COMPUTATIONAL FLUID DYNAMICS (CFD) OF AN AIR IN OIL (*WET*) ANNULAR SEAL

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* San Andrés, L., Yang, J., and Lu, X., 2018, "on the Leakage, Drag Power and Rotordynamic Force Coefficients of an Air in Oil (Wet) Annular Seal, a CFD Analysis Anchored to Test Data," ASME J. Eng. Gas Turbine Power.

Computational Resource

Provided by Texas A&M University
High Performance Research Computing

CFD of a two-phase flow in an annular seal*

HPRC Cluster	Ada
Software	ANSYS® FLUENT
A typical job → 3D unsteady state CFD case (mesh node count 1.7×10^6)	
# of cores	40
Memory	2000 per process/CPU
Run time	16 CPU hour per period of rotor whirl)

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A need: subsea pumping & compression

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.

O&G price will increase! subsea production facilities will be more common (North Sea & Brazil → Gulf of Mexico → Artic) as extreme engineering will enable five year or longer reliability.

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A *Wet* Smooth Surface Annular Seal

Geometry

Rotor diameter $D = 2R$	127 mm
Seal length L	46 mm
Clearance C_r	0.203 mm

Operation at

Inlet pressure	P_S = 2.5 bar(a)
Outlet pressure	P_a = 1 bar(a)
Shaft speed	3.5 krpm (23.3 m/s)
Inlet pre-swirl	0

Test Data*
- For inlet GVF = 0.9, leakage decreases by 25% and drag power reduces by 85% when compared to flow/power for oil seal.
- *Wet* seal force coefficients are frequency dependent and GVF dependent.
- Test seal shows self-excited acoustic resonance at ~12 Hz (SSV).

* San Andrés, L., and Lu, X., 2018, "Leakage, Drag Power and Rotordynamic Force Coefficients of an Air in Oil (Wet) Annular Seal," ASME J. Eng. Gas Turbine Power, 140.

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The aim of the research

To complement experimental work by revealing flow field structures in multiple-phase flow seals through Computational Fluid Dynamics (CFD) and to validate/update engineering (BFM) predictive tools.

Contents

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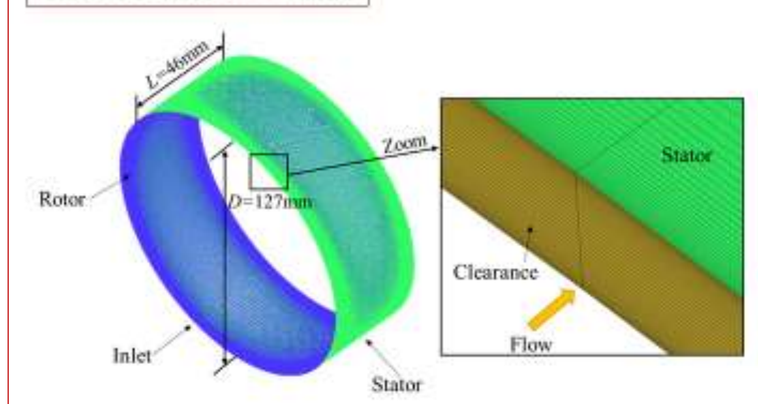
CFD Approach and Mesh

Ansys-Fluent® hosted by TAMU HPRC

Multi-phase model	Eulerian-Eulerian model
Drag model	Schiller and Naumann
Air in mixture	ideal gas
Bubble size	1×10^{-6} m

Laminar flow for oil and mixtures

3D mesh (node # 1.7 Million)



Upstream & downstream plenum sections not needed in a laminar flow seal.

