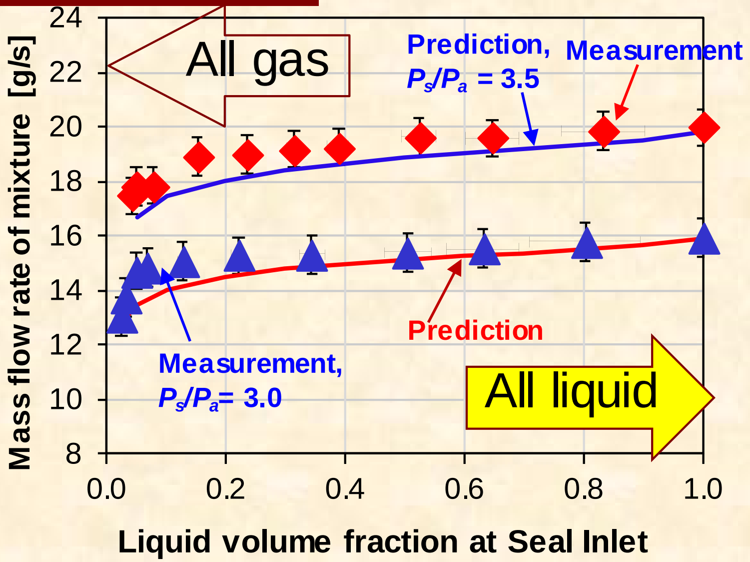
$$\alpha_{in} = \frac{Q_g\left(P_a/P_s\right)}{Q_l + Q_g\left(P_a/P_s\right)}$$

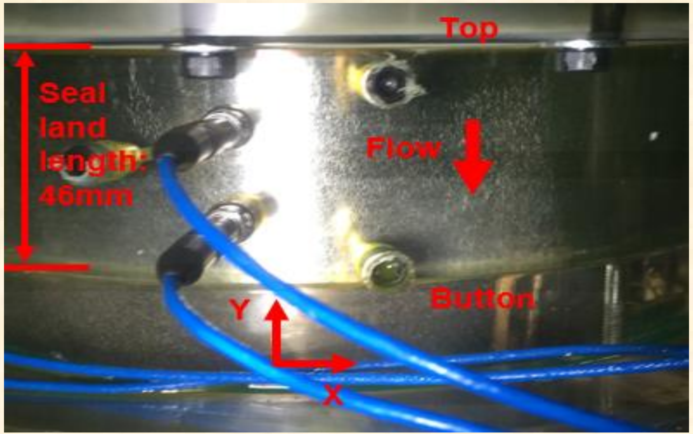
Flow visualization → inlet GVF = 0-0.9. *Ps*/*Pa*=2.5. Speed 0 rpm



Plain seal: flow rate vs LVF (0 rpm)

Leakage

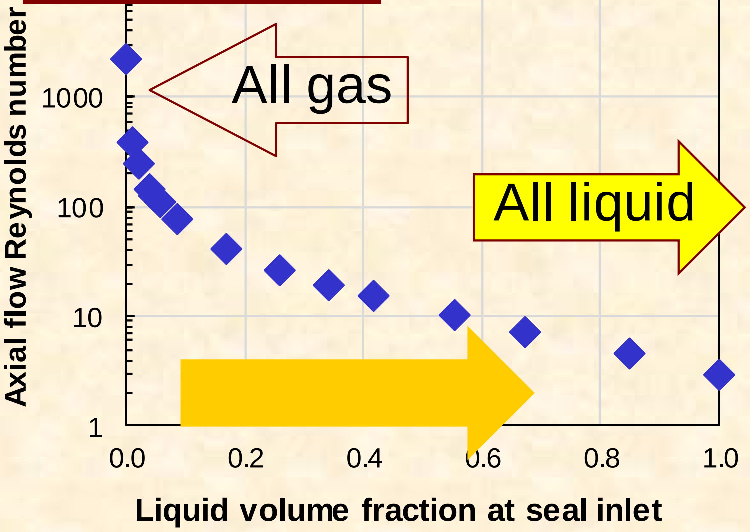




P_s/P_a=1.5, inlet LVF= 2%

Clearance ~ 0.2 mm

Reynolds



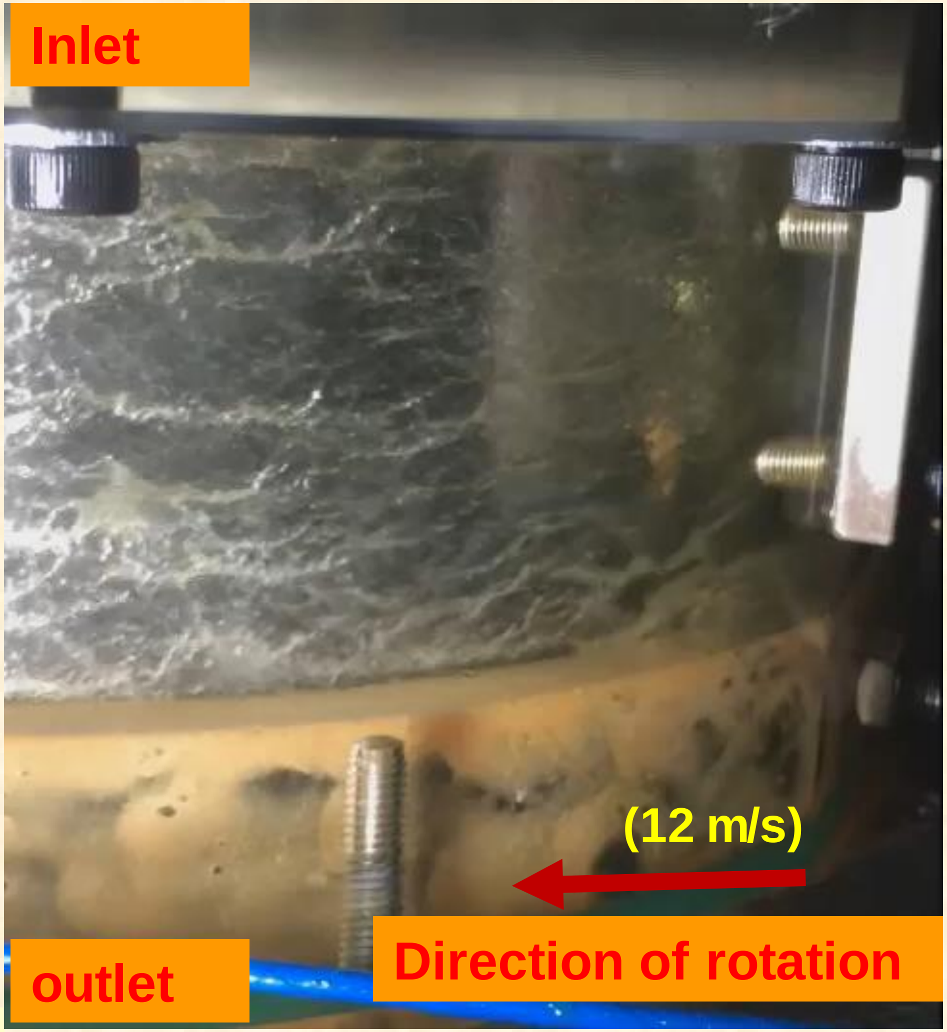
- **Leakage increases with inlet LVF.**
- **Reynolds # drops from > 1,000 (air) to low magnitude as LVF increases.**

Flow with shaft spinning

$P_s/P_a = 2$, speed 1.8 krpm

Stroboscope light with frequency 30 Hz freezes shaft motion

Air bubbles coalesce and merge to make streamlets →



Laminar flow Reynolds #:
Re_c = 153, Re_z = 245 at exit plane

Seals’ leakage and drag torque

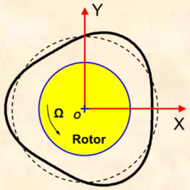
Leakage (oil only) LVF=1 (liquid only)

Normalized to:

$$m_l = \frac{1}{12}\frac{\rho_l}{\mu_l}\pi D c^3 \frac{\Delta P}{L}$$

Three-wave seal leaks more than plain seal.

Plain seals #1 & 2:
(c_1 = 0.203 mm, c_2 = 0.274 mm)



#3
Three-wave seal
(c_m=0.191 mm)

#4
Grooved seal
(c_r=0.211 mm)

Upstream step clearance
(c_T=0.164mm, c_B=0.274 mm, L_T=0.11L).

Downstream step clearance
(c_T=0.274 mm, c_B=0.164 mm, L_T=0.82L).

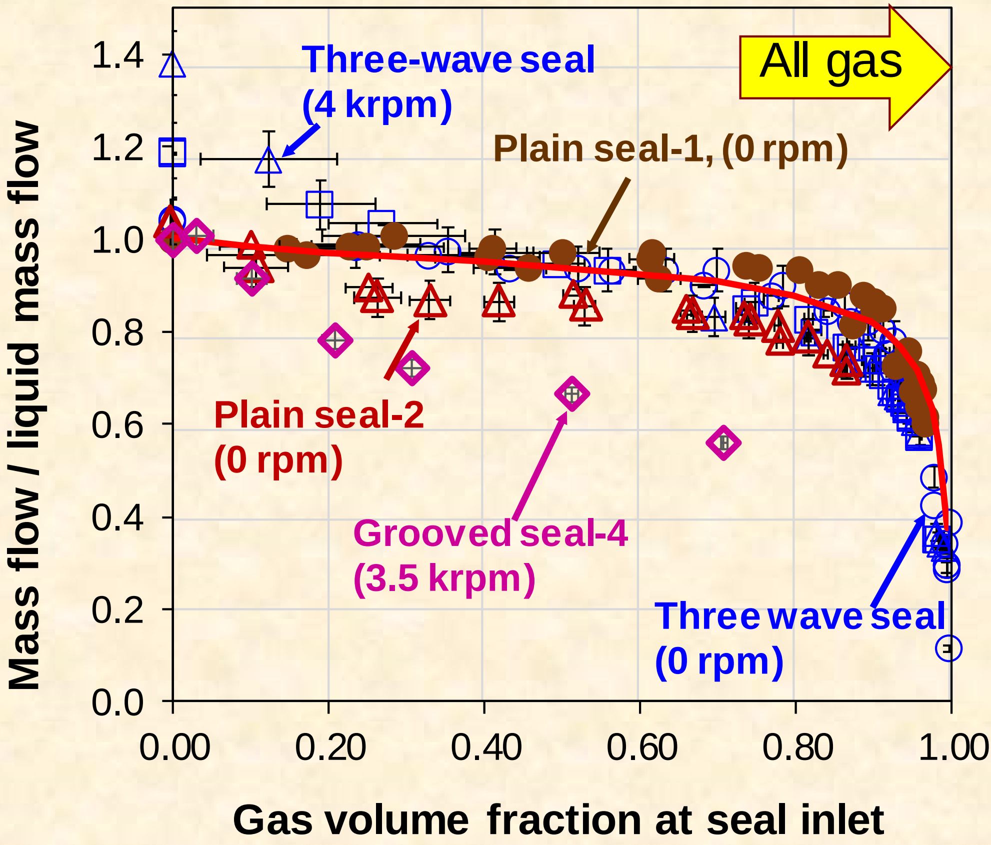
Leakage (Mixture) → gas volume fraction increases

Normalized with respect to liquid (GFV=0)

$$m = \frac{m_{mixture}}{m_{liquid}}$$

Leakage for all seals shows same trend as GVF increases → it drops!

