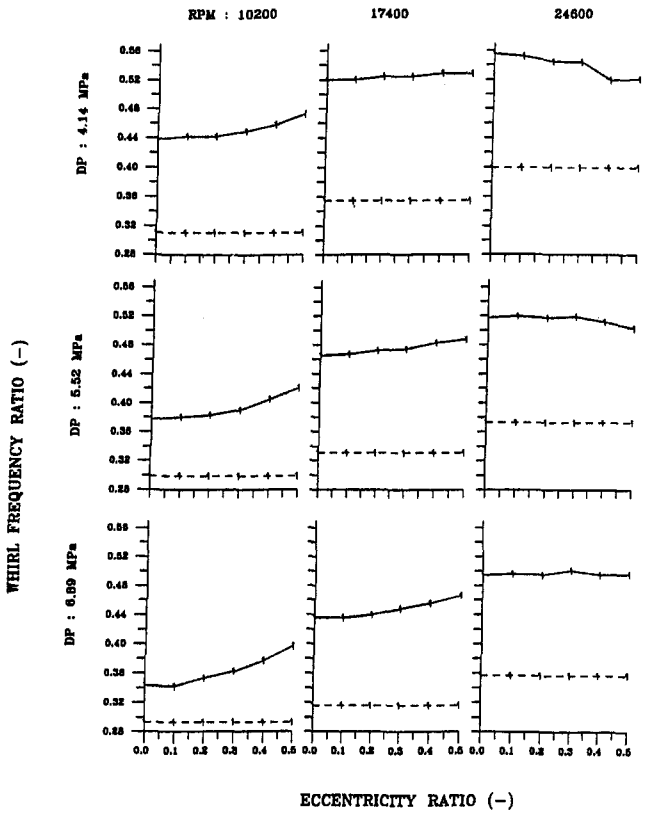

Fig. 7 Comparison between experimental and predictions for whirl frequency ratio: solid = experimental, dashed = theory

K_{xx}, $K_{yy} \approx$ 1 MN/m (10%)
K_{xy}, $K_{yx} \approx$ 2 MN/m (6%)
C_{xx}, $C_{yy} \approx$ 1 kNsec/m (5%)
C_{xy}, $C_{yx} \approx$ 1 kNsec/m (10%)
M_{xx}, $M_{yy} \approx$ 1.5 Kg (50%)

Hence, uncertainties are reasonable for direct stiffness, cross-coupled stiffness, cross-coupled damping, and direct damping, and worse for inertia coefficients.

Conclusions

An annular smooth seal with $L/D = 0.45$ and $C/R = 0.0029$ has been tested on a wide range of static eccentricity ratios (0 to 0.5), running speeds (10,200 to 24,600 rpm), and pressure drops (4.14 to 6.89 MPA). High-speed data for smooth seals running at eccentric positions have been generated.

Leakage flowrate increases slightly with increasing eccentricity ratio, increases with increasing pressure drop, and decreases with increasing running speed. Theoretical predictions are accurate over the whole range of static eccentricity ratio, running speed and pressure drop.

Rotordynamic coefficients are more sensitive to changes in static eccentricity than theoretically predicted, especially cross-coupled stiffness and direct damping. Direct stiffness is slightly under predicted by analyses based on bulk-flow theory, but for the first time, theoretical predictions for direct inertia agree well with measurements.

The "normal" rotordynamic-coefficient model for annular seals is

$$-\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = \begin{bmatrix} K & k \\ -k & K \end{bmatrix} \begin{Bmatrix} \Delta x \\ \Delta y \end{Bmatrix} + \begin{bmatrix} C & c \\ -c & C \end{bmatrix} \begin{Bmatrix} \Delta \dot{x} \\ \Delta \dot{y} \end{Bmatrix} + M \begin{Bmatrix} \Delta \ddot{x} \\ \Delta \ddot{y} \end{Bmatrix} \quad (3)$$

This eccentricity-independent model has been assumed to hold out to eccentricity ratios of 0.5. The test results presented here show that the rotordynamic coefficients are, in fact, significantly more sensitive to eccentricity than predicted. Hence, for some cases, seals may need to be treated more like hydrodynamic bearings with coefficients which vary with the static eccentricity position.

References

Childs, D. W., and Hale, R. K., 1993, "A Test Apparatus and Facility to Identify the Rotordynamic Coefficients of High-Speed Hydrostatic Bearings," ASME JOURNAL OF TRIBOLOGY, Vol. 48.

Childs, D. W., 1993, *Turbomachinery Rotordynamics: Phenomena, Modeling, and Analysis,* Wiley Interscience, New York.

Falco, M., Mimmi, G., and Marenco, G., 1986, "Effects of Seals on Rotor Dynamics," *Proceedings of the IFToMM International Conference on Rotordynamics,* Tokyo, Japan, pp. 655–662.

Graf, K., and Staubli, T., 1990, "Comparison of Two Identification Models for Rotordynamic Coefficients of Labyrinth Seals," *Interfluid,* First International Congress on Fluid Handling Systems, Essen, Germany.

Kanki, H., and Kawakami, T., 1984, "Experimental Study on the Dynamic Characteristics of Pump Annular Seals," *Vibrations in Rotating Machinery, Proceedings of the Third IMechE International Conference on Vibrations in Rotating Machinery,* York, England, pp. 159–166.

Lindsey, W. T., and Childs, D. W., 1995, "The Effects of Converging and Diverging Axial Taper on the Rotordynamic Coefficients of Liquid Annular Pressure Seals: Theory Versus Experiment," accepted for the 1995 ASME Vibration Conference, Sept. 17–21, Boston, MA.