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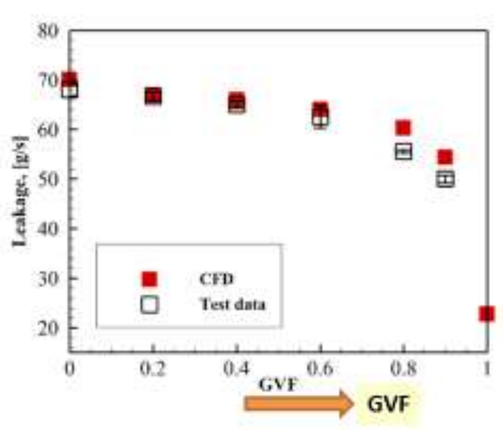
Seal Leakage: CFD vs Tests vs BFM

ΔP = 1.5 bar, rotor speed 3.5 krpm, inlet GVF = 0~1

Fluid	GMF	Test [g/s]	CFD [g/s]
Pure oil	-	68.0	70.1
Air in oil			
GVF = 0.2	0.0008	66.7	66.9
GVF = 0.4	0.002	65.0	66.0
GVF = 0.6	0.005	62.6	64.1
GVF = 0.8	0.01	55.6	60.3
GVF = 0.9	0.03	50.0	54.4
Pure air	-	-	23.4

GMF: mass volume fraction.



- CFD leakage reproduce test data [4].
- GMF is small (~0.03) even for GVF = 0.9.

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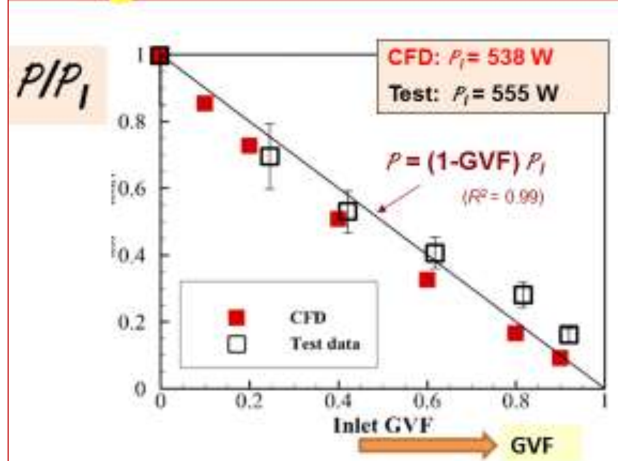
Drag Power Loss P: CFD vs Test



$P = T_D \cdot \Omega$

T_D: Drag torque.
Ω: rotor speed.

P_l: power for oil seal.
P: power bubbly seal.

- Drag power decreases linearly with gas content → P ~ (1-GVF) P_l.
- CFD drag power agrees with test data.

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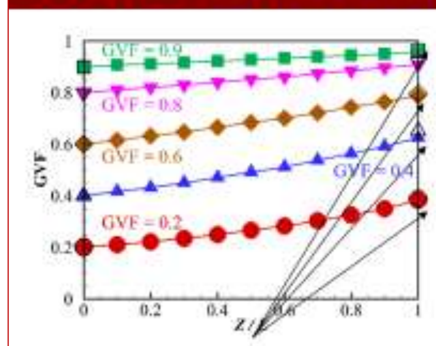
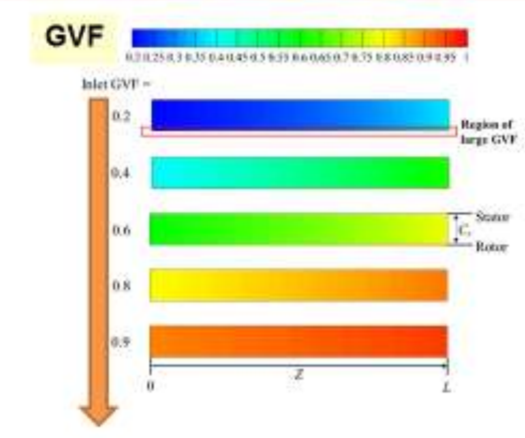
2D CFD: GVF vs axial length & across clearance





At exit plane (z=L):

$$GVF_{out} = \frac{P_s \cdot GVF_{in}}{P_s \cdot GVF_{in} + P_a \cdot (1 - GVF_{in})}$$

- GVF increases as pressure drops. CFD GVF verifies simple formula.
- For small GVF: thin layer (~ 5% C_r) with GVF > 0.8 near rotor surface → (denser) oil is thrown away from rotor surface (Ro# < 1).