Predictions agree with test data.



$C_{seal\#1}$ = 0.203 mm; $C_{seal\#2}$ = 0.274 mm

$C_{seal\#3}$ = 0.191 mm; $C_{seal\#4}$ = 0.211 mm

Drag torque (mixture) T_{seal}

Shaft speed:
1.5, 2.5, 3.5 krpm

normalized to all liquid condition

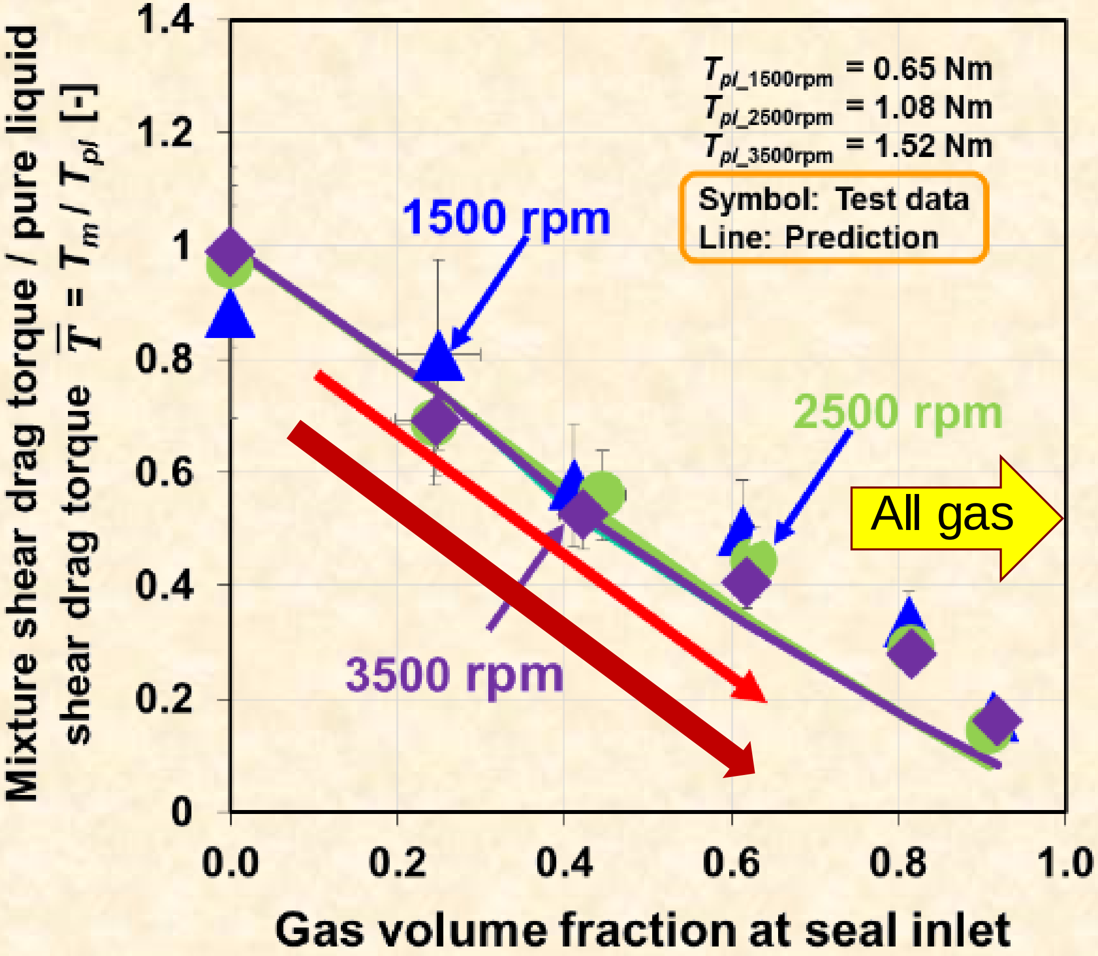
prediction

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

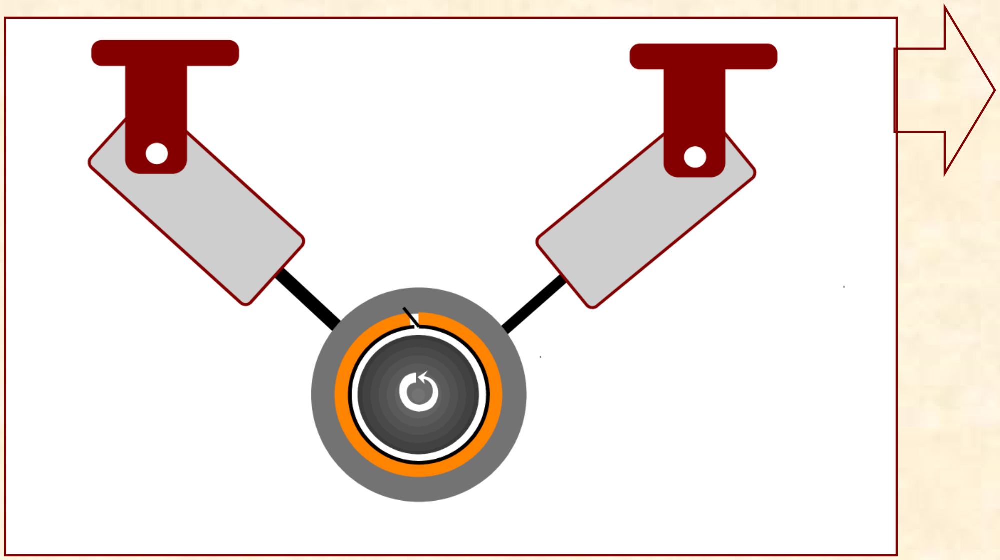
Torque <u>linearly</u> decreases with GVF.

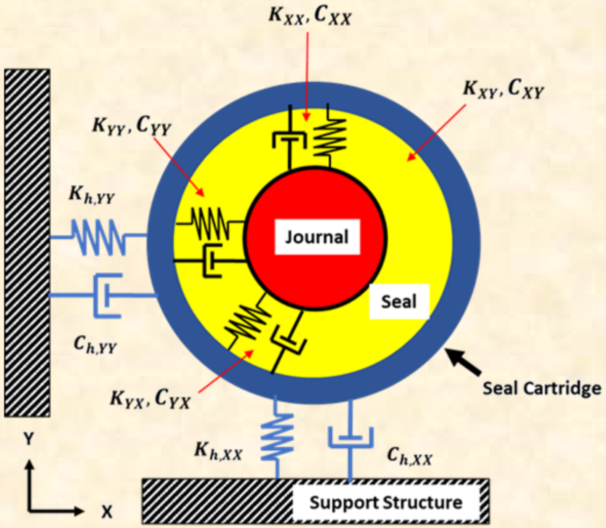
GVF = 0 → 0.9
85% reduction in torque

Grave implication for pump and motor reliable performance.



