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References

DISCUSSION

J. Cherubim

The paper presents some good work that demonstrates the possibilities of the use of foil bearings in practice. Since there has been relatively few recorded applications of foil bearings, this paper should assist others in their application. Another significant part of the paper is the tests at high temperatures which demonstrated the ability of the foil bearing to operate successfully with large temperature gradients.

Although the author concludes that the bearings are simple to manufacture, it might be reasoned otherwise when one considers the meticulous care and effort that was expended in the construction of the test apparatus. It appears that not all of the effort was required for test measurements but was needed in order to permit successful operation.

In the test rig it appears as if the foils are shielded from the turbine discharge air. However, it seems that the foils are still in the air path. Referring to Fig. 6 in Part I, could the cool turbine discharge air account for the difference in gap width in the self-acting mode? When the turbine is used in acceleration, it may cool the foils, reducing the foil length and decreasing the gap width. In deceleration the turbine is not in use and the foils would not be subject to any cooling effects.

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For the particular situation treated in this paper, $(\frac{\partial R}{L})$ and $3 \left( \frac{FR}{Eh} \right)$ are about equal, therefore it would be desirable to make both terms as large as possible. $(\frac{\partial R}{L})$ is more or less fixed by space and is likely to vary with the number of foil segments in each bearing; the three segment design is probably near optimum. Increasing $\frac{FR}{Eh}$ means increasing foil loading and ultimately one would be up against material limitations of the foil. Thus it appears to be difficult to alleviate the film thickness

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This paper adds to a series of articles by this author and his colleagues to apply foil bearings in the support of high speed rotors. While previous efforts have emphasized high speed operation and dynamic effects, the present paper is the first reported study on severe thermal effects. Altogether, three aspects of temperature were encountered; namely, thermal transient, temperature level, and temperature gradient. Taking each of those in the reverse order: steady-state axial temperature gradient of 70-135 deg F/in. appeared to have caused no ill effects at all for the 2.5-in-dia shaft; thermal transient had made its influence known in more ways than one; the temperature level itself, aside from its usual importance related to friction and wear (consequently, the ability to survive the contact during deceleration of the first run is certainly an impressive feat), and possibly creep strength, contributes only in the indirect manner as related to the magnitudes of the thermal gradients and transients.

Perhaps the most important transient thermal effect is caused by a temporary temperature difference $(\partial T)$ between the foil and the bulk of the journal leading to a change in the film thickness $(\delta h)$. A rather simple formula to account for this process in the self-acting mode can be derived from Eq. 10. As shown in the sketch, the ends of the foil segment are fixed at A and B. The geometry of the foil surface is simply assumed to be a composite of planes (AB, DE) and cylindrical (BCD). Local variations of gas film thickness at entry (B) and exit (D) are neglected. During the thermal transient, journal size, journal speed, the positions of the anchoring points are assumed to be unchanged, the foil-bearing gap is assumed to increase by $\delta h$ so that the foil is depicted as $AB'C'D'E$. Neglecting second order effects, the change in foil length is

$$\delta L = 6\delta h$$

where $\theta$ is the wrap angle. Let the change of foil tension per unit width be $\delta F$, then

$$\frac{\delta L}{L} = \frac{\delta F}{\delta L} + a\delta T$$

where, $L$ is the length, $\alpha$ is the coefficient of linear thermal expansion, $F$ is the modulus of elasticity, and $t$ is the thickness of the foil, respectively. One last equation relates the film thickness to the foil tension and the journal speed

$$h = 0.643\left( \frac{6\delta h}{F} \right)^{1/3}$$

During thermal transient, above relation yields

$$\delta h = -\frac{2}{3} \frac{\delta F}{F}$$

Eliminating $\delta F$ and $\delta L$ among equations (1), (2), and (4), one obtains

$$\delta h = \frac{3\alpha h \delta T}{2} \left( \frac{\delta R}{L} + \frac{7}{2} \left( \frac{FR}{Eh} \right) \right)$$

For the particular situation treated in this paper, $(\frac{\partial R}{L})$ and $3 \left( \frac{FR}{Eh} \right)$ are about equal, therefore it would be desirable to make both terms as large as possible. $(\frac{\partial R}{L})$ is more or less fixed by space and is likely to vary with the number of foil segments in each bearing; the three segment design is probably near optimum. Increasing $\frac{FR}{Eh}$ means increasing foil loading and ultimately one would be up against material limitations of the foil. Thus it appears to be difficult to alleviate the film thickness

change during thermal transients and one must learn to prevent thermal transient from happening in the first place. Preheating of the pressurizing gas is certainly a promising approach. It may also be worthwhile to insulate the backside of the foil segments from the environment as much as possible. The author's precaution to avoid convective gas currents is an indication of his foresight. Consideration of radiation and conduction insulation may also help.