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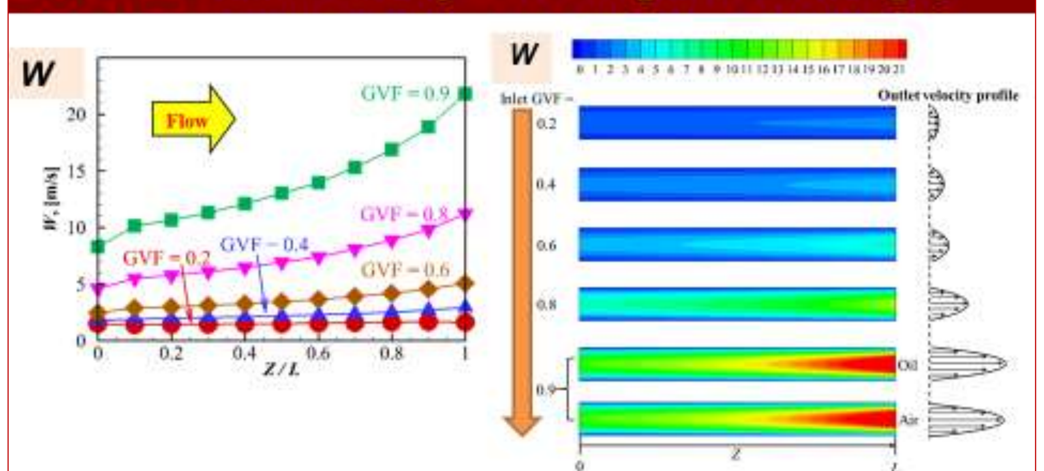
2D CFD: axial velocity *W* vs length & across gap



- Same speed for oil & air → homogeneous flow!
- Increase of GVF → larger axial speed, fastest at the seal exit (since mixture density decreases).

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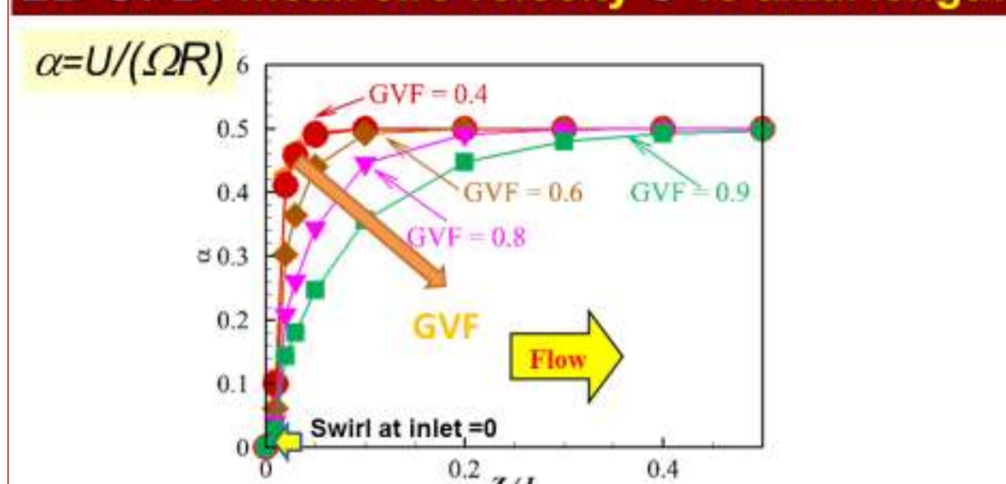
2D CFD: mean circ velocity *U* vs axial length

$\alpha = U/(\Omega R)$



- For low GVF, (mainly) oil quickly reaches ½ surface speed (α~0.5) near seal inlet ← Slow Stokes flow.
- As air content increases, GVF → 1, circumferential speed *U* takes longer to develop towards α~0.5 ← Axial flow dominates.

