both "on-design" and "off-design" modes and would they expect different forms of Figs. 2 and 3 for different values of preload, loading direction, etc.

Authors' Closure

The authors would like to thank Dr. Gethin for his interest in our paper and his insightful comments.

Regarding the first question, indeed, the film thickness depends on the rotation of each pad and this rotation is fully taken into consideration in our analysis. In fact, as shown in Eq. (3), the film thickness, h, is a function of the eccentricity of the shaft (ϵ_i) relative to the center of the internal pad curvature. Hence, the film thickness is a function of both the shaft position in the bearing and of the tilting angle of each pad.

Concerning the second question, we wish to reemphasize that the objective of this paper was to show the principle behind the generalization of thermohydrodynamic results and to illustrate the utility of THD design charts which we developed for this purpose. In addition to the prediction of the maximum temperature, the method enables one to rapidly compute a realistic effective viscosity, which could be used with the classical isothermal solutions to determine all the bearing performance parameters as minimum film thickness, oil flowrates, maximum pressure, power loss, etc. For this purpose, the design charts presented are very convenient.

Because of the large number of parameters that are involved in a tilting-pad bearing, we have presented results only for eccentricity ratio of $\epsilon = 0.5$. These results presented are, therefore, best suited for light and moderate loads. Under heavy loaded conditions, additional charts would have to be developed or one must perform a full THD analysis. Furthermore, thermoelastic deformations of the bearing elements may need to be taken into account (cf. Fillon et al., 1992 and Bouchoule et al., 1995).

Concerning the last question, the generalization of THD results can be extended to other bearing configurations. In fact, we have developed similar THD design charts for plain journal bearings (cf. Khonsari et al., 1995). There, we provided THD charts and a complete design procedure with three sets of eccentricity ratios corresponding to low, moderate, and relatively high loads.

Additional Reference

Bouchoule, C., Fillon, M., Nicolas, D., and Barresi, F., 1995, "Experimental Study of Thermal Effects in Tilting-Pad Journal Bearings at High Operating Speeds," presented at the STLE/ASME Tribology Conference in Orlando, FL, October 8–11, 95-Trib-49, pp. 1–7 (to appear in ASME JOURNAL OF TRIBOLOGY).

Thermohydrodynamic Analysis of Tilting-Pad Journal Bearings Operating in Turbulent Flow Regime[1]

A. Z. Szeri.[2] This is a competent paper, and the authors are congratulated for the excellence of their work. The thermohydrodynamic problem of lubrication in the superlaminar regime is, indeed, a difficult one.

Under laminar flow conditions there are few secrets left. That we are still unable to obtain satisfactory agreement with experiments is due, in my opinion, not to any theoretical difficulty but to the fact that one does not input correct parameter values. Even if correct parameter values are inputted, the results so obtained are applicable to specific cases only. To deduce general conclusions, the bearing should be considered as part of a system, and it is the system that should be analyzed rather than the components. The only significant theoretical problem left is inlet flow and mixing in the inlet grove, but then mixing is a problem well beyond the narrow confines of lubrication.

Under superlaminar flow condition the situation is quite different. We can, at best, only contemplate that superlaminar means turbulent in the accepted sense, even at high Reynolds number. That some superficial agreement is reached with experiment when using Reichardt's eddy viscosity proves very little. In any case, the eddy viscosity model was proposed for high Reynolds number turbulent flow. My question to the authors is, what is their justification for using results from a high Reynolds number empirical model, to explain what takes place during transition from laminar flow to who knows what?

Would it not be preferable to calculate and display the residue, rather than the authors' convergence criteria? One can get stuck at a local minimum and end up with a sizable residue, yet satisfy the "convergence criteria."

J. E. L. Simmons[3] **and S. J. Dixon.**[4] We would like to congratulate the authors on their paper which contributes to our understanding of the complex behavior of journal pad bearings in the laminar/turbulent flow transition region. Earlier experimental work by ourselves (Simmons and Dixon, 1994), cited by the authors, suggests that clearance ratio, specific load, and load direction all have an effect on the onset and significance of the turbulent regime. We would be interested to know if the authors have had any success in recreating any of these phenomena with their model.

We notice that the bearing geometry used by the authors hitherto provides for centrally pivoted journal pads. Do the authors have any plans to extend their work to examine the effects of offset pivot pads in the same laminar/turbulent regime?

S. Taniguchi.[5] The authors are to be commended for comparing their careful calculation of THD models to experimental data of large tilting-pad bearings. The effects of flow transition on the global performance of large bearings have been explained successfully. Quantitative investigation, not only in measurable data but also in the details of calculated results, can lead to an important conclusion. We would like to discuss three points.

(1) We have been attracted by the requirement of a large number of grid points in THD analysis. It is a very important and serious problem for industrial use of numerical program of THD. We had also been aware of the importance of mesh size selection, and employed smaller meshes near the wall than those of internal fields in the referenced paper (Taniguchi et al., 1990), but it might be inadequate. This requirement for an accurate THD solution can be explained as follows:

We consider the laminar case and neglect the term of transversal convection. Then the energy equation is written as,

$$\frac{\partial \overline{T}}{\partial \theta} = \frac{1}{a}\frac{\partial^2 \overline{T}}{\partial \overline{y}^2} + \Phi \tag{A1}$$

where

$$a = P_e \overline{h}^2 \overline{u},\ \Phi = \frac{N_e}{R_c}\frac{1}{\overline{h}^2 \overline{u}}\overline{\mu}\left(\frac{\partial \overline{u}}{\partial \overline{y}}\right)^2 \tag{A2}$$

[1] By L. Bouard, M. Fillon, and J. Frene published in the January 1996 issue of the JOURNAL OF TRIBOLOGY, Vol. 118, pp. 225–231.

[2] Robert L. Spencer Professor and Chair, Department of Mechanical Engineering, University of Delaware, Newark, DE.

[3] Professor and Head, Department of Mechanical and Chemical Engineering, Heriot-Watt University, Edinburgh, Scotland.

[4] Michell Bearings, Newcastle Upon Tyne, U.K.

[5] Mitsubishi Heavy Industries, Ltd., Hiroshima Research & Development Center, Hiroshima, Japan.