Experimental identification of force coefficients

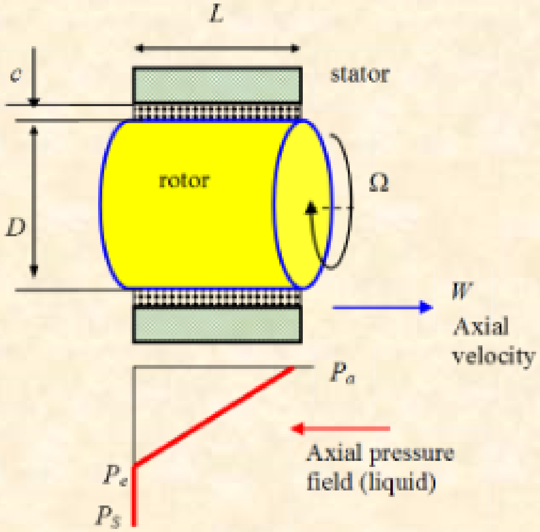


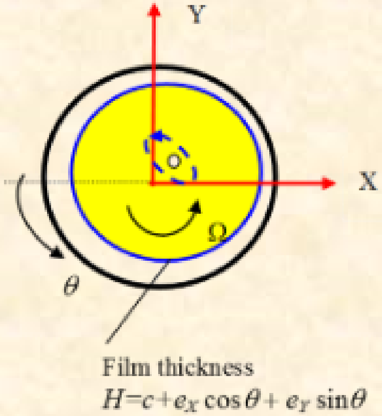


2-DOF system for seal and support structure

Dynamic force coefficients

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





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

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

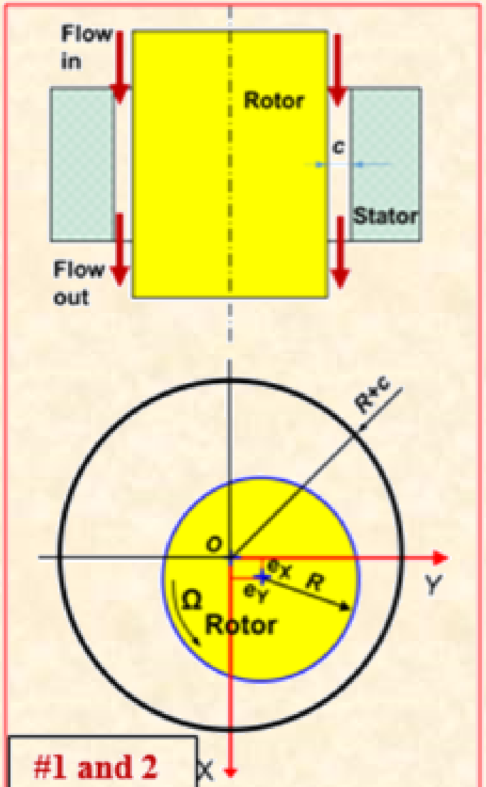
Components of seal complex stiffness H

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

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

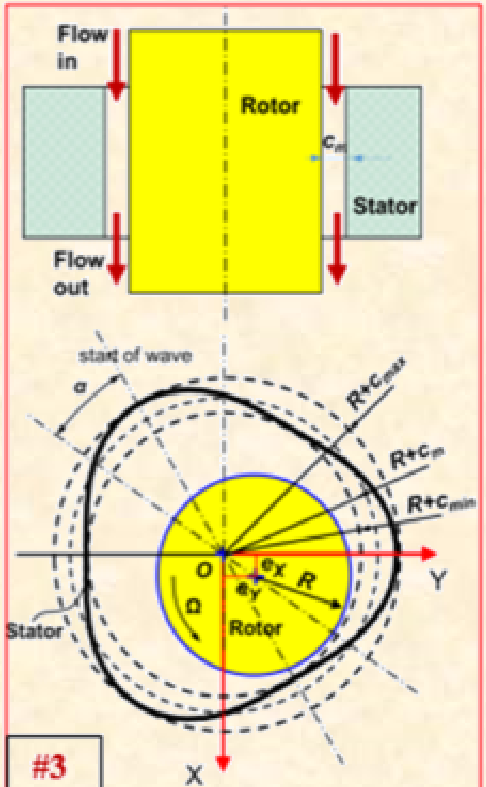
$C_{eff} = C - k/\omega = [\mathrm{Im}(H_{xx}) - \mathrm{Re}(H_{xy})]/\omega$ **Effective Damping**

Force coefficients for plain cylindrical seals and three-wave seal



#1 and 2

Plain cylindrical seal



#3

Three-wave seal

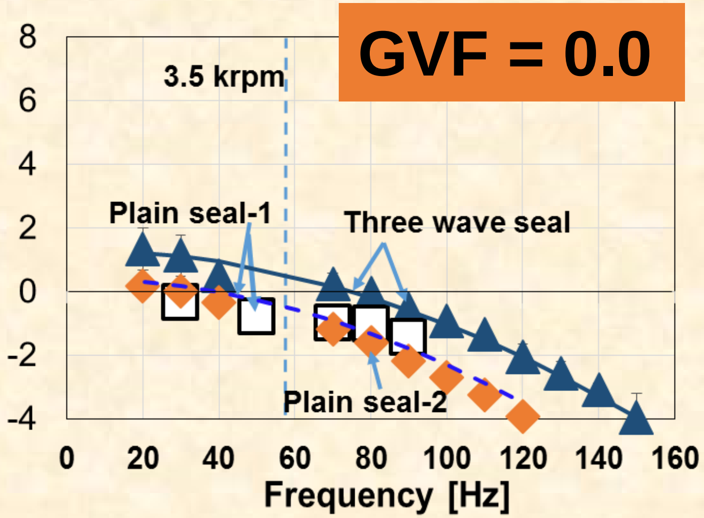
#1 & # 2
Plain seals
c_1=0.203 mm, c_2=0.274 mm **(worn)**

#3
Three-wave seal c_m=0.191 mm

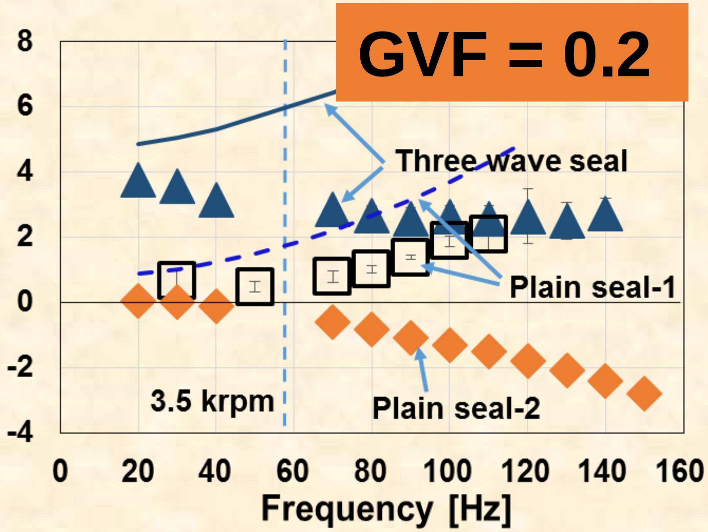
Direct dynamic stiffness *K* (MN/m)

Symbols: test results **Lines: predictions**

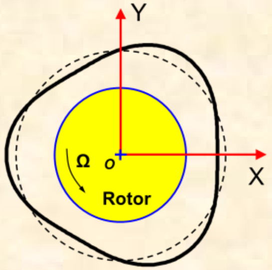
- ◻ Plain seal-1
- ◆ Plain seal-2
- ▲ Three-wave seal

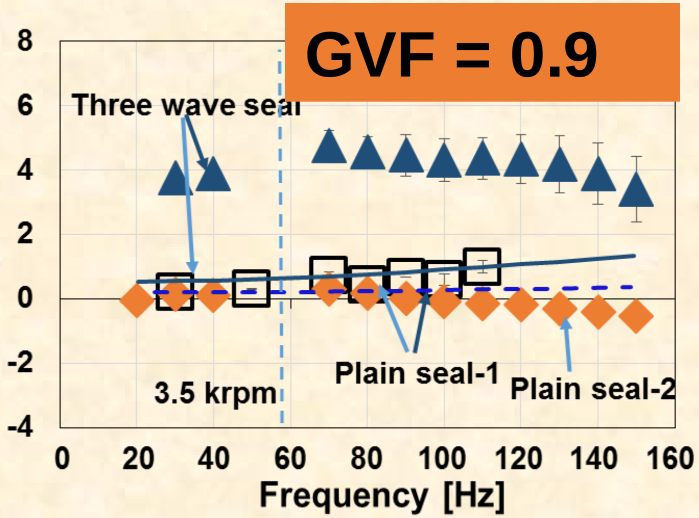


GVF = 0.0



GVF = 0.2



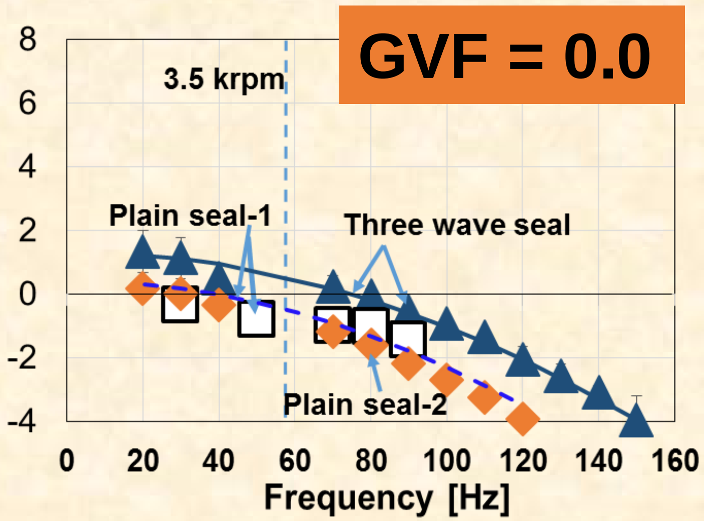


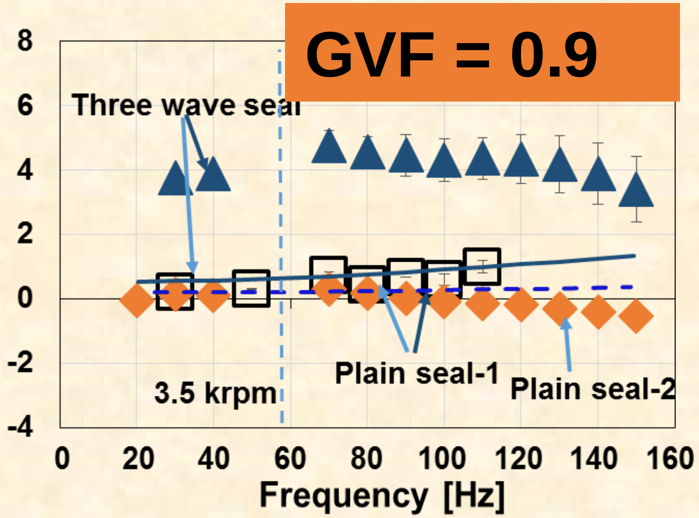
GVF = 0.9

Direct dynamic stiffness *K* (MN/m)

