DISCUSSION

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The authors have presented a complete analysis for a cryogenic hydrostatic journal bearing, accounting for variable lubricant viscosity with respect to both pressure and temperature.

Two separate three-dimensional analyses have been performed for the shaft and the bearing in order to establish the boundary conditions for the fluid film energy equation. However, for the fluid film, a two-dimensional energy equation has been utilized. It has been reported in the literature [25, 26] that the temperature variation and, consequently, the fluid viscosity variation across the film may be considerable depending upon the fluid inlet conditions. Is there any indication or justification of how important is the temperature variation across the film for a cryogenic hydrostatic journal bearing? Furthermore, the authors use heat-transfer coefficients to represent the boundary conditions at the fluid/journal interface and the fluid/bushing interface. The formal way of formulating these boundary conditions is, of course, to maintain temperature and temperature gradient continuity at each boundary. Definition of the heat-transfer coefficient is that at some point away from the interface the fluid temperature goes to a bulk value. Representing the heat transfer to both solids by heat-transfer coefficients implies that a considerable volume of the fluid near mid-plane is at a uniform (in z direction) bulk temperature. Could this be a valid approach in a very thin film with anticipated large-temperature gradients across it?

It is known that the determination of heat-transfer coefficients depends strongly on the fluid properties, flow conditions, and the geometry. Are these factors comparable in authors reference [18] and the present study?

Also, the boundary conditions governing the bushings are not clear. What is q_{env}, is it a known quantity? Should not the inside boundary condition of the bushing be q_b'' as related to equation (18)? Are h_b and T_a consistent in equation (24)? Should T_a be $\bar{T}_f$ supposing that this equation represents the heat transfer between the fluid film and the bushing?

As to the solution technique, the authors set the pressure in one pocket at a time equal to unity and zero everywhere else. This way, they obtain a set of ($n=20$) hydrostatic pressure fields $(P_{ij})_k$ due to the pocket pressures. Furthermore, they obtain an extra pressure field (hydrodynamic) (ρ_{ij}) due to the journal rotation. The final solution for the pressure is taken as the sum of all $(P_{ij})_k$ and (ρ_{ij}) based on the principle of superposition. However the principle of superposition does not hold for nonlinear operators as is the case of the Reynolds equation for a compressible lubricant and pressure and temperature dependent lubricant viscosity.

Can the authors give an approximation of the relative magnitudes between the part of the pressue due to journal rotation (P_{ij}), and the part of the pressure due to the pocket pressures $(P_{ij})_k$?

Additional References

25 Huebner, K. H., "Application of Finite Element Methods to Thermohydrodynamic Lubrication," *International Journal for Numerical Methods in Engineering*, Vol. 8, 1974, pp. 139–165.

26 Khonsari, M. M., and Beaman, J. J., "Thermohydrodynamic Analysis of Laminar Incompressible Journal Bearings," *Transactions ASLE*, Vol. 29, No. 2, Apr. 1986, pp. 141–150.

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Authors' Closure

The authors are indebted to Messrs. Mourelatos and Gecim for their thorough reading of the paper and for the constructive comments included in the discussion.

We shall try to answer the questions in the order they were posed.

In the case of the cryogenic hydrostatic bearing with two rows of pockets there is no experimental evidence attesting or rejecting the theory of sharp temperature gradients across the lubricating film. It is worth pointing out however, that every pocket is a source of fresh fluid which practically comes in at supply temperature. For the case presented here, such a configuration should contribute, in our opinion, to a rather flat profile of the temperature gradient, thus justifying the concept of lumped (bulk) fluid temperature across the gap.

The authors recognize the widespread use of the method of matching temperatures and temperature gradients at the journal/lubricant and bushing/lubricant interfaces, respectively. For the model presented here we have used (*a*) a lumped across film temperature assumption and (*b*) were reinforced in our approach by the results reported by Gazley [18]. We might add that for the lumped fluid film model assumption, *only* the introduction of a bulk heat transfer coefficient is a reasonable and correct approach.

Due to oversight the authors stand corrected for the equations (24) in the text. Correctly, they should read

$$k_b \frac{\partial T}{\partial r}\bigg|_{r=R_{b2}} = h_{b2}\left(T_b|_{r=R_{b2}} - T_a\right) = q_{env}$$

$$+k_b \frac{\partial T}{\partial r}\bigg|_{r=R_{b1}} = h_{b1}\left(T_b|_{r=R_{b1}} - T_f(x,y)\right) \qquad (24)$$

Also in equations (10*a*), (11), and (12), the velocity v should be replaced with the symbol ω.

For the constant properties, the principle of component solutions superposition holds. In the case of variable properties lubricant, the principle of superposition is used *only to guess a starting solution*. The starting solution is calculated with constant properties. Subsequently, the component solutions are never changed. The pressure in the pockets (for variable properties) is obtained by matching the flow at the restrictor inlet to the flow out of the pocket and onto the bearing land. The pressure on the lands between the pockets was calculated by means of a variable property Reynolds equation. Thus, the computational interdiction introduced by the nonlinear character of the variable property model is not denied by recalculation of component solutions and use of influence coefficients.

