thrust-bearing speed-unit loading characteristic curve.

The following input data were used in the calculations:

1 Bearing dimensions and pad-thickness profile, Fig. 13.

2 Viscosity-temperature characteristics, specific heat, and density of 2190 oil, reference [2], p. 181.

3 Pad inlet temperature, [2], Fig. 2.

4 Radial temperature gradient of back face of pad—0 deg/in.

5 Circumferential temperature gradient of back face of pad

$$q_\theta = 2.76 \frac{N^{0.645}}{h^{0.260}}$$

where

q_θ = circumferential temperature gradient, deg/radian from inlet edge
N = speed, rps
h = minimum film thickness, in.

The empirical equation in item 5 was obtained by a least-squares procedure from the test measurements of thermocouple 8 (*b*), which recorded back-face temperature. Except for this item, all input data used were independent of the test measurements, following exactly the procedure established in [2].

DISCUSSION

C. Ettles[2] and A. Cameron[2]

The authors are to be congratulated on publishing one of the few papers giving actual measurements of oil film thicknesses and temperatures in full sized thrust bearings and the only paper known to us where these values are quoted for a ship at sea; truly it is a tour de force.

We have given [6] a theoretical analysis of thrust pad distortion with the assumption of conical pressure and a parabolic temperature distribution through the thickness of the pad, which we consider as a thick circular plate of area equal to the sector shaped pad. These assumptions enable a simple analytical expression for the distortions to be derived. The results are very similar to the computed values of the authors. We have plotted them in Fig. 14. Have the authors any idea why these differ so seriously from the measured values of distortion? Are these inaccurate in some way? It certainly seems odd that they are almost constant from 70 to 500 psi. As the majority of the distortion is thermal and the temperature difference is very small at low loads, it is difficult to see the rational of this constancy.

The most important factor in the distortion is the temperature, and we used a generalization of some tests on a 4½ in. P.C.D. bearing. It is interesting that the authors' results fit in with our expression for temperature as shown in Fig. 15. Have the authors any further data, which could corroborate these temperatures? They are most important.

[2] Imperial College of Science and Technology, University of London, London, England.

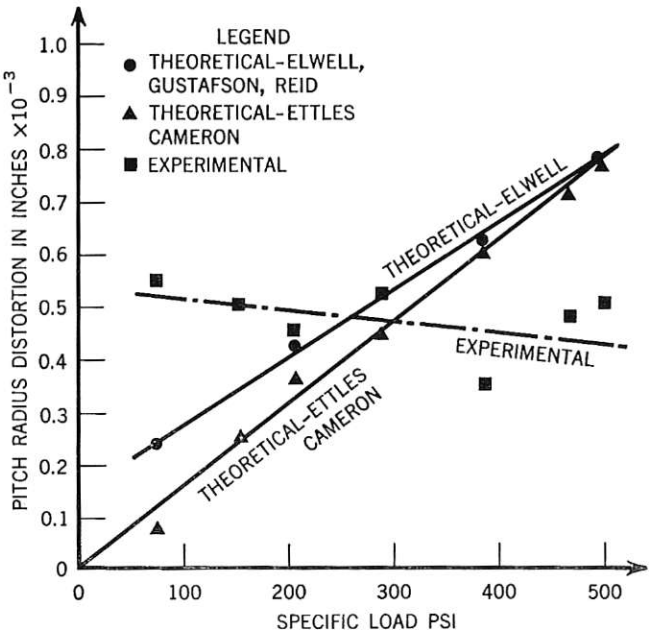

Fig. 14 Variation of pitch line distortion with load

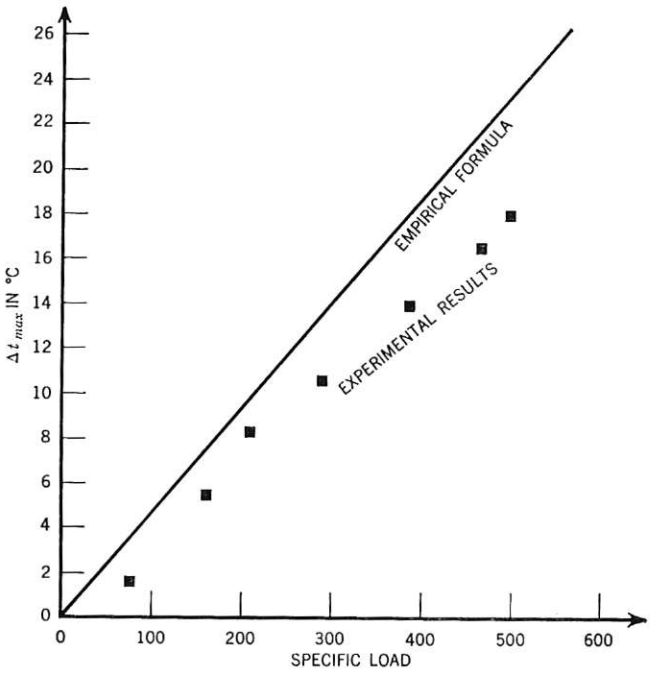

Fig. 15 Maximum temperature difference between front and back faces. Comparison with empirical formula: $\Delta t_{max} = 0.045\, p_{avg} + 2.0\, V^{2.75} \times 10^{-8}$.