Symbols: test results Lines: predictions

□ Plain seal-1
◆ Plain seal-2
▲ Three-wave seal


GVF = 0.0


GVF = 0.9

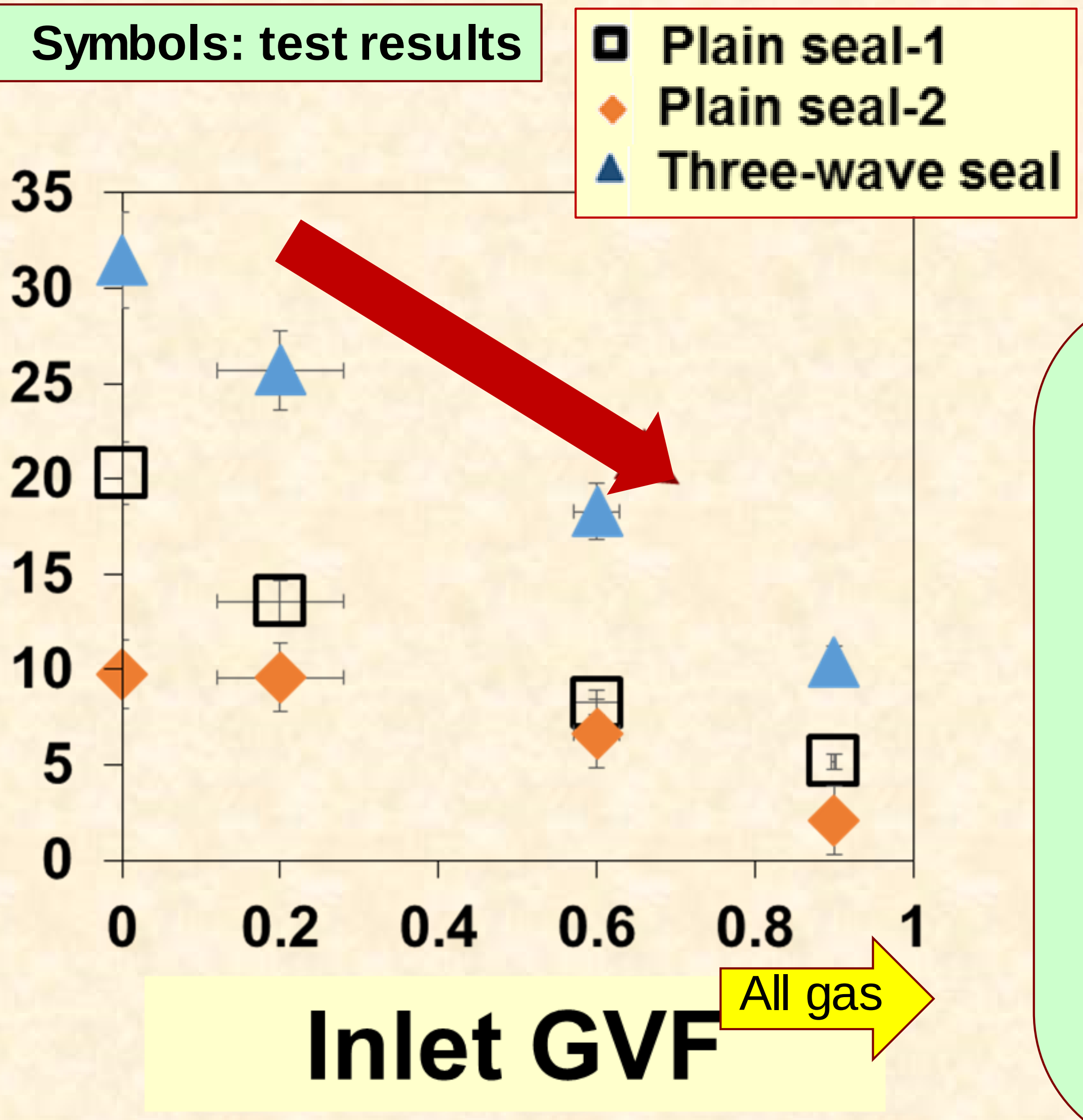
Three wave seal (#3) has largest *K* (promotes static stability).

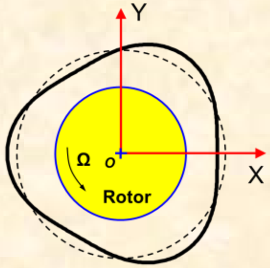
Worn seal (#2) shows lowest *K*.

***K* : soft to hard as GVF increases. Lesser added mass!**

Direct damping coefficient *C* (kN.s/m)

Symbols: test results





***C* is frequency independent**

Three wave seal (#3) has largest *C*.

Worn seal (#2) shows smallest *C*.

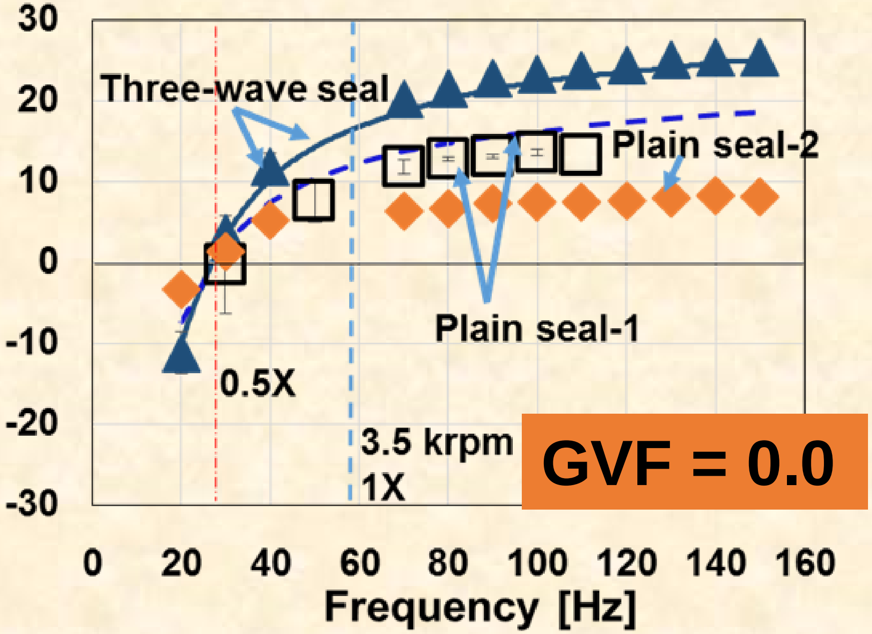
***C* drops with GVF**

$$C \sim C_l\,(1-\mathrm{GVF})$$

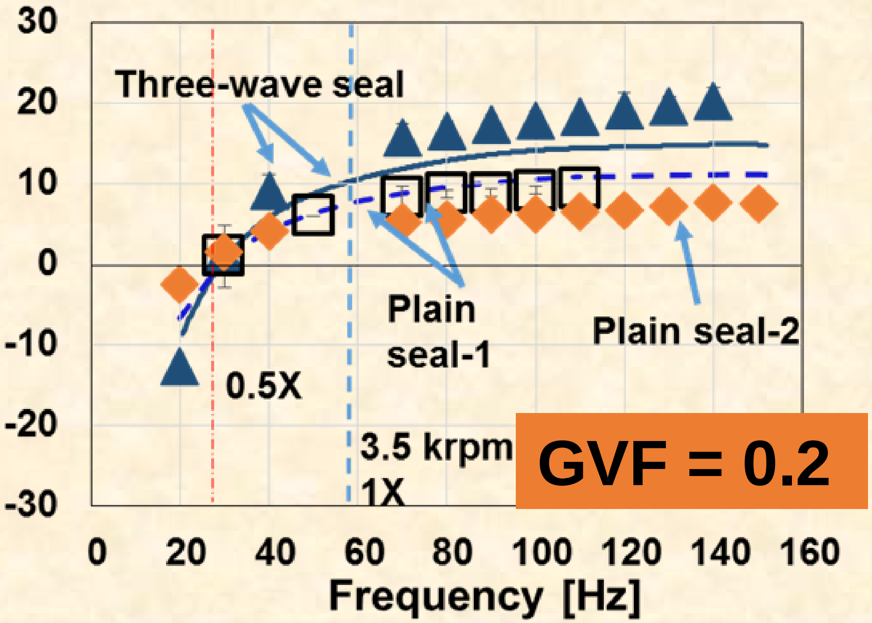
Effective damping (kN.s/m)