Author's Closure

It is gratifying to note that Dr. Pan is impressed with the ability of the foil bearing to survive contact under extremely adverse thermal conditions. The state of the foil, which survived and remained operational, following rapid coastdown and rubbing at high temperatures, is illustrated in Fig. 5. Shown therein are transverse profilometer traces and magnified photographs of wear marks on the molybdenum foil-segment \( A_{II} \) (see left column, second row from top in Fig. 9), to be compared with the profilometer trace of a virgin foil-sample.\(^\text{12}\) The wider track, close to the upper edge, is essentially a burnishing mark. The narrow track, at nearly one-third the foil width from the lower edge, is approximately 100 \( \mu \) in. deep. The latter might have been caused by a hard foreign particle, but not necessarily during the rapid thermal transient, Fig. 5.

The effects of slack and of temperature were discussed in reference [4]. The simple estimate of gap change \( \delta h \), given by Dr. Pan in equation (5), would be useful, indeed, if fixed foil extremities, fixed lines of attachment, and other assumptions with regard to geometry corresponded to a realistically simplified model. In actual application, the geometry and boundary conditions involve many additional parameters.

It is difficult also to conceive of an extremely sudden, yet spatially uniform temperature jump \( \delta T \) throughout the entire foil. Moreover, the jump is unlikely to occur if one conceives that the gas used in pressurization need not be stone-cold and that the backside of the foils can easily be sheltered from convective and radiant heat losses. Nor is the interior of a real turbine-casing likely to be subjected to chilling convection currents present in the ambient surroundings of an air-driven simulator.

Even if one were to accept Dr. Pan's estimate as a guideline to design, the magnitude of

\[
\frac{\delta h}{\delta T} = aR \left( \frac{\theta R}{L} + \frac{3}{2} \frac{PR}{E \theta h} \right)
\]

(5)

can be decreased by increasing the wrap angle \( \theta \), decreasing the effective length \( L \) by means of more efficient foil guides and locks, and by decreasing the extensional rigidity \( E \). Furthermore, if one were to postulate a sudden temperature jump for a conventional journal bearing, for which

\[
\frac{\delta h}{\delta T} = aR
\]

(6)

there would be no room left for maneuvering and tradeoffs.

Mr. Cherubim is quite right in observing that meticulous care and considerable effort were expended in the construction of the experimental apparatus. Such care is an essential ingredient of good experimental research, since it enhances both the accuracy and the precision of data. Serious experimenters also recognize that preparatory steps are invariably more time-consuming than the execution of actual experiments.

None of the foregoing remarks contradicts the statements relevant to simplicity of foil-bearing construction and to operation unimpaired by very considerable distortions and misalignments. Indeed, a single strip of foil, combined into an attractive configuration with a few studs and clamps, is simple in comparison with the complexity of pivoted-shoe and spiral-groove bearing systems and their exacting precision requirements.

Few will take issue with Mr. Cherubim's observation that foils contract on cooling and his deduction that this may affect the magnitude of the gap width. The influence of dissipative heating and convective cooling was discussed in considerable detail in references [3] and [4], Part 2. In particular, the transient gap width during coastdown, following a sudden decrease in the rate of convective cooling, is shown in Fig. 13 of reference [3].

To summarize, we have demonstrated the operational and survival capabilities of gas-lubricated foil bearings under very adverse thermal conditions. Intentionally and unintentionally, the severity of tests exceeded by a large margin the conditions anticipated in actual applications.

\(^{12}\) The profilometer traces were obtained with the foil segment \( A_{II} \) and the foil sample tautly wrapped around 1-in-dia plug gauges.