p_{avg} = lb/in.²
V = in./sec
t_{max} = deg C

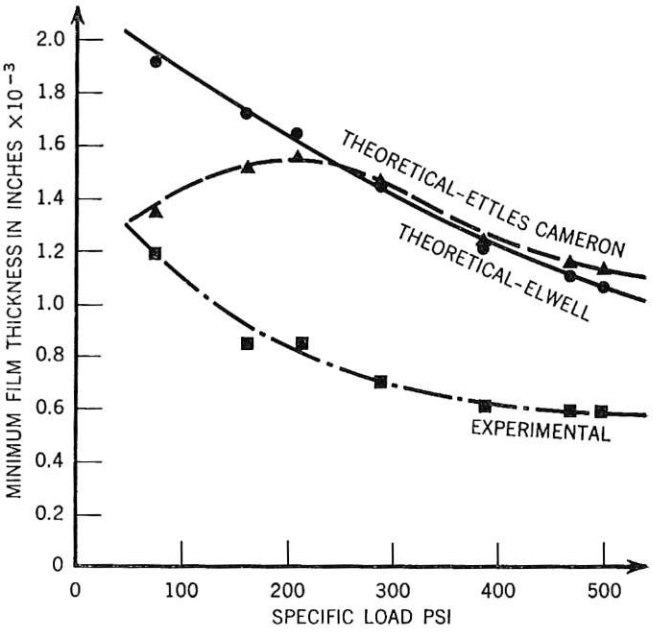

Fig. 16 Variation of minimum film thickness with load

We assumed the pad distorted into a spherically crowned shape. Using Raimondi's theoretical results [7] we calculated the minimum oil film thickness and these are shown in Fig. 16. Our theory and the authors are both in substantial agreement, but differ from the experimental results. Could it be that the inlet viscosity was lower than the expected value due to the hot oil carry over?

The diffusion of heat from the thin boundary layer of oil adjacent to the rotor depends very strongly on whether it is laminar or turbulent. If the oil film thickness at exit is taken as the characteristic length then the Reynolds number of this layer is $\sim 11 - 5$. Andrade [8] has shown (from the discharge of fluid jets into free static fluid) that laminar conditions are found for Reynolds numbers up to 30.

Although the decay of temperature from the top of this laminar layer must be very rapid, the rotor surface temperature remains constant (Christopherson [9]). Hence, at the inlet to the next pad the temperature gradient is considerable. The placing of thermocouples as much as $0.010 - 0.020$ in. above the rotor surface could give readings less than the true value, so that the oil entered the pad at a hotter temperature than recorded. This would lead to a calculated film thickness greater than actual. Could this be the explanation of the difference? We are studying the theory of the boundary film and thermal layer, which we hope to publish shortly.

We would very much welcome the authors' views on these queries. Their very number indicates our interest in their most important paper.

Additional References

6 C. Ettles and A. Cameron, "Thermal and Elastic distortions in Thrust Bearings," The Institution of Mechanical Engineering Conference, May, 1963.

7 A. A. Raimondi, "The Influence of Longitudinal and Transverse Profile on the Load Capacity of Pivoted Pad Bearings," *Trans. American Society Lubrication Engineering*, 1960, vol. 3, p. 265.

8 E. N. Andrade, "The Velocity Distribution in a Liquid-Into Liquid Jet. Part 2; The Plane Jet," *Proc. Phys. Soc.* (London), 1939, vol. 51, p. 784.

9 D. G. Christopherson, "A Review of Hydrodynamic Lubrication," Mechanical Engineering Conference on Lubrication, *Wear*, 1957.

Authors' Closure

We sincerely appreciate the efforts of Mr. Ettles and Dr. Cameron in presenting their stimulating discussions.

Despite the fact that bearings of this type have been very widely used for half a century, there are still many aspects of their performance which have not been adequately investigated or explained.

The distortion of the segments under load is one of these. The data reported in this paper are unfortunately not sufficient to explain the divergence between theory and experiment in Fig. 14. The best we can say is that the bearings which were tested did not have the perfect surfaces assumed in the analytical work, leaving the explanation of this discrepancy for the future, when better experiments and more knowledge may permit its answer. Our measurement accuracy, for instance, was estimated to be only ± 0.0002 in. New instrumentation has subsequently been developed which can improve on this, and knowing what to look for may produce the answer in the future.

In regard to the temperature data referred to in Fig. 15, we have no other data than that given in the paper.

It is highly probable that the actual oil inlet viscosity *was* significantly lower in the experiments than assumed in the analytical work, as suggested in the discussion. This matter also requires further investigation, as our second recommendation presented.