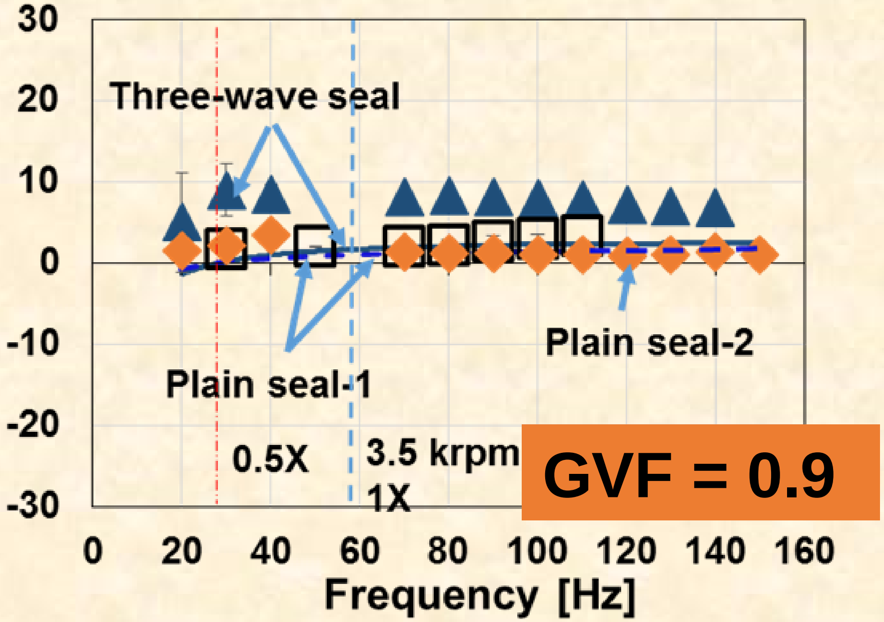
$C_{eff} = C - k/\omega$

Symbols: test results Lines: predictions



■ Plain seal-1
◆ Plain seal-2
▲ Three-wave seal

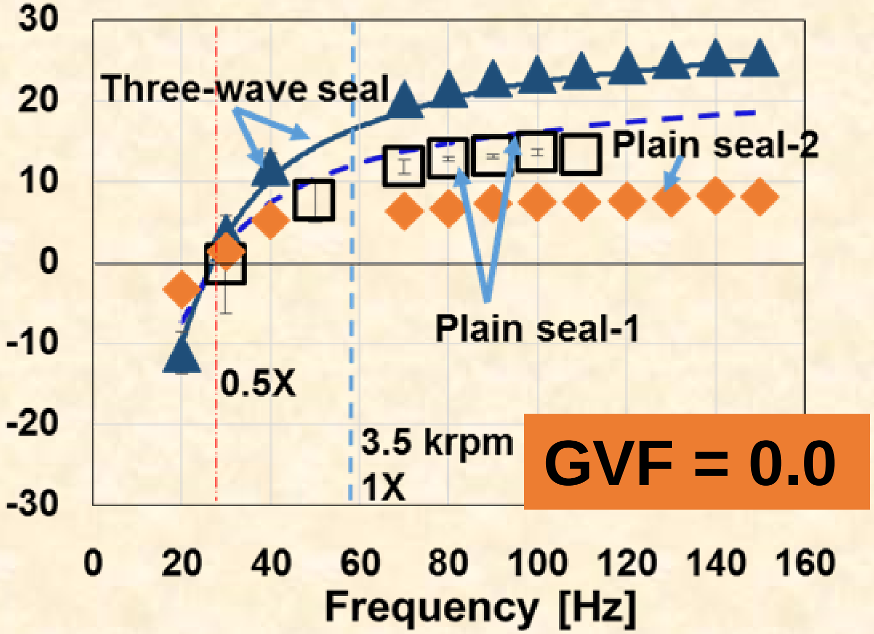




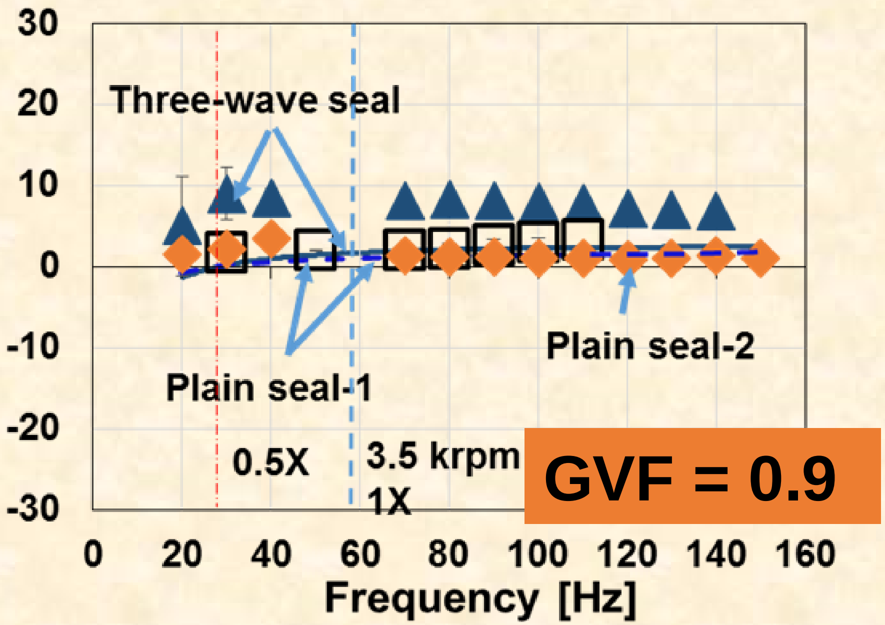
Effective damping (kN.s/m)

$C_{eff} = C - k/\omega$

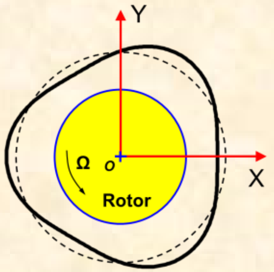
Symbols: test results **Lines: predictions**



GVF = 0.0



GVF = 0.9



- □ Plain seal-1
- ◆ Plain seal-2
- ▲ Three-wave seal

For stability, $C_{eff} > 0$ is a must.

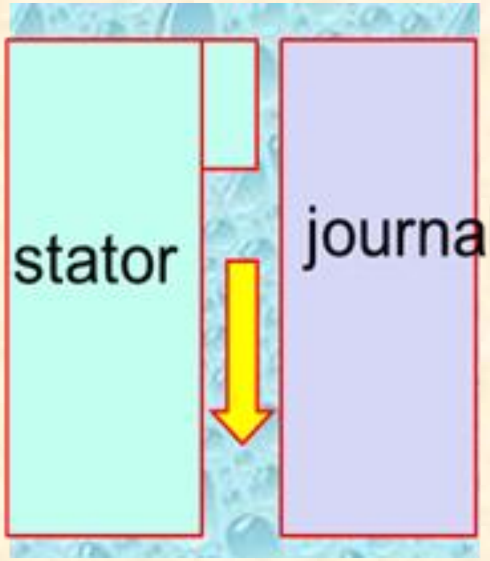
Increase in GVF → C_{eff} drops.

Cross frequency drops from ~ ½ X to a low magnitude.

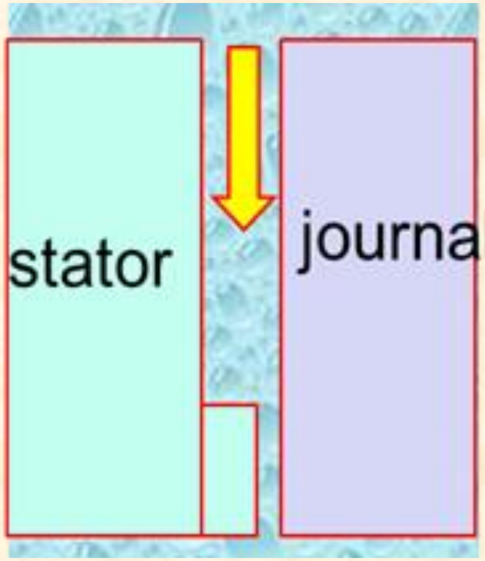
Force coefficients for step clearance seals

Typical rim seals in hydraulic turbines

Upstream step clearance
c_T=0.164 mm, c_B=0.274 mm, L_T=0.11L



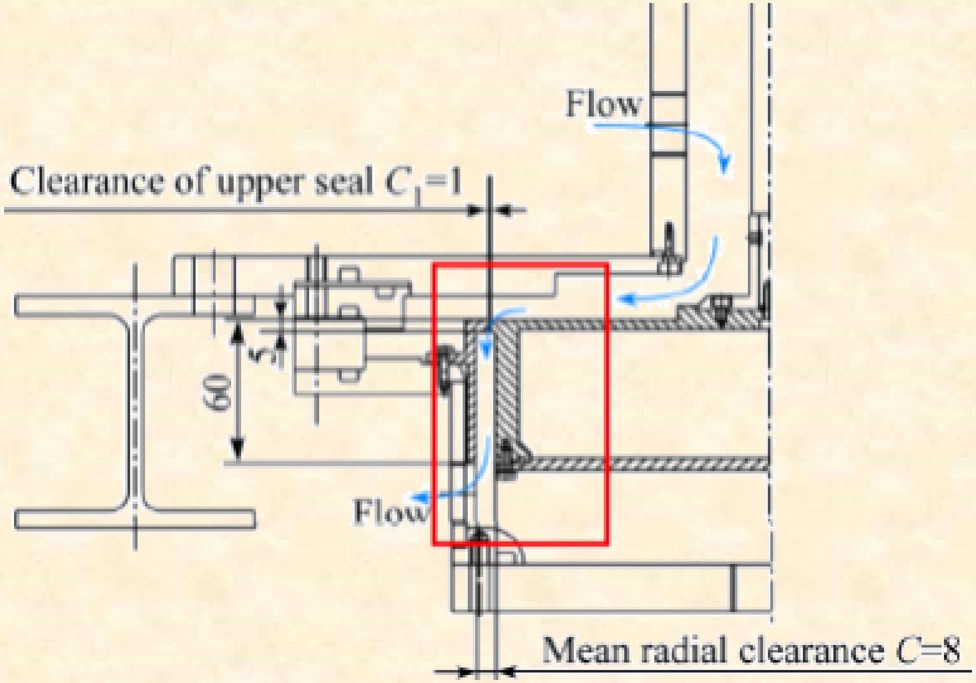
Downstream step clearance
c_T=0.274 mm, c_B=0.164 mm, L_T=0.82L.



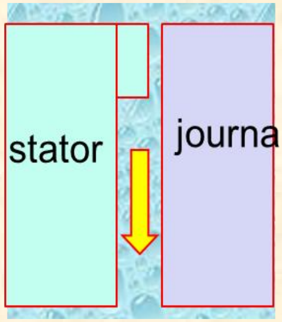
Direction of flow

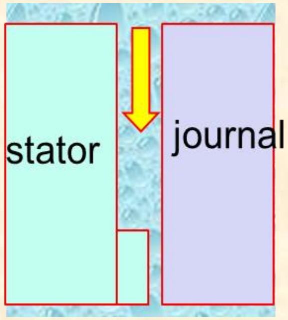
Step clearance seals in hydraulic turbines

- **Pump-turbines installed with (rim) upstream step clearance seals vibrate at a natural frequency (below structural one) & even w/o shaft rotation.**



- **But these units do not (self) vibrate when installed with a downstream step clearance seal.**



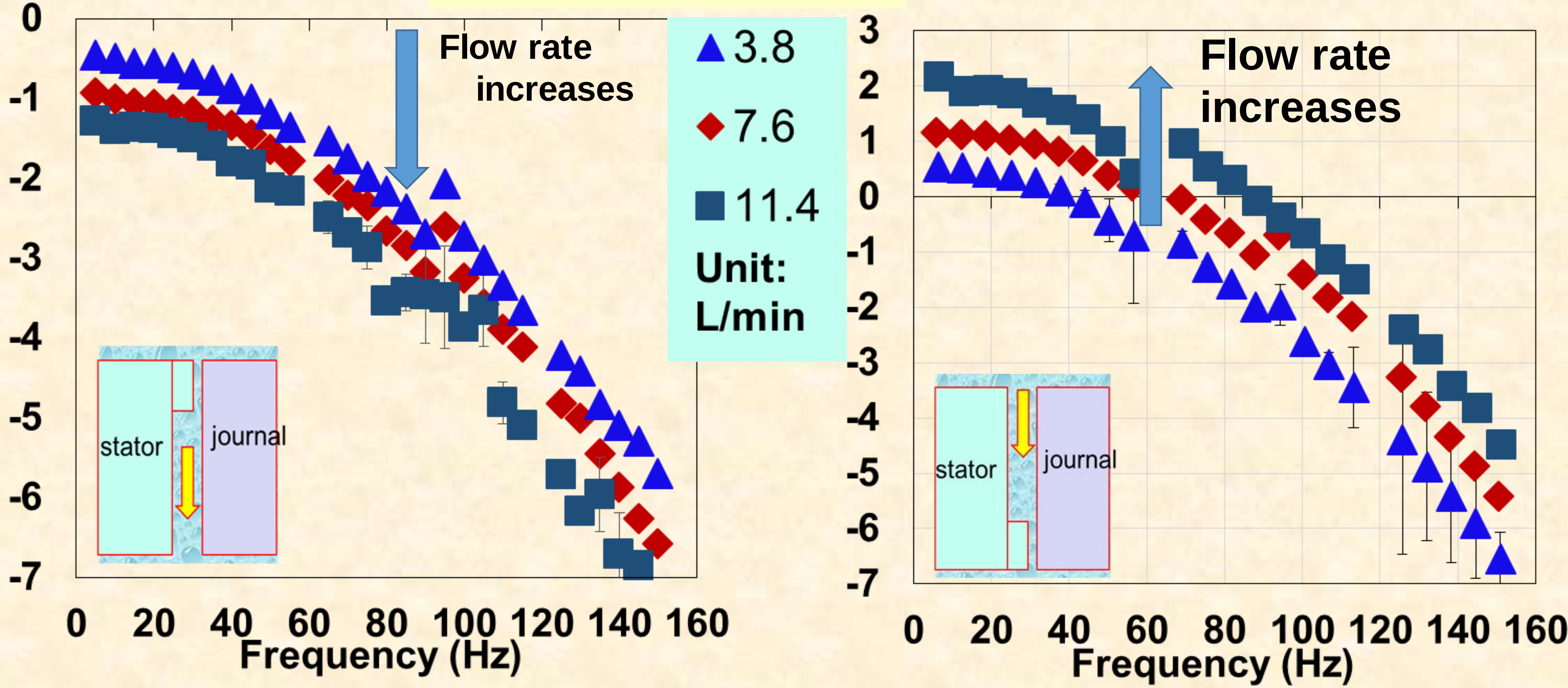


Nishimura, H., et al., **2016,** “Sub- and Super-Synchronous Self-Excited Vibrations of a Columnar Rotor due to Axial Clearance Flow,” 28th IAHR Symposium on Hydraulic Machinery and Systems, Grenoble, France, July 4-8.

