Continuation Proposal to the TAMU Turbomachinery Research Consortium

LEAKAGE AND FORCE COEFFICIENTS IN A WET ANNULAR SEAL WITH A WAVY SURFACE

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SIGNIFICANCE

In the oil and gas industry, multiphase pumps may handle oil flow with large gas volume fraction (GVF) [1] and wet gas compressors must work with up to 5% in liquid volume fraction (LVF) [2]. Designing and constructing reliable oil boosting and wet gas compression systems to handle gas-oil mixtures will eliminate L/G separators, saving up to 30% in capital investment in a typical L/G separation station [1].

Centrifugal compressors handling heavy density *wet* gases must account for the effect of damper (balance piston) seals shifting critical speeds [2]. The liquid content does affect the force coefficients in annular seals [3]. Vannini *et al.* in 2014 [2] report a severe 45% sub synchronous vibration (SSV) in a single stage centrifugal compressor operating with a *wet* gas (LVF=3%). Liquid trapped in the labyrinth seal caused the SSV, whose onset and persistence started with a LVF as low as 0.5%.Vertical pumps with plain cylindrical bore bushings are also prone to show SSV, the absence of a static load generating large cross-coupled stiffnesses [4].

SUMMARY OF WORK 2015-2016

Fig. 1 shows the TRC *wet* seal test rig comprised of a transparent seal cartridge supported atop by a steel pipe that is connected to a firm structure. Two rigid-mounted electromagnetic shakers (100 lb) excite the seal cartridge via long stingers. A mixture of air in ISO VG 10 oil flows through the pipe into the seal with diameter (D) = 127 mm, length (L) = 46 mm, and radial clearance (c) = 0.203 mm.

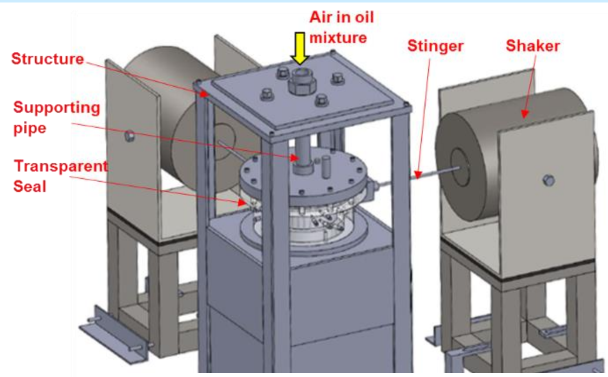

Fig 1. Side view of the seal test rig

In 2015-2016, further measurements with mixtures were conducted with shaft speed to 3.5 krpm (surface speed ΩR = 23.3 m/s). TRC-SEAL-01-16 [3] details the measurement procedure, data analysis and findings. For tests with a supply / discharge pressures P_S / P_a = 2.5, Fig. 2 shows the normalized mass flow rate of a gas in oil mixture flowing through the seal vs. gas volume fraction (GVF) for 3 rotor speeds and a stationary journal. Note the flow with either a pure lubricant (GVF=0) or a mixture with GVF→0.92 is not affected by shaft speed as the flow is laminar for most conditions. The mass flow rate decreases continuously as the inlet GVF increases, 0 → 0.92. The large difference in densities between the liquid (oil) and gas (air) components determines that the liquid content in the mixture be rather large even when the GVF is high. Predictions agree with the measurements.

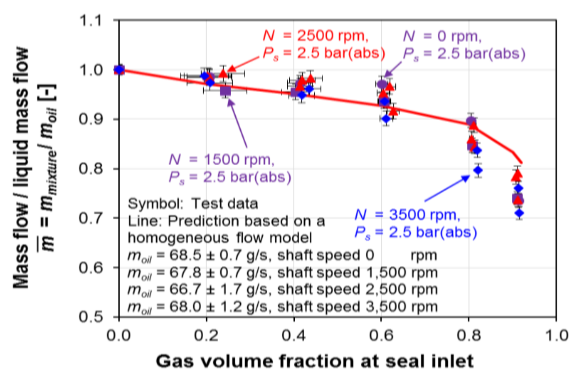

Fig 2. Mass flow rate vs. inlet gas volume fraction. Supply / discharge pressure P_S / P_a = 2.5. Inlet temperature T_{in} = 33 °C ~35 °C. Journal speed 3,500 rpm (ΩR = 23.3 m/s).