Lund, J., 1965, "The Stability of an Elastic Rotor in Journal Bearings with Flexible, Damped Supports," ASME *Journal of Applied Mechanics,* pp. 911–920.

Marquette, O., 1995, "Experimental Versus Theoretical Comparison of the Static and Dynamic Characteristics of One Smooth and Two Grooved Liquid Annular Seals with L/D of 0.457," TL-SEAL-5-95, Turbomachinery Laboratory, Texas A&M University.

San Andres, L., 1991, "Analysis of Variable Fluid Properties, Turbulent Annular Seals," ASME JOURNAL OF TRIBOLOGY, pp. 694–702.

DISCUSSION

R. C. Hendricks[2]

This reviewer would like to thank the authors for an interesting paper and would appreciate the authors expanding upon the following questions:

First, some minor questions that the authors can readily dispense with.
What is the diameter of the rotor?
What does short L/D mean if long $L/D = 0.45$.
What is low inlet swirl?
What were the tolerances in respect to "stable within tolerances"?
What are low, medium, and high speeds?

A transformation can be made for the centered position to formulate a time independent problem, yet at the eccentric position a similar transformation would not work. What transformation was used or could the authors further explain how the eccentric case was handled with the bulk code?

The bulk codes are ill-equipped to handle recirculation effects. First, did the authors notice if recirculation might be effecting their results? Second, was there an attempt to compare the hydroseal and sciseal codes, for example, a full NS code to

[2] MS 301-5, NASA Lewis Research Center, Cleveland, OH 44135.

Copyright © 1997 by ASME

Downloaded From: http://tribology.asmedigitalcollection.asme.org/ on 09/16/2016 Terms of Use: http://www.asme.org/about-asme/terms-of-use

assess effects such as those of recirculation? I realize the expense of running Sciseal for the eccentric case, yet the centered position with the transformation should perform satisfactorily.

The data of Morrison show "rotation" of the pressure field. This rotation is nearly 270-degrees from the inlet to the exit of the seal and has a significant influence on the dynamics. Preliminary work with sciseal has illustrated reasonably good agreement with a low-Reynolds number model (although not too sensitive to the model) over a range of Taylor and Reynolds numbers. This pressure field rotation has also been shown by Hendricks for the case of static seals at high Reynolds numbers in tests with fluid hydrogen over a wide range of flow rates and thermophysical property conditions. It is also known that the pressure distributions and dynamic coefficients are influenced by swirl brakes. First, have you noticed a similar rotation of the pressure field, as seen by Morrison and others, in your experiments? Please explain! Second, how has such a pressure distribution influenced your calculation of stability coefficients? Please explain!

Authors' Closure

Taking Mr. Hendricks questions in order, our response follows:

(*a*) The diameter is provided in Table 1.

(*b*) Our comments concerning short L/D seals refers to the seals tested by Lindsey and Childs (1995) which had $L/D = 0.17$.

(*c*) The tester was designed to provide zero intentional preswirl.

(*d*) What were the tolerances in respect to, "stable within tolerances?" This statement in the paper refers to targets for steady-state operating conditions of ΔP, speed, and temperature. Taking these parameters in order, our tolerances are, respectively, within ±35 kPa, ±20 Rpm, and ±1°C.

(*e*) Low, medium, and high speeds refer to the 10200, 17400, and 24600 rpm values of Figs. 3 through 7.

(*f*) Concerning "transformations," there is no transformation introduced in the bulk-flow model (San Andres, 1991), since the bulk-flow model does not involve radial variation in any of the dependent variables.

(*g*) Concerning "recirculation," there is no evidence for (or against) recirculation in the test data. The authors have not made comparisons to SCISEAL. In regard to full N.S. (Navier-Stokes) solutions, Nordmann et al. (1986) made comparisons between bulk-flow solutions and N.S. codes and obtained basically identical results for small motion about a centered position. Given the differences between measurements and bulk-flow results demonstrated in this paper at higher eccentricities, comparisons to N.S. predictions by other researchers would certainly be welcome. However, the discrepancies between measurements and bulk-flow predictions might arise because of errors in modeling the upstream and downstream boundary conditions, rather than unmodeled radial variations in the dependent variables.

(*h*) Concerning Morrison et al. (1995) laser velocimeter data for flow in the annular seals, Morrison's operating conditions were drastically different from the present test cases. Specifically, the present test program has much higher ΔP's, much tighter clearances, higher-speeds, etc. Moreover, Morrison used a synchronously precessing seal. Given these differences, we would not necessarily expect similar phenomena. Finally, there are no measurements in our test program related to the phenomenon cited. Concerning calculations, the first-order perturbations in the bulk-flow solutions account for pressure perturbations due to small motion of the rotor relative to the stator, as is the case for the experimental measurements.

Additional References

Nordmann, R., Dietzen, F., Janson, W., Frei, A., and Florjancic, S., 1986, "Rotordynamic Coefficients and Leakage Flow for Smooth And Grooved Seals in Turbopumps," *Proceedings of the Second IFToMM International Conference on Rotordynamics,* Tokyo, Japan, pp. 619–627.

Morrison, G. L., Winslow, R. B., and Thames, H. D., III., 1995, "Phase Averaged Wall Shear Stress, Wall Pressure and Near Wall Velocity Field Measurements in a Whirling Annular Seal," Presented at the ASME TURBO EXPO '95—Land, Sea & Air Conference, June 5–8, 1995, Houston, Texas, ASME Paper No. 95-GT-101.

Downloaded From: http://tribology.asmedigitalcollection.asme.org/ on 09/16/2016 Terms of Use: http://www.asme.org/about-asme/terms-of-use