Discussion

R. A. Baudry.[6] The authors are to be congratulated for a very clear and significant contribution to the theory of the pivoted-pad thrust bearing.

It is not only very interesting but very important to know that a relatively small amount of crowning of the pad is sufficient to permit a centrally supported thrust-bearing pad to work as efficiently as an offset pad. This feature permits a thrust bearing to operate efficiently in either direction of rotation and is of consequence from both the manufacturing and operating standpoints. Pivoted-pad thrust bearings have been made for many years with a central pivot. Recent tests on such a large thrust bearing as shown in Fig. 14, herewith, indicate that a thick tapered oil film is established very rapidly and remains an appreciable time after the machine has been stopped. It is interesting to note that for this particular bearing a temperature gradient of 1 deg C per in. of pad thickness corresponds to approximately a crowning of 0.001 in. which, according to the authors, is sufficient to explain the formation of the oil film. After stopping, the appreciable oil film shown in Fig. 14 is due in part to the temperature deformation of the pad.

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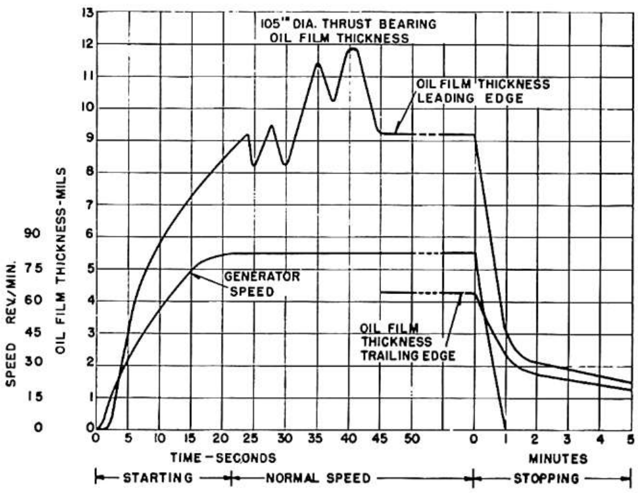

Fig. 14 Test Results on Pivoted-Pad Thrust Bearings

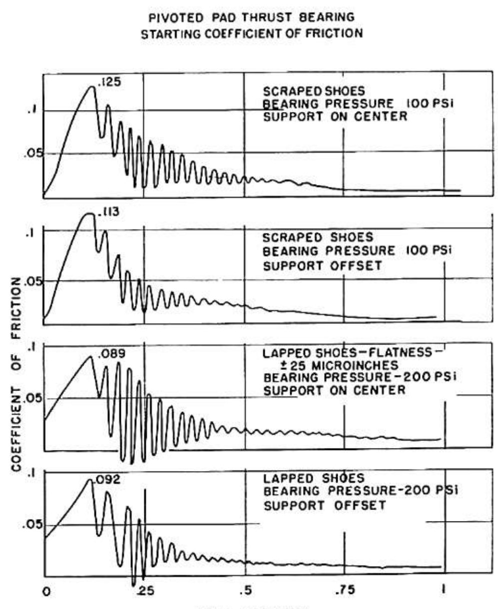

Fig. 15 Starting Coefficient of Friction for Pivoted-Pad Thrust Bearing

Other starting tests made recently on a small thrust-bearing testing machine indicate also that centrally and offset pivoted pads operate equally well. In Fig. 15 are shown four typical tests. The two upper ones are made with pads scraped to a surface plate; the two lower ones with pads having a lapped surface and having a crown smaller than 50 microin. as measured with an optical flat.

The pad with a lapped surface had a slightly smaller starting coefficient of friction but showed larger torque oscillations. These oscillations are self-induced and usually are present when starting a thrust bearing.

The starting period of a thrust bearing starting under load is the most critical interval of all operation on account of the solid and boundary friction and, therefore, there is a limit to the amount of crowning that should be given to the pads of such machines. Apparently, experience indicates that the normal manufacturing variations, temperature gradients, temperature and viscosity effects of the oil are sufficient to permit satisfactory operation of centrally pivoted thrust bearings.

F. Osterle[7] and E. Saibel.[8] As was pointed out in this paper, if adiabatic lubricant flow is assumed for the slider-bearing problem, it becomes necessary to take the variation of viscosity and density with temperature into account. There are several ways of doing this available in the literature. These involve varying degrees of computational difficulty.