(Single frequency) dynamic loads exerted on the test seal produced motions from which dynamic force coefficients were determined. Specific operating conditions are pressure supply / discharge ratio = 2.5 and delivery of oil (only) and also an air in oil mixture with inlet GVF = 0.2 → 0.9.

Fig. 3 displays the test damping coefficient (C_{XX}) vs. excitation frequency for operation with shaft speed = 0, 2.5 krpm and 3.5k rpm. Shaft speed decreases slightly the damping coefficient which shows a more dramatic reduction as the frequency increases. Most notably, the gas content in the mixture affects dramatically the generation of damping. With shaft speed = 2.5 krpm and pressure supply/discharge = 2.5, Fig. 4 depicts the seal cross coupled stiffnesses (K_{XY}, K_{YX}) drop with both frequency and gas content in the mixture, GVF = 0.0 → 0.9. Shown in Fig. 5, at 2.5 krpm shaft speed, the seal whirl frequency ratio (WFR) drops from 0.5 for a pure oil condition to 0.44 for a mixture with GVF = 0.4.

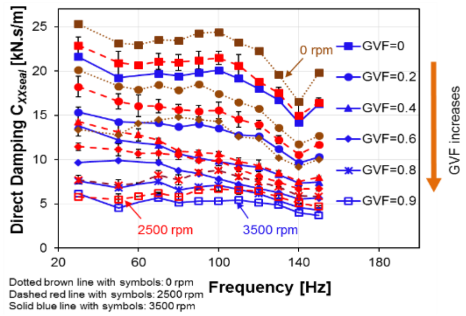

Fig 3. Seal direct damping (C_{XX}) vs. frequency (ω). Journal speed: 0 rpm to 3.5 krpm.

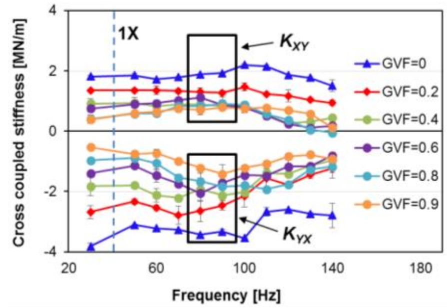

Fig 4. Seal cross coupled stiffnesses vs. frequency (ω). Journal speed 2.5 krpm.

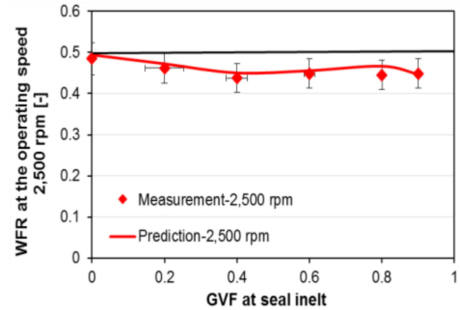

Fig 5. Whirl frequency ratio (WFR) vs. inlet gas volume fraction. Journal speed 2.5 krpm.

Supply / discharge pressure P_S / P_a = 2.5. Inlet temperature T_{in} = 33 °C ~35 °C.

PROPOSED WORK 2016-2017

Leader *et al.* [4] successfully reduced SSV in a vertical pump by replacing plain cylindrical bushings with a three-lobe profile design. Earlier in 1994, Dimofte [5] introduced wavy-surface journal bearings for either gas or liquid lubrication with benefits of stability due to a centering stiffness generated by the mechanical preload. Either lobular or waved surface configurations to replace cylindrical bushings are simple and cost-effective yet not tested with a mixture of liquid in gas prevalent under *wet* conditions. A seal lobular shape, elliptical in particular, can be attained in practice by simply crushing the bushings when installed in a pump.

