Table 2 Comparison of periodical variation amplitude

Probe		Measurement	The model (Calculation)
Oil film thickness	H	0.17 mm	0.224 mm
	V	0.14 mm	0.163 mm
	X	0.195 mm	0.182 mm
	Y	0.19 mm	0.235 mm
Pin force	P	2.8 kgf	1.7 kgf
Temperature	T	14°C	19°C
	U	15°C	16°C
Period		99 sec.	102 sec.

Table 3 Effect of seal oil effective temperature on slow whirl

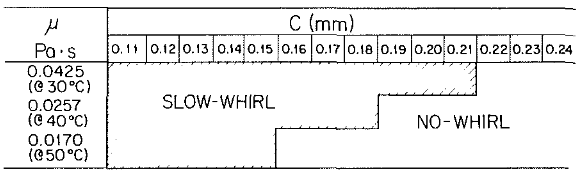
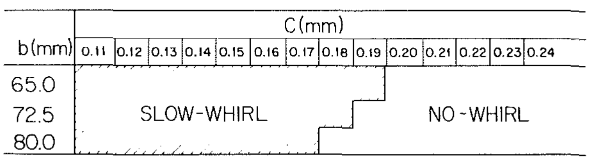


Table 4 Effect of radial thickness on slow whirl



The mechanism of this slow whirl is displayed schematically in Fig. 7. The solid line shows the seal-ring oval shape at a certain time in which the ring operates at the equilibrium position 0_R with a minimum film thickness $h_{\min}$. According to equation (9), the maximum oil film temperature $T_{f\max}$ is created at downstream with a shift angle from $h_{\min}$ position. This temperature distribution will cause, after a certain time, the oval deformation shown with a broken line. Thus the oval shape of the ring rotates slowly and periodically according to the shift of hot spots.

The magnitude of peak to peak variations are compared with the corresponding measurement in Table 2. These results show that theoretical prediction gives reasonable agreement with the experiment.

One other experimental observation worth bringing out here is the dependency of the slow whirl on the effective viscosity. The stable operation and no slow whirl was obtained at the smaller viscosity, i.e., by feeding the higher temperature seal oil. This statement was also supported by the theory.

The theory is applied to a series of 60 cm dia seal-rings, and the effect of the different design parameters is studied. The results are summarized in Tables 3 and 4, and indicate what sets of conditions can induce the slow whirl. To prevent the slow whirl, the larger clearance C, smaller viscosity μ, and larger radial thickness b may be beneficial. All of these calculations were performed by keeping other design parameters constant at their respective value, displayed in Table 1.

One of the shortcomings of the model may be the introduction of the scaling constant K_δ and the shifting angle η. If one can calculate these constants from the design parameters including the operating conditions, it will be improved as a complete one. However, one of the reasons to resign the trial was the complexity of the actual seal-ring configuration, and the fact that one can not evaluate the absolute errors involved in each assumption of the model.

It may be interpreted that the simplification is contributed by introducing these constants, and when choosing the appropriate constants, the analytical model can well simulate the slow whirl phenomena observed on the large seal-ring.

Conclusions

Efforts have been made to realize the slow whirl of large seal-ring in the clearance shaft seal. The theoretical model proposed in the paper predicts the oval deformation due to nonuniform temperature distribution created in seal-ring, and that the oval shape rotates slowly and periodically according to the shift of hot spots. One has to take into account that thermal deformation may be significant to design the seal-ring for increasing shaft diameter.

References

1 Dow, T. A., and Burton, R. A., "The Role of Wear in the Initiation of Thermoelastic Instabilities of Rubbing Contact," ASME Journal of Lubrication Technology, Vol. 95, No. 1, 1973.

2 Heckmann, S. R., and Burton, R. A., "Effects of Shear and Wear on Instabilities Caused by Frictional Heating in a Seal-Like Configuration," *ASLE Trans.*, Vol. 20, No. 1, 1977.

DISCUSSION

R. C. Hendricks[1]

The authors have reduced a complex problem to one which can be rapidly assessed.

Both our studies and those of the authors indicate the sensitivity of the pressure and temperature profiles to small perturbations and show similar locations for the "thermal bump." Such a situation causes an unbalanced torque couple on the support structure which precesses in the direction of rotation at a rate controlled by thermal diffusion in the angular direction and amplitude controlled by structural stiffness and radial thermal diffusion.

The results are opposite to those for rolling contacts where the classical Blok solution implies a heated zone upstream of the line of contact. Presumably as clearances become smaller and asperity contact begins, there will be a shifting of the warm zone and may produce a precession in the opposite direction.

Control of such case or seal distortions can be effected through the heat sink capacity of the material and or proper cooling, (eg. fins, heat pipes, heat sinks could be used). The use of a thin shroud low heat sink capacity ring would produce substantial distortion vs one of higher heat sink capacity. In authors experiment the shaft did not change, but in some cases the shaft can cause the disturbance to grow.

In our experiments, a vapor cavity is produced which has a profound effect on the pressure and temperature distributions. For our experiment, we have not looked for a precession effect, but we have noticed a difference in profiles for a survey in the direction of motion to that opposite the shaft motion.

Additional Reference

3 Braun, M. J., Hendricks, R. C., "Some Temperature, Pressure, Rotation and Vapor Cavity Characteristics of Shaft Seal With an Eccentric Casing."

[1] NASA Lewis Research Center, Cleveland, Ohio.

A motion picture supplement is available through NASA LeRC Photographic Branch.

A. Z. Szeri

The Authors are congratulated for an interesting presentation of what is not only a novel thermal instability but is also an important design problem. It is encouraging that a first order approximation to the problem yields good agreement with experiments. Nevertheless the discusser wonders what was the Authors' main reason for employing a quasi-steady approximation to film thickness. The discusser also suggests that the "shifting angle" η might show up naturally as a result of letting the viscosity vary with temperature. The importance of a viscosity distribution is also suggested by Table 3 where a mean temperature rise of only 20°C reduces the just stable value of the clearance from 0.21 mm to 0.15 mm.

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Authors' Closure

The authors appreciate useful comments by Mr. Hendricks and Prof. Szeri. Regarding comments for locations of the thermal bump, the theory has introduced the shifting angle, which is the essence for the slow whirl prediction. Also, the effective cooling or the effort to reduce the circumferential temperature difference created in the ring is useful for the prevention of the phenomenon, as seen in Tables 3 and 4.

The reason for neglecting the time dependent dynamic term for film thickness is that this contribution becomes unimportant when comparing the shaft rotational wedge term.

It would be left for the research in the future to predict the temperature field more precisely from the operating and design parameters.