Dynamic stiffness for step clearance seals

Re(H)=K-$\omega^2 M$ (MN/m)

0 rpm. Liquid only

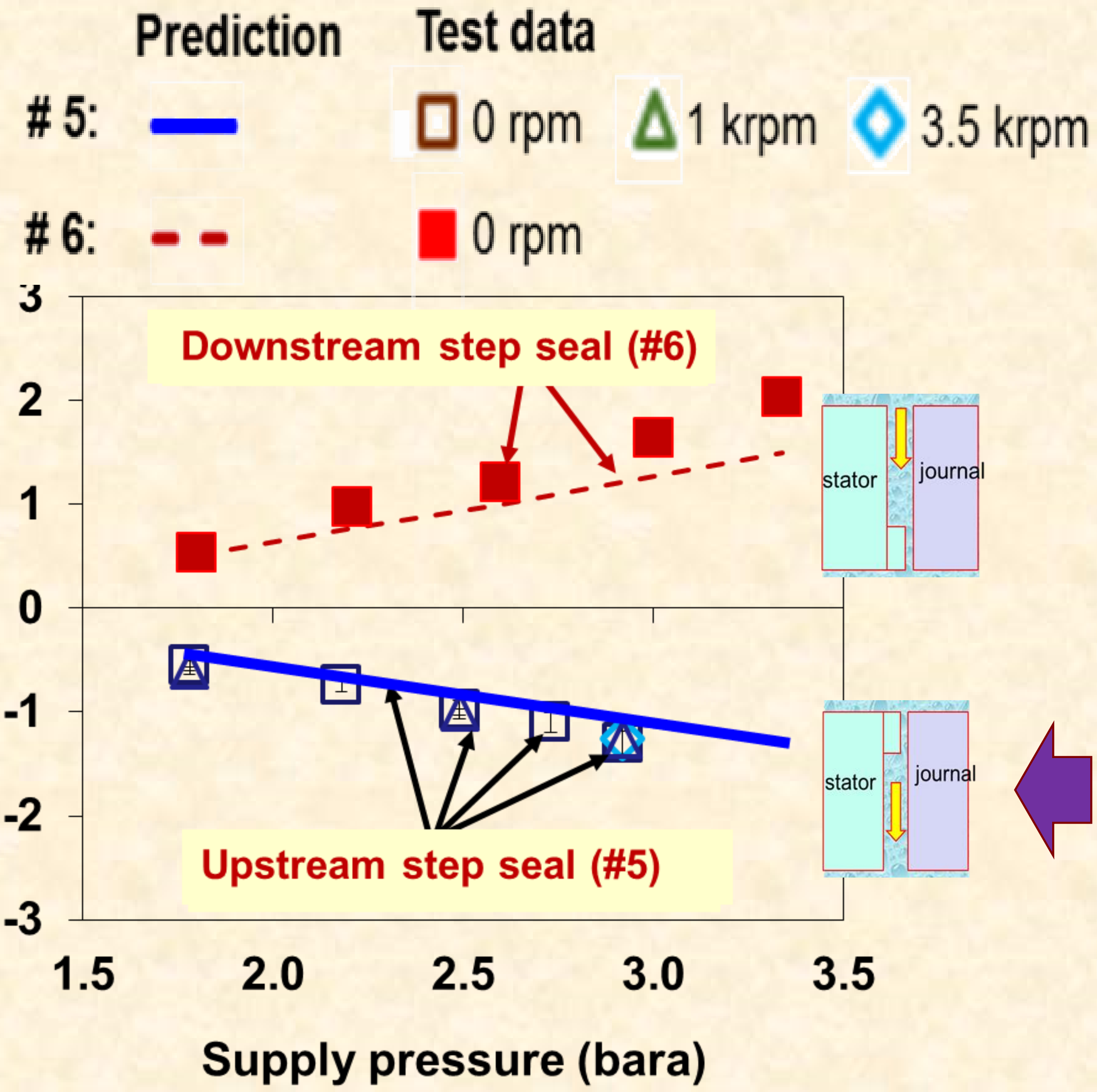


K< 0, C >0

K > 0, C >0

|K| grows with flow rate (supply pressure)

Direct stiffness for step clearance seals *K*(MN/m)

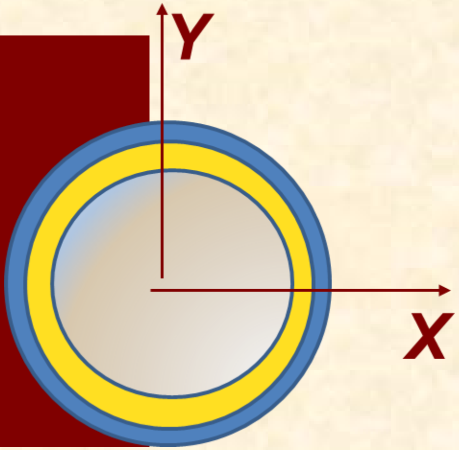


0 -3.5 krpm. Liquid only

|*K*| ~ supply pressure (flow), not a function of shaft speed.

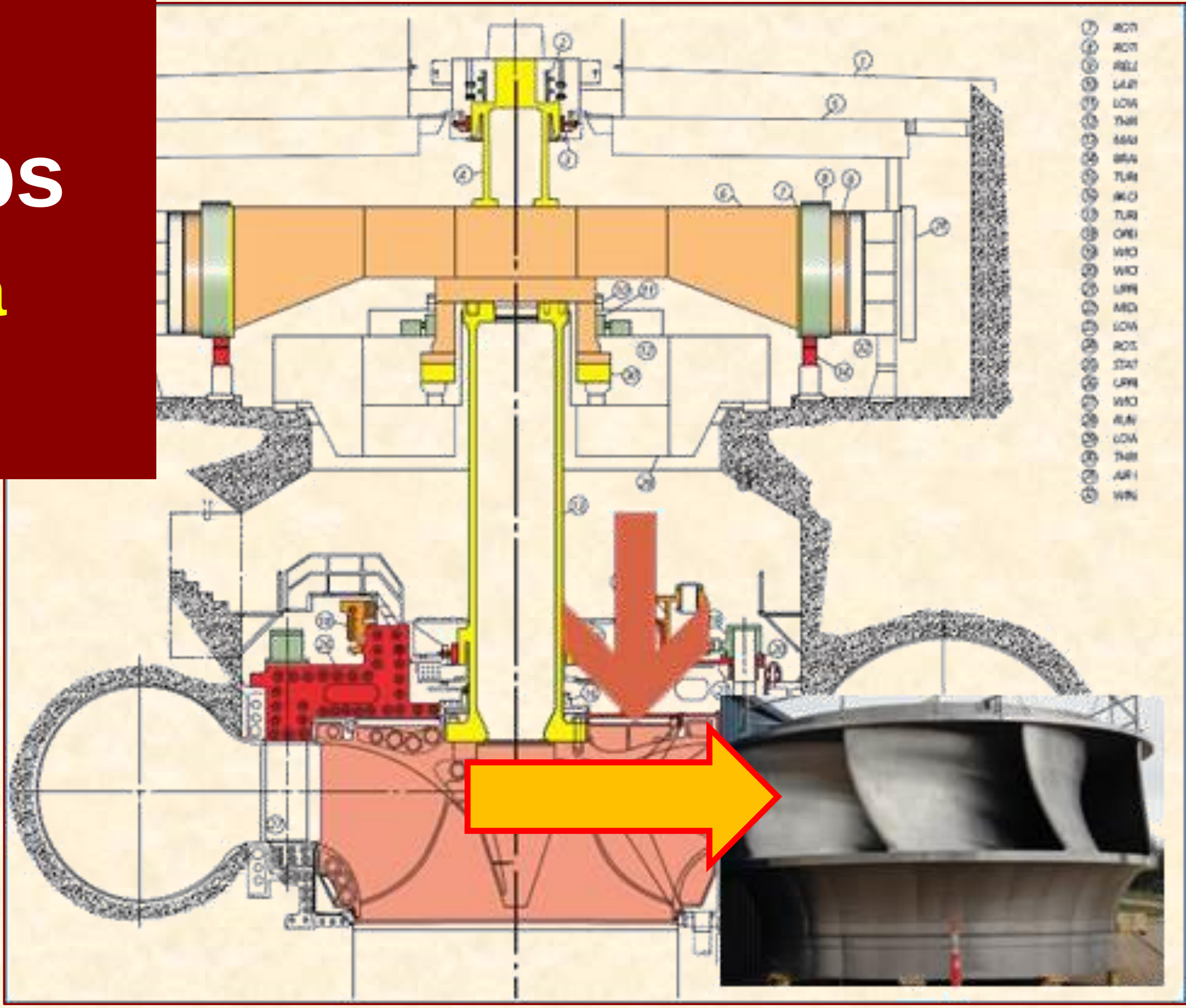
→ negative stiffness for upstream narrow clearance step seal may cause a static instability.

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 amplitude of motions.

