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APPENDIX A

Perturbation Coefficients

$$B_s a_{0s} = [1 + (u_{\theta 0}/bu_{Z0})^2]^{\frac{ms+1}{2}}, B_s = \left(1 + \frac{1}{4b^2}\right)^{\frac{ms+1}{2}}$$

$$B_r a_{0r} = \{1 + [(u_{\theta 0} - 1)/bu_{Z0}]^2\}^{\frac{mr+1}{2}}, B_r = \left(1 + \frac{1}{4b^2}\right)^{\frac{mr+1}{2}}$$

$$B_s a_{1s} = [1 + (u_{\theta 0}/bu_{Z0})^2]^{\frac{ms-1}{2}}$$

$$B_r a_{1r} = \{1 + [(u_{\theta 0} - 1)/bu_{Z0}]^2\}^{\frac{mr-1}{2}}$$

$$A_{1z} = [a_{0s}\sigma_s(1-ms) + a_{0r}\sigma_r(1-mr)]/2f^4$$

$$A_{2z} = [(ms+1)\sigma_s a_{1s} u_{\theta 0} + (mr+1)\sigma_r a_{1r}(u_{\theta 0} - 1)]/2b^2 f$$

$$A_{3z} = [a_{0s}\sigma_s(2+ms) + a_{0r}\sigma_r(2+mr)]/2f^2 + 2q/f^2 - [a_{1s}\sigma_s(1+ms)u_{\theta 0}{}^2 + a_{1r}\sigma_r(1+mr)(u_{\theta 0}-1)^2]/2b^2$$

$$A_{1\theta} = [\sigma_s a_{0s} u_{\theta 0}(1-ms) + \sigma_r a_{0r}(u_{\theta 0}-1)(1-mr)]/2f^3$$

$$A_{2\theta} = (\sigma_s a_{0s} + \sigma_r a_{0r})/2f^2 + [\sigma_s(1+ms)a_{1s}u_{\theta 0}^2 + \sigma_r(1+mr)a_{1r}(u_{\theta 0}-1)^2]/2b^2$$

$$A_{3\theta} = [\sigma_s ms a_{0s} u_{\theta 0} + \sigma_r mr a_{0r}(u_{\theta 0}-1)]/2f - f[\sigma_s a_{1s}(1+ms)u_{\theta 0}{}^3 + \sigma_r a_{1r}(1+mr)(u_{\theta 0}-1)^3]/2b^2$$

DISCUSSION

G. L. von Pragenau[3]

This paper compares analysis with test data of damping seals and other seals to demonstrate the rotational effect of roughness. Damping seals offer low leakage and good whirl stability that increases the speed limit more than three times above the first critical speed. Damping seals recently replaced some labyrinth seals in the high pressure oxygen turbopump of the space shuttle main engine to eliminate whirl. The paper uses Hirs' turbulent bulk flow model corrected to new velocity vectors. Hirs' velocity vector has a Couette flow that is 50 percent of the rotor surface speed. Instead, a new Reynolds number should be derived from the velocity vector. Such a redefinition unclutters the formulation and could have helped the authors to reduce complexity and help the readability of their treatise. The paper is a welcome effort in confirming the benefits of damping seals through an independent analysis. The new seal type can lower weight and improve efficiency of high performance turbomachinery. We are grateful for God-given inspiration.

E. D. Jackson[4]

This paper extends the authors previous works which assumed smooth seal surfaces to cover the surface roughness effects on the high pressure annular seal dynamic coefficients. As noted by the authors, the value of a rough stator/smooth rotor surface profile in increasing the effective damping by reducing the cross-coupled stiffness term has been pointed out previously (as e.g., Von Pragenau [6]). In fact this concept has been demonstrated in one of the seals for the SSME HPOTP (high pressure oxidizer turbopump) to improve stability margins. The authors have shown the development of this capability for use in their analytical model of the seal dynamics.