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3D CFD for Oil Seal (GVF=0)

Force Coefficients

	Test	CFD
K [N/m]	-	-0.25
k [MN/m]	3.8	4.6
C [kN-s/m]	20	25.4
M [kg]	5.5	7.0
WFR=k/CΩ	0.52	0.49

Rotor diameter D	127 mm
Seal length L	46 mm
Clearance C_r	0.203 mm

- CFD validate analytical solution.
- The gap between CFD and test data maybe due to inaccurate estimation of clearance (~8% higher will deliver test *C*, *k*).

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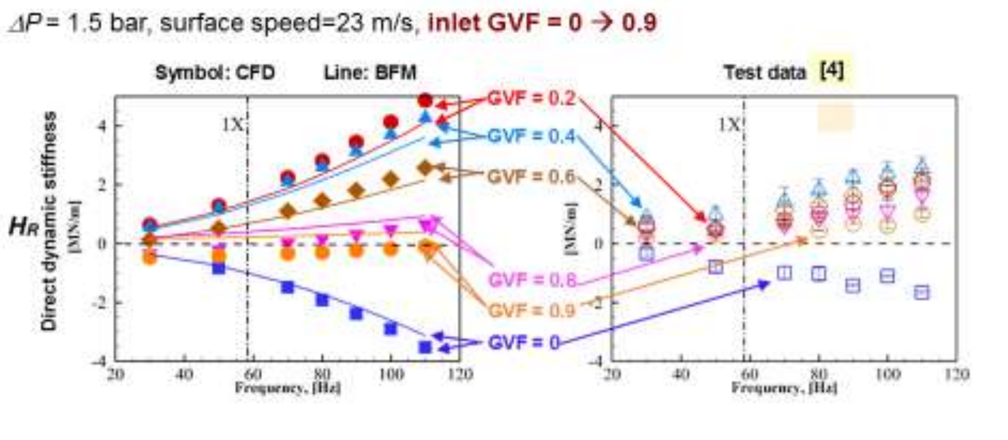
Direct Dynamic Stiffness for *Wet* Seal

ΔP = 1.5 bar, surface speed=23 m/s, inlet GVF = 0 → 0.9



- Centering stiffness > 0 for GVF>0.2 → a strong hardening effect.
- Test stiffness shows peak magnitude at GVF = 0.4, different from CFD and BFM predictions (max. at GVF = 0.2).

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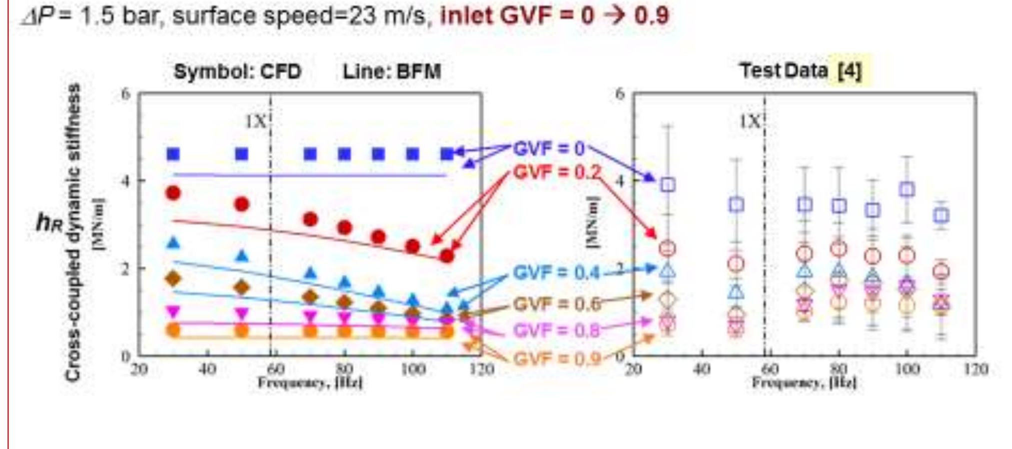
Cross-coupled Dynamic Stiffness for *Wet* Seal

ΔP = 1.5 bar, surface speed=23 m/s, inlet GVF = 0 → 0.9



- BFM prediction a little lower than CFD result.
- Test stiffness decreases with frequency (ω/Ω <1) and then increases; not seen in CFD and BFM predictions.