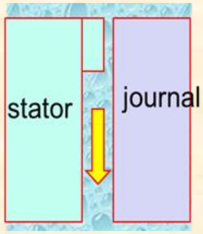


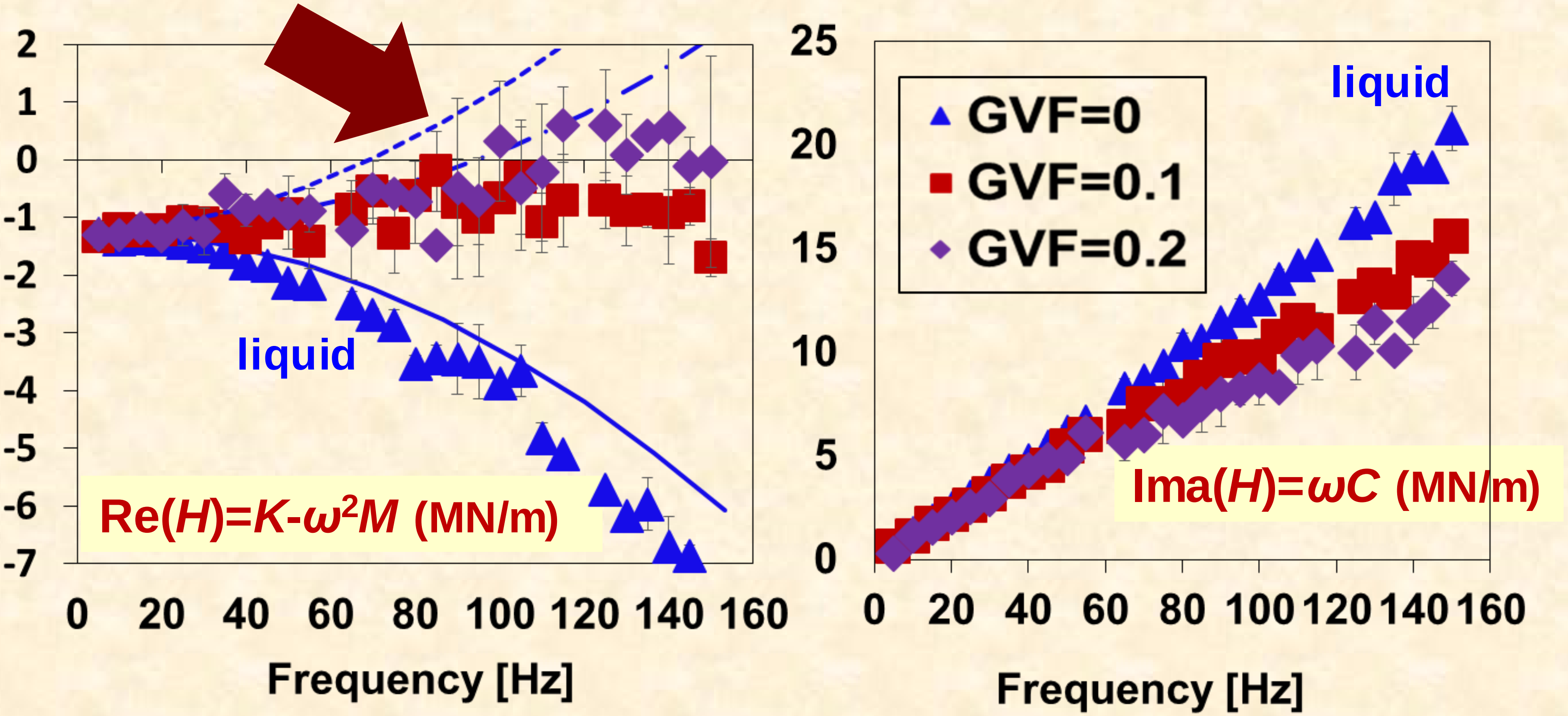
Air injection increases *K* (upstream step seal)!

- **All liquid seal, *K* < 0 and reduces quickly with frequency.**
- **Air injection reduces damping but increases dynamic stiffness → *K* >0.**

Symbols: test results

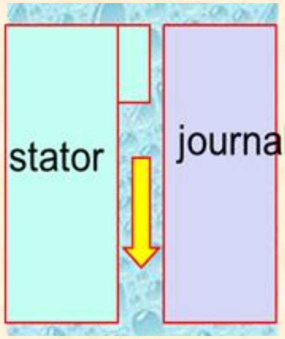
Q_l = 11.4 L/min, P_s=2.9 bara **0 rpm**





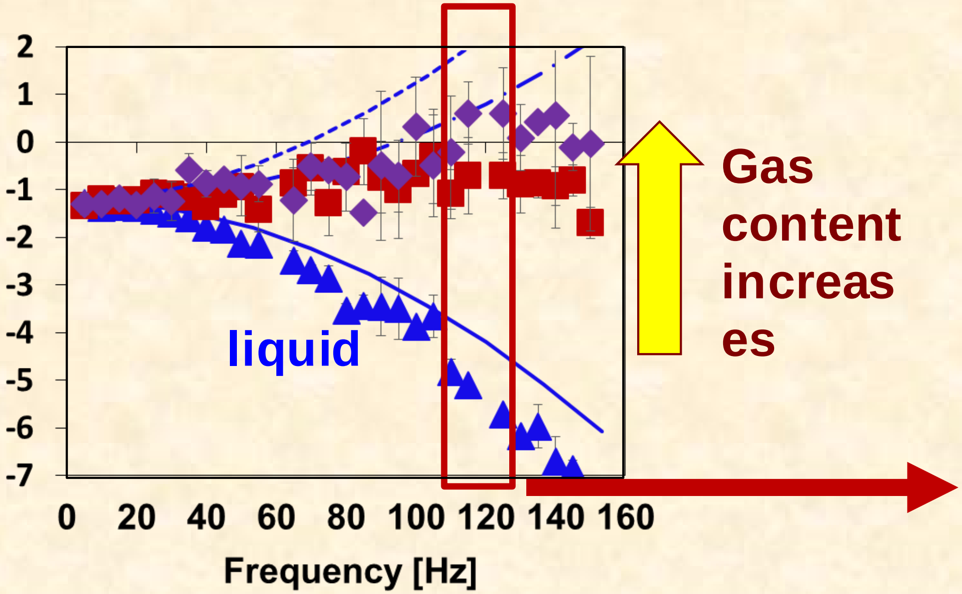
Air injection turns a negative stiffness into a positive centering one ($K > 0$).

(upstream step seal)



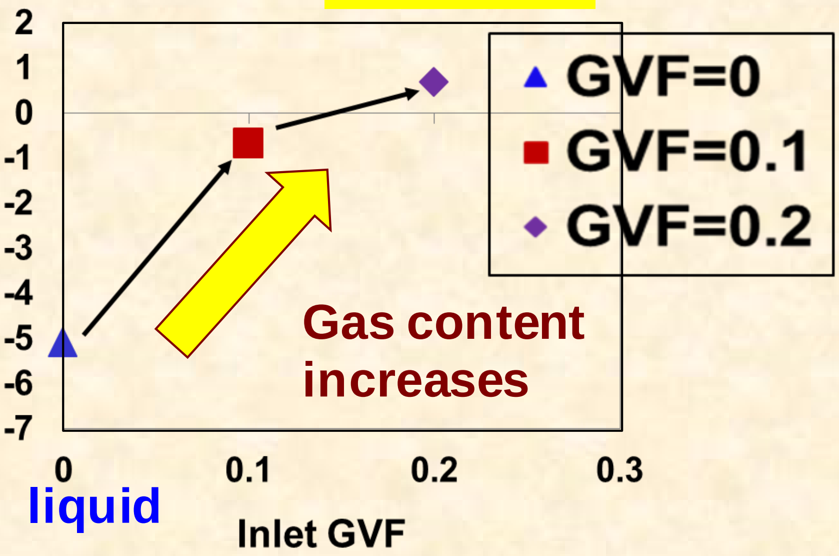
0 rpm

Q_l = 11.4 L/min,
P_s=2.9 bara



Gas content increases

At 115 Hz

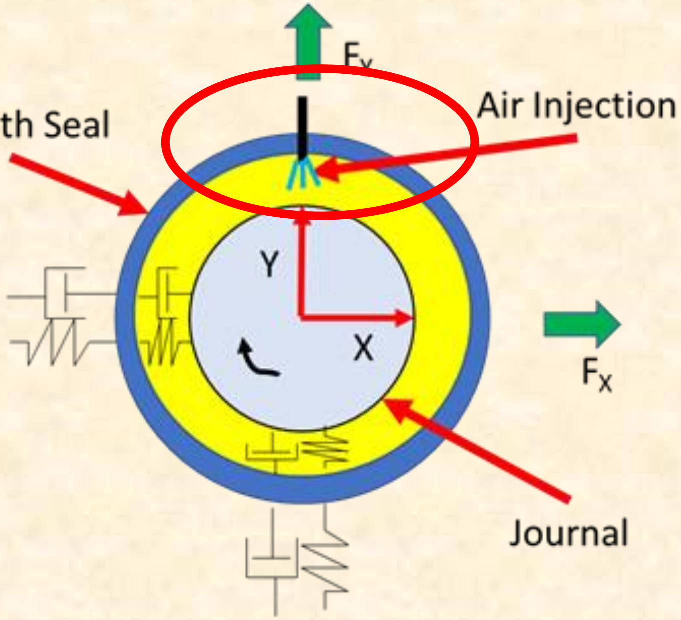


Gas content increases

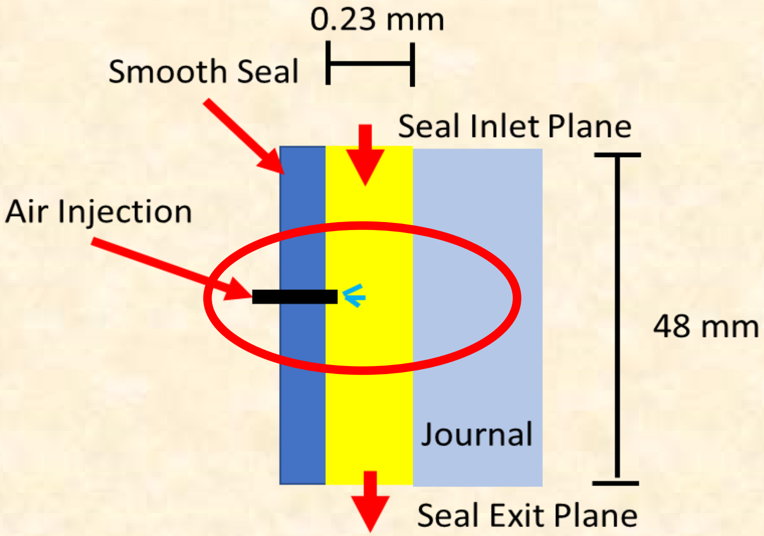
$K > 0$ brings static stability to vertical turbines.