Probably the most notable contribution of the present paper is the experimental data obtained, and the authors are to be commended for the extent and systematic coverage of key parameters affecting the seal performance. The authors approach of deriving effective seal roughness terms from leakage tests was believed to be sound, the same approach having been taken by Rocketdyne in seal tests (A-3). This approach is useful to industry in that it is much easier to measure seal leakage rates than to measure the actual seal dynamic coefficients. However, the authors' data do not show consistent variations for the theoretical versus the experimental results for all 4 seals. This leaves the implication that the effective roughness is not in itself adequate to characterize the seal dynamics. The authors have not presented any indications of their expected data accuracy, and it would be of interest to know if the indicated variations of theoretical and experimental results could be explained in part or in whole by these accuracies. For example, the accuracy of using the curve-fit approach to back out the added mass term would appear to be highly suspect and could explain why the present data show opposite trends to previous water data.

It is hoped that the excellent work in this seal dynamics area being performed by Dr. Childs and his associates at Texas A&M will be continued to attempt to bring our analytical models into even better agreement with data and to provide data for an even larger variation of seal types.

R. D. Brown[5]

The experimental work described in this paper is an important addition to previously published work. In particular the results confirm the suggestion that a rough stator reduces the forward cross-coupling effect and hence decreases potential instability.

However, the authors correlate their data using YAMADA'S work.

quoting

$$\lambda = nR_a{}^m\left(1 + \left(\frac{\bar{V}}{2R\omega}\right)^2\right)^{\frac{1+m}{2}}$$

YAMADA actual expression was slightly different

$$\lambda = nRa^m\left(1 + \left(\frac{7}{8}\right)^2\left(\frac{\bar{V}}{2R\omega}\right)^2\right)^{\frac{1+m}{2}}$$

the $7/8 = n/n+1$ arising from an assumed 1/7th power law for the turbulent velocity profile. I would be interested to

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know of the change in n and m (if any) that results from such a correction.

On the experimental side the authors report an increase in m for the round hole pattern. Indeed the directly measured friction factor increases slightly with Ra. As the authors point out this is inconsistent with pipe friction data.

Previously unpublished work at Heriot-Watt using round hole pattern stators gave measured friction factors which decreased with Ra. The seals were of L/D ratio 0.5 and a radial clearance of 0.231 mm. The holes were positioned on an equilateral grid with a side of 2.54 mm, the hole diameters were 1.524 mm initially (later 2.057 mm).

Correlating using Yamada's formulation gave

$$\lambda = .0556 Ra^{-0.17}\left[1+\left(\frac{7}{8}\frac{Rr}{Ra}\right)^2\right]^{0.415}$$

$$4000 < Ra < 13{,}000$$

Now the positive slope of λ_c is obtained directly from

$$\frac{-\partial P}{\partial Z} = \sigma \frac{\rho V^2}{2}$$

This equation appears dimensionally incorrect. From Black's original work

$$P_1 - P_2 = \frac{1}{2}\rho V^2(1+\xi+2\sigma)$$

where $\sigma\rho V^2$ is the pressure drop along the seal due to friction. This may be the reason for λ_c increasing with Ra.

Authors' Closure

D. Brown's observation with respect to a difference between the friction-factor correlation of this paper and that of Yamada is correct, and follows directly from Hirs' model. Specifically, Hirs uses the average "bulk-flow" velocity and makes no assumption with respect to the velocity distribution. Since we are using Hirs' model, we have no reason to calculate m,n values using Yamada's correlation; however, we would expect no discernible difference. Concerning the positive values for ms in Table 1 for the hole-pattern seal, subsequent tests of eight additional hole-pattern configurations have consistently yielded the expected negative values. We have no explanation for this aberrant result.

E. D. Jackson's point that the performance of the theory varies considerably in predicting the seal coefficients from seal to seal is certainly true. The theory is consistent in that it *always* does a good job of predicting the net damping coefficient and *rarely* yields any consistent agreement with added-mass terms. With regard to the direct stiffness coefficients, the theory seems to improve as the average clearance increases. A partial explanation for this systematic deviation is the entrance-loss. The present theory (and other published theories) assume that the entrance loss is a constant. However, our test results show a definite Reynolds number dependency, and subsequent calculations including this dependency have shown a general improvement in the agreement between theory and experiment for the direct stiffness coefficient.

The authors disagree with G. von Pragenau's view that Hirs' model necessarily implies a Couette flow that is 50 percent of shaft speed. The results of Fig. 4 clearly show the possibility for circumferential velocities that are different from 0.5 $R\omega$.

The authors appreciate the time taken by the commenters in reviewing and commenting on this paper and welcome their suggestions and inquiries.