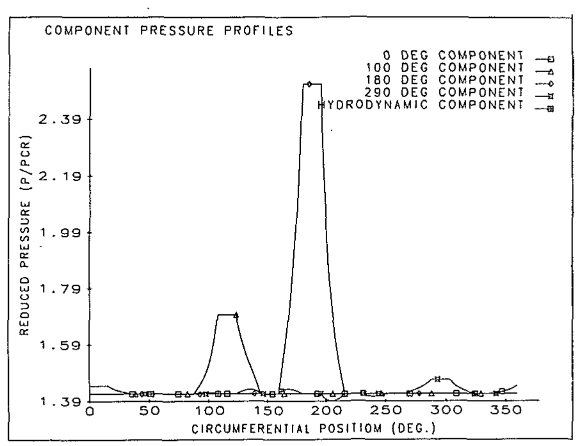

Fig. 8 Typical comparison between four hydrostatic components magnitude and the corresponding hydrodynamic component for liquid hydrogen. (100 KRPM, ϵ_x = .7)

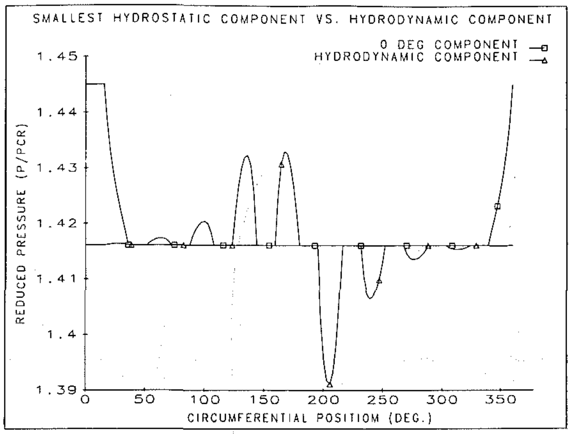

Fig. 9 Typical magnitude comparison between the smallest hydrostatic component and the corresponding hydrodynamic component for liquid hydrogen. (100 KRPM, ϵ_x = .7)

The last question raised by the discussers is rather interesting and lends itself to a lengthy response. In fact one could conduct a whole parametric study on the subject of comparative contribution of the hydrostatic components versus the hydrodynamic components.

For the case presented here, where liquid hydrogen is the working fluid, the dynamic viscosity is rather low (see Table 1), thus contributing little to the load carrying capability of the bearing. Figure 8 shows the hydrostatic pressure components at the minimum (MC) and maximum clearance and at 90 deg upstream and downstream of the MC. Superimposed, one can see by comparison, the magnitude of the hydrodynamic component and evaluate its contribution to the final pressure map. Figure 9 shows a comparison of the same hydrodynamic component and the lowest hydrostatic pressure component. It is apparent that in the region of the minimum hydrostatic pressure the contribution of the hydrodynamic component is of the same order of magnitude as its hydrostatic counterpart.

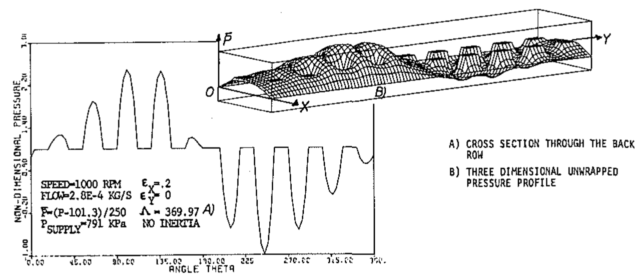

Fig. 10 Pressure map for a twenty pocket hydrostatic bearing rotating at 1000 rpm and using oil as the working fluid

Numerical experiments performed by the authors with oil as a the working fluid, have shown that even at low angular speeds the contribution of the hydrodynamic component is significant when viscosity is high, Braun et al. [27]. Figure 10 shows both qualitatively and quantitatively what happens in and around the pockets when oil of dynamic viscosity of 30.8 poise (3.08 Pa.s) flows in a journal hydrostatic bearing which rotates at 1000 rpm. The value of the viscosity has been chosen very high in order to dramatize its effects upon the hydrodynamic component when the bearing functions at low angular velocities.

It is evident from Fig. 10 that the hydrodynamic component can take over in magnitude, to the extent at which the hydrostatic effect in the covergent region is minimal by comparison.

Additional Reference

27 Braun, M. J., Hendricks, R. C., and Mullen, R. L., "Studies of Two Phase Flow in Hydrostatic Journal Bearings," Cavitation and Multiphase Flow Forum–1984, FED-Vol. 9, Energy Sources Technology Conference, New Orleans, La., Feb. 12–16, 1984.