sults from this experimental investigation have proven this particular bearing geometry to be predictable and reliable when subjected to a wide range of applied static loads and speeds. Finally, using the foundation started with this work, investigations are underway to identify the rotordynamic coefficients for this test bearing operating in a wide range of conditions including the load-on-pad (LOP) and load-between-pad (LBP) configurations.

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APPENDIX A

Simple Thermal Model to Estimate Power Losses

Considering the bearing and housing as a lumped parameter system, the transport of energy is modeled in a simplified form. Using the conservation of energy principle (Moran and Shapiro, 1988) the drag power losses is equal to:

$$\text{Torque}\,\Omega = \text{Drag power} = \left[\frac{(2\pi L)(T_{\text{pad}} - T_{\text{housing}})}{\dfrac{\ln\left(\dfrac{R_3}{R_2}\right)}{K_B} + \dfrac{\ln\left(\dfrac{R_4}{R_3}\right)}{K_{fd}} + \dfrac{\ln\left(\dfrac{R_5}{R_4}\right)}{K_h}}\right] + \rho Q[c_p(T_o - T_i)] \quad \text{(A.1)}$$

where ρ is the density of the oil, and Q is the volumetric flow rate of oil into the bearing housing. The R terms in Eq. (A.1) represent the radii of the defined thermal layers. Specifically, R_2 is the radius of the flexure pad thermal layer, R_3 the radius of the outer bearing layer, R_4 the radius of the fluid pool layer, and R_5 is the radius of the aluminum housing layer. The first term in this equation is the energy transfer rate due to radial heat conduction through the bearing, while the second term represents energy transfer rate by the fluid mass flow (i.e., convection). Torque losses were estimated for each load and speed with the above equation using measurements of the oil inlet and outlet temperatures (T_o; T_i), average bearing pad and housing temperatures (T_{pad}; $T_{housing}$), and the measured oil flow rate of the working fluid (Q). Results are given in the following tables for three journal speeds.

DISCUSSION

R. Gadangi[2]

I would like to acknowledge the authors' effort in putting together this paper. This paper will serve as a good comparison/validation tool for computer programs. It would be helpful to the reader if the authors can further explain the following issues:

(1) In the conclusions section, the authors say that experimental results are in good agreement with computer results. In Fig. 10, experimental results show the expected trend (power loss increases with speed), while the computer program predicts almost similar power loss for 1800 rpm and 4500 rpm, and lower power loss for 3000 rpm. Could the authors explain this discrepancy?

(2) In conclusion (1), the authors say that the bearing stiffness increases with applied static load at constant speed or increase in speed at constant load. This conclusion is not always true. The increase or decrease in bearing stiffness depends on the Sommerfeld number at which the bearing is operating and the bearing geometry. I would like the authors to comment on this.

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Comments

(1) Figure 8 shows the temperature versus load curves at 4500 rpm. The difference between the outlet and inlet temperatures remains fairly constant, if not decreases a little with increase in load. It would have been more helpful if the experiments were run at constant inlet temperature and flow rate. This would have given better estimates for the bulk temperature rise and hence a better estimate for horse power loss.

(2) It would have been helpful if the authors had provided a comparison of the measured temperatures versus predicted temperatures.

Authors' Closure

The authors thank Dr. Gadangi for his comments on the experimental work carried at our laboratory. In regard to the first question posed, Fig. 10 presents the *dimensionless* power loss in the bearing for both theoretical predictions and measurements. The drag power is shown divided by the quantity $(2\pi\mu\Omega^2R^3L/c)$ which corresponds to the power derived for two concentric cylinders. Note that the test results are about 20 percent larger than the theoretical predictions at the highest test speed (4500 rpm).

The authors agree with Dr. Gadangi's argument on the generic behavior of stiffness in a hydrodynamic fluid film bearing. Our comments are particular to the test bearing, i.e., an observed bearing stiffening with both load and speed.

The supply (feed) lubricant temperature to the bearing was not kept constant on the experiments. This does not constitute an intentional oversight and it was dictated by the lack of funds to acquire an adequate oil cooling system. The authors believe that presenting the results in the form of a defect temperature (Fig. 9) allows the interested reader to grasp the important thermal rises in the bearing pads. Comparison of the measured temperatures to theoretical predictions based on a THD model of the flexure-pivot tilting pad bearing will be reported in the near future.