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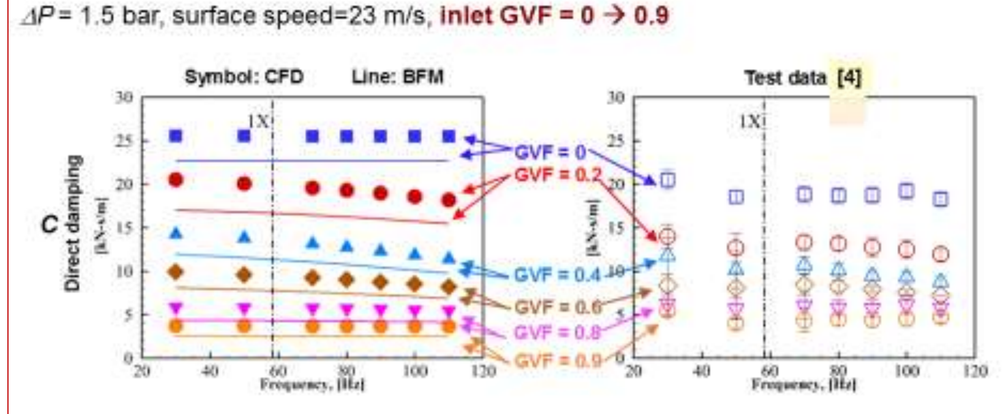
Direct Damping for *Wet* Seal

ΔP = 1.5 bar, surface speed=23 m/s, inlet GVF = 0 → 0.9



- BFM < CFD prediction.
- As GFV increases, direct damping decreases steadily.
- CFD damping matches well test data for inlet GVF > 0.6.

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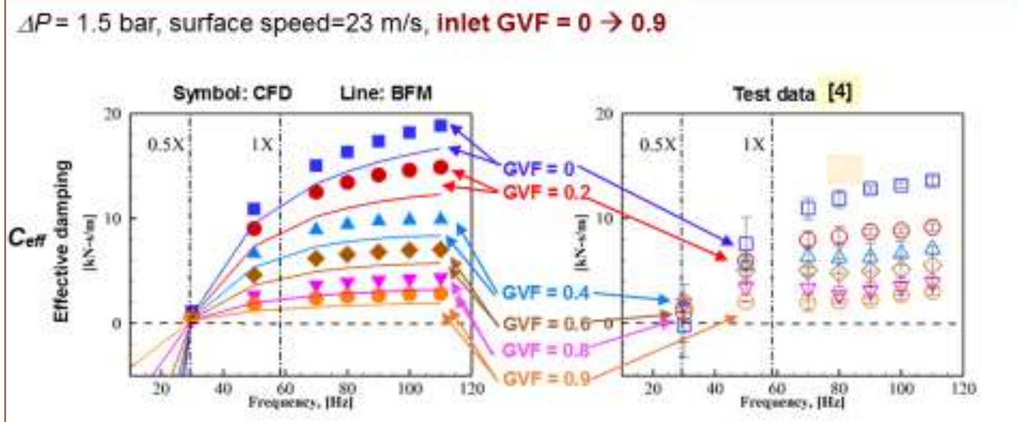
Effective Damping for Seal $C_{eff} = (C - h_R/\omega)$

ΔP = 1.5 bar, surface speed=23 m/s, inlet GVF = 0 → 0.9



- Effective damping decreases linearly with an increase in gas content.
- Cross-over frequency is ~ ½ X for oil seal & decreases as GVF grows.

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Conclusion

- CFD predictions (leakage, power loss) agree with test data, and also produce high fidelity flow field variables, including pressure, speeds, and GVF.
- Operation with a low GVF (< 0.4) produces a significant hardening effect which makes positive the direct stiffness. Test data shows same rapid stiffness increase as GVF→ 0.2.
- Stiffness hardening effect is due to the dramatic reduction in sound speed brought by a small amount of gas (fluid becomes more compressible).
- The combination of test results and CFD analyses furthers the engineering of seals for *wet* gas compressors and bubbly liquids in multiple phase pumps.

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Acknowledgments

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

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