compressible flow annular seals. The analysis was based on an extension of the analysis of Nelson and Nguyen (1988). The analysis was used to produce a parametric study on the effect of tilt on the performance of straight and tapered annular seals. The seal configuration analyzed was typical of those found in cryogenic turbopump applications. The results presented support the following conclusions:

1) The seal leakage decreases as tilt increases.
2) All rotordynamic coefficients increase as tilt increases.
3) The whirl frequency ratio increases for substantial values of tilt.
4) For the seals investigated herein, the effects of tilt on straight and tapered seals was the same.
5) The effects of tilt are not detrimental in cases where the added mass is not significant and the degree of tilt is small.

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DISCUSSION

L. San Andres[1]

The paper presents the results of a numerical analysis on the effect of fixed rotor misalignment on the rotordynamic force/displacement coefficients of an incompressible liquid, turbulent flow annular pressure seal. The motion of the fluid in the thin film annular clearance is governed by fully developed, bulk-flow equations with turbulent shear parameters determined for rough surface seal surface conditions using Moody's friction factor formulae. The present analysis extends earlier work of Nelson et al. (1990b) for analysis of seal performance at arbitrary rotor center positions.

An application related to a LO2 annular seal is presented. Numerical predictions are calculated for 2 uniform clearance seals, and a third with a convergent tapered geometry. The value of stator surface roughness for a diamond knurl pattern used in the calculations is equal to 40 percent of the seal clearance. This value seems to be extremely large and out of

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the range for Moody's friction factor model to be valid. Could the authors explain this issue? I would like to add that for the example presented, surface roughness values on the order of 2 percent of seal clearance give similar results as those presented by the authors using the computer program of San Andres (1991). The inlet swirl is taken as 60 percent of the rotor surface speed and the numerical predictions show large values of the cross-coupled stiffness which bring the whirl frequency ratio to values close to 0.60. In this respect, I consider the results rather unusual since the use of a roughened stator surface is precisely desired to reduce leakage and improve the seal dynamic stability characteristics (i.e., reduce WFR). Then, in this application a roughened seal surface is clearly not beneficial which seems rather relevant (if correct). Could the authors explain more in detail this issue?

The whirl frequency ratio formula presented is due to Lund (1965). San Andres (1990) extended the original analysis to include inertia force coefficients. In regard to the WFR it was found that the cross-coupled inertia coefficient, $M_{XY} > 0$, has a detrimental effect on the stability ratio.

The investigation has shown that rotor-tilt relative to the seal inlet plane produces a significant increase in direct stiffnesses. This effect could be completely reversed if the tilt is relative to the seal exit plane as some of my recent calculations show. Have the authors also found the same behavior in their studies?

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D. Elrod[2]

Nelson and Nguyen (1987) showed that the choice of a friction factor equation affects seal rotordynamic coefficient predictions for nominally centered seals. Would the authors comment on the probability that the predicted dependence of rotordynamic coefficients on tilt is affected significantly by the choice of a friction factor equation?

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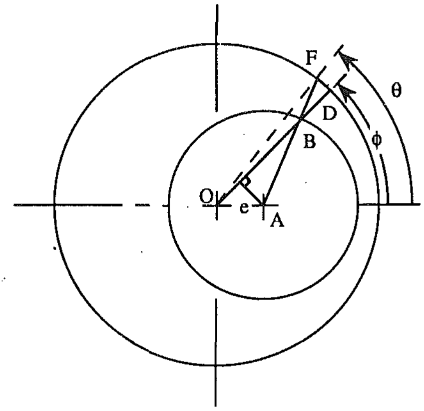

Fig. 13

A better understanding of and appreciation for this work would be gained from plots of the nominal "zeroth-order") pressure profiles around the circumference of the seal. For example, one presumes that the pressure profile at $\theta = \pi$ differs from that at $\theta = 0$. This would lead to a prediction that the stiffness in the tilt direction depends on the direction of perturbation along the axis.

Finally, would the authors clarify the clearance function in Eq. (5)? For the eccentric journal in Fig. 13, the clearances BD and BF are

$$BD = R + C_o - (R^2 - e^2 \sin^2\phi)^{1/2} - e\cos\phi$$

and

$$BF = [(R + C_o - e\cos\theta)^2 + e^2\sin^2\theta]^{1/2} - R$$

in which R is the rotor radius, C_o is the difference between the stator and rotor radii, and e is the distance from the stator center to the rotor center. What clearance does Eq. (5) represent?