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

Bubbles injection to increase stiffness



Top View

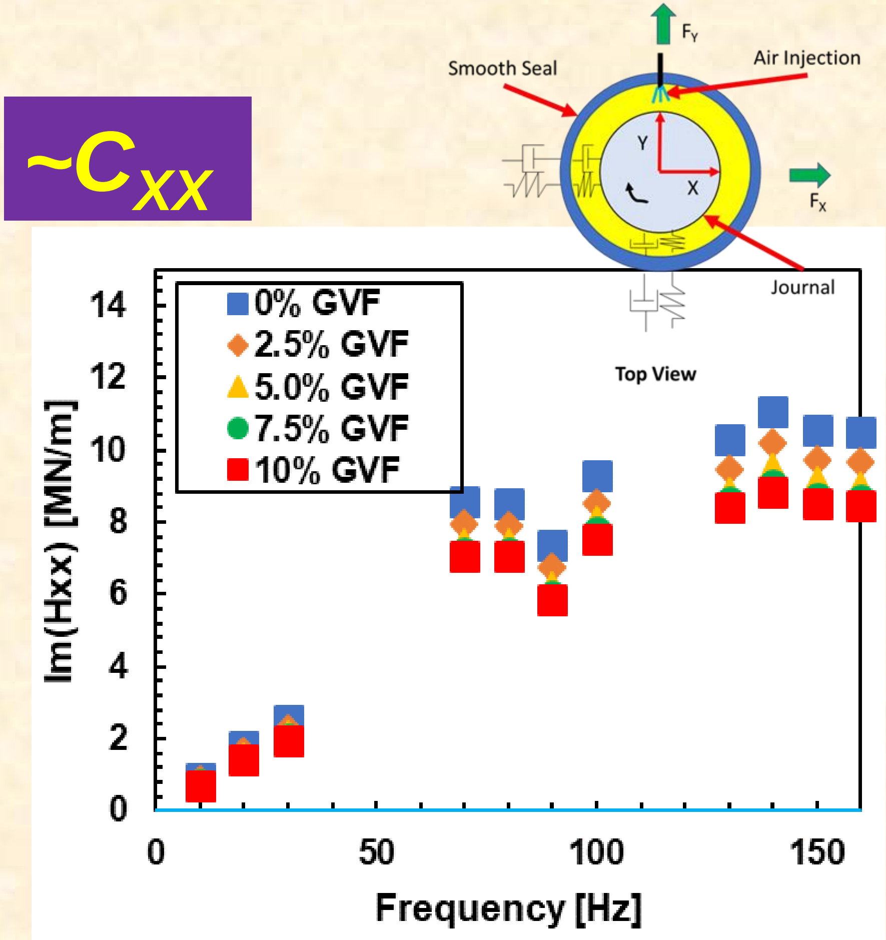


Side View

Injection of bubbles reduces damping

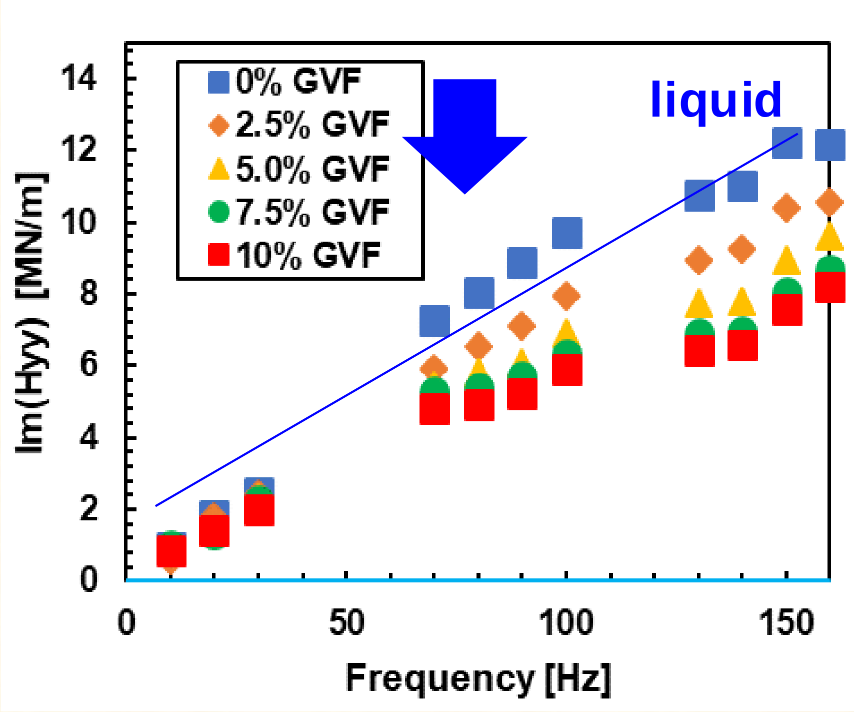
0 rpm,
Ps/Pa=2.5

~C_{XX}



~C_{YY}

Y- direction of bubbles injection

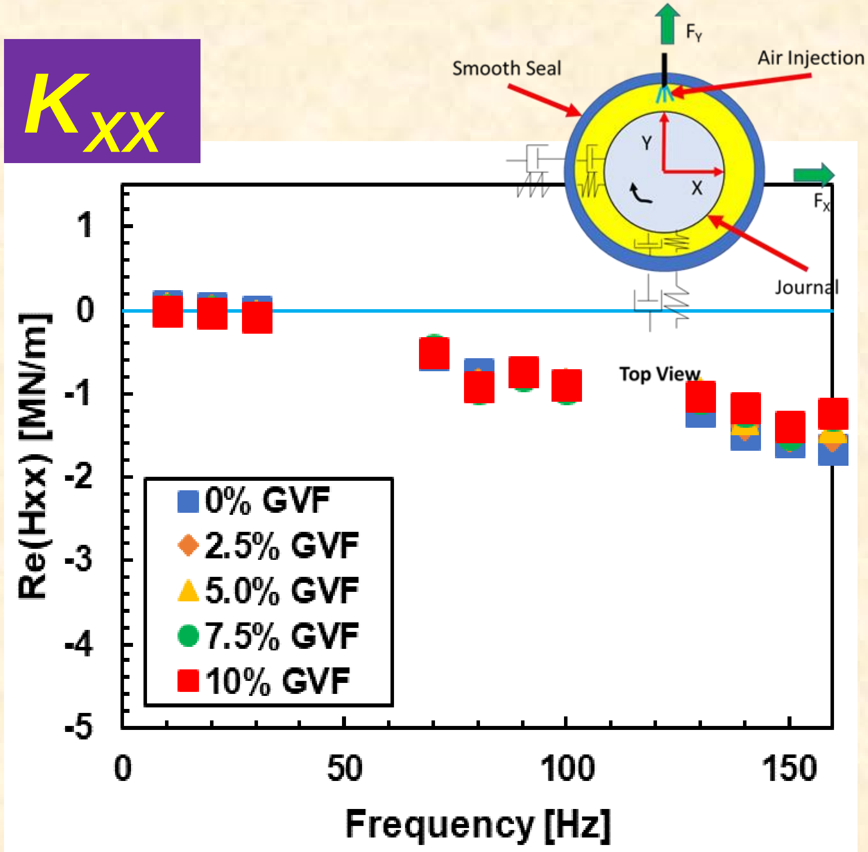


$C_{XX} > C_{YY} > 0$ as GVF increases

Injection of bubbles increases *K*

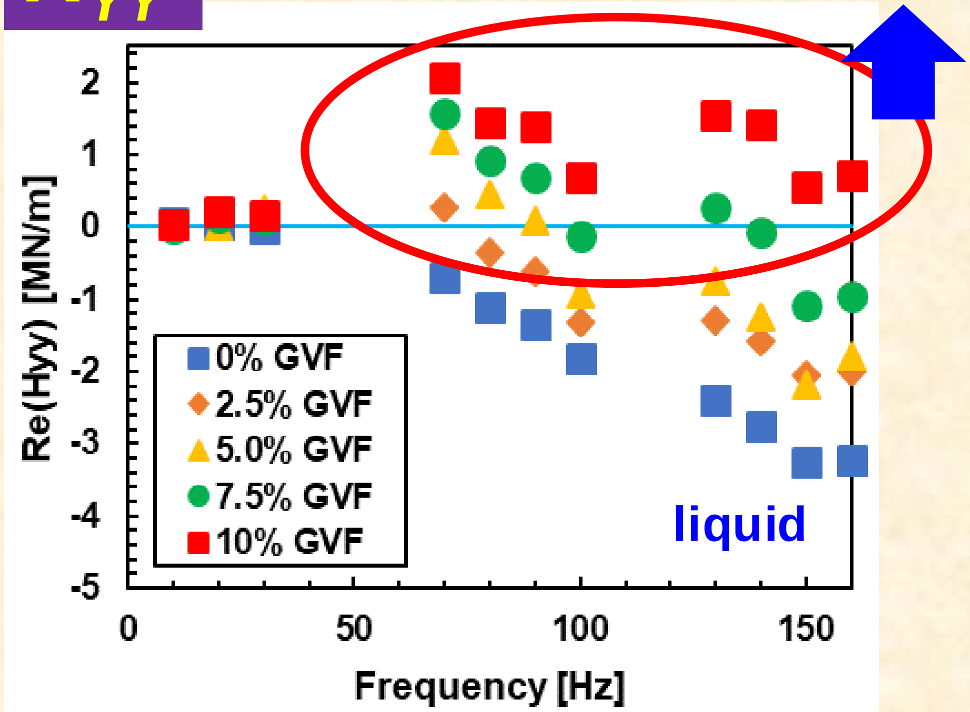
0 rpm, *Ps/Pa*=2.5

K_{XX}



K_{YY}

Y- direction of bubbles injection



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

Stiffness asymmetry promotes rotor stability!

Conclusion

ON MULTIPLE PHASE PUMP SEALS: LEAKAGE AND GAS INJECTION TO CONTROL SEAL CENTERING STIFFNESS

Conclusion

(a) Three wave seal leaks more than plain seal. The downstream step clearance seal leaks the least.

(b) Mass flow rate and drag torque drop continuously with an increase in gas volume fraction (GVF).

(c) Force coefficients are frequency dependent for operation with gas/oil mixtures.

(d) Three wave seal shows largest direct stiffness *K*.

(e) Cross stiffness *k* decreases with both frequency and GVF.

(f) Damping *C* decreases with GVF → $C \sim C_l$ (1-GVF)

(g) C_{eff} increases with frequency and drops with GVF. Cross over frequency is ~ ½ X.

(h) Air injection produces seal stiffness hardening & asymmetry → increases stability (good for vertical systems).

Acknowledgments

Thanks to TAMU Turbomachinery Research Consortium

Send questions (?) to lsanandres@tamu.edu