During year IV, the project will continue to quantify the effect of a *wet* mixture on seal leakage and force coefficients. Fig. 6 depicts the *new* seal as a three-lobe wavy configuration with a ratio of clearances, c_{min}/c_{max} = 0.162 mm/0.244 mm = 0.66. The average clearance c_{ave} = 0.203 mm is that of the current installed seal (L/D = 0.36). Seal operation will range from a stationary shaft to journal rotation at 4 krpm (surface speed 27 m/s) and with air supplied at 5 bar (abs) and less. Air in oil mixtures will have an increasing GVF (0 to 100%). Dynamic loads exerted of the seal cartridge with a single frequency (10 Hz - 150 Hz > 2X) will serve to identify the system complex stiffnesses. Frequency dependent force coefficients will follow as a function of the GVF. Specific tasks include:

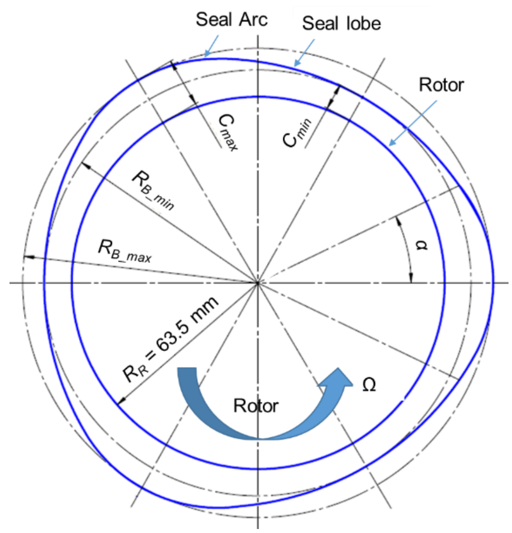

Fig 6. Wavy seal schematic view: D = 127 mm, L = 46 mm, c_{max} = 0.244 mm, c_{min} = 0.163 mm, c_{ave} = 0.203 mm.

a) Design and manufacture of new seal cartridge, install new cartridge, assemble the test rig and trouble shooting.
b) Measure seal leakage at various conditions of GVF, pressure supply, and shaft speed.
c) Conduct dynamic load tests and identify frequency dependent force coefficients.
d) Modify existing flow model in HSEALMIX© to include multiple-lobe or wave geometries.

BUDGET FROM TRC FOR 2016-2017

	Year IV
Support for graduate student (20 h/week) x $ 2,400 x 12 months	$ 28,800
Fringe benefits (2.7%) and medical insurance ($360) x 12 months	$ 5,098
Registration and travel to technical Conference	$ 1,000
Tuition three semesters ($ 400 credit hour x 24 ch/year)	$ 8,712
Supplies: Seal manufacturing $1,350	$ 1,370
Total Cost	**$ 44,980**

REFERENCES

[1] Gong, H., Falcone, G., *et al.*, 2012, "Comparison of Multiphase Pumping Technologies for Subsea and Downhole Applications," Oil and Gas Facilities, pp. 36-46.

[2] Vannini, G., Bertoneri, M., Del Vescovo, G. and Wilcox, M., 2014, "Centrifugal Compressor Rotordynamics in Wet Gas Conditions," Proceedings of the 43rd Turbomachinery & 30th Pump Users Symposia, Houston, TX. September 23-25.

[3] San Andrés, L., Lu, X., 2016, "Leakage and Rotordynamic Force Coefficients in an (Air in Oil) Wet Seal: Influence of Shaft Speed," **TRC-SEAL-01-16**, Annual Progress Report to the Turbomachinery Research Consortium, Texas A&M University, May.
[4] Leader, M.E., Conner, K.J., Lucas, J.D., 2010, "Evaluating and Correcting Subsynchronous Vibration in Vertical Pumps," 26^{th} International Pump Users Symposium, September 2010, Texas A&M University, Houston, TX, USA.
[5] Dimofte, F., 1994, "Wave Journal Bearing with Compressible Lubricant – Part I: The Wave Bearing Concept and a Comparison To The Plain Circular Bearing," Tribol. Trans., **38**(1), pp. 153–160,