The simplest technique is to consider the viscosity and density constant at values corresponding to the average of the inlet and outlet lubricant temperatures. The authors presumably employed this technique with the convex pad. A better approximation is to assume the viscosity to vary linearly with distance along the slider, the linearity constant being adjusted so as to satisfy the energy equation for adiabatic flow. The authors discuss this method in connection with the flat slider case. Recently the writers[9,10] developed a method for taking viscosity variation into account for convex pads of exponential form which is exact to the extent to which the viscosity variation with temperature is representable by an exponential.

Experience has shown that even the rough approximation afforded by the first method (constant viscosity at an average value) yields reasonably accurate results for the load capacity. However, such is not the case with the location of the center of pressure. The predicted location of this point is rather sensitive to the degree of approximation employed in taking the viscosity variation into account.

As an illustration of this, consider the authors' result that by imposing the condition that the center of pressure be at the midpoint of the pad and computing the load by the constant–viscos-

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[9] "On the Solution of the Reynolds Equation for Slider-Bearing Lubrication—IV: Effect of Temperature on the Viscosity," by F. Osterle, A. Charnes, and E. Saibel, Trans. ASME, vol. 75, 1953, pp. 1117–1123.

[10] "On the Solution of the Reynolds Equation for Slider-Bearing Lubrication—VI: The Parallel-Surface Slider Bearing Without Side Leakage," by F. Osterle, A. Charnes, and E. Saibel, Trans. ASME, vol. 75, 1953, pp. 1133–1136.

ity technique zero load is predicted while the linearly varying viscosity technique predicts an appreciable load.

As a further example, the authors' linearly varying viscosity treatment of the parallel-surface slider predicts a center of pressure very nearly at the mid-point of the pad, Fig. 5 of the paper. This fact figures in the authors' conclusion that density variation alone is incapable of accounting for the load capacity of centrally pivoted pads.

Our exact treatment of the parallel-surface slider bearing, taking account of both density and viscosity variations, predicts a center of pressure considerably upstream of the mid-point of the pad (see Fig. 1[10]), which presents a slightly better case for the "density wedge" as being, partially at least, responsible for the load capacity of centrally pivoted pads.

In the light of the foregoing discussion we think that an interesting investigation would be the determination of the sensitivity of center-of-pressure determinations to the way in which viscosity variations are taken into account.

R. B. Thicke.[11] The authors are to be complimented on this original and interesting paper. It will, no doubt, interest engineers throughout the country who are involved in thrust-bearing design.

Our assumptions for purposes of calculation are that the inclination of the shoe is determined by variation of viscosity and location of support. We assume that load capacity depends on inclination of the shoe, speed, average viscosity, and side leakage. These can be translated into a chart which provides rapid calculation on a trial-and-error basis and gives results which are not much different from those of Norton's "no side-leakage" method or Kingsbury's optimum-conditions method.

Our experience has shown that a shoe with a support over a small area near the center will become convex in operation, although probably to a greater extent than the authors propose. Evidently this explains why a shoe with a central support works nearly as well as one with a support at the optimum position.

Because pressure is higher at the center of the shoe, the shoe will become concave if supported over a large area provided the shoe is of such dimensions as to allow the pressure to overcome temperature distortion. Mr. Gynt[12] found that the operation of spring-mounted bearings was improved by concentrating the supporting springs toward the center of the shoe to eliminate concavity. The authors' paper helps to explain Mr. Gynt's findings.

One point which the writer feels should be stressed is the danger of application of the authors' theory to bearings such as vertical water-wheel-generator thrust bearings which must start under load. As static load on a bearing with shoes of a convex profile probably would be concentrated over a small area, the danger of a wipe would be increased at the instant of start-up even though the tendency of the shoes to roll would build up an oil film rapidly. The writer believes further experiments are necessary before a convex-profiled shoe should be used in such an application.

However, because of the cost, the possibility of controlling the profile during manufacture seems remote even though theoretically a considerable improvement in operation will result. This leaves the design engineer with the interesting problem of flexibilities and temperature distortion if an attempt is made to control the profile during operation.

[11] Engineer, Product Design Section, Canadian Westinghouse Company, Ltd., Hamilton, Ontario.

[12] "Recent Development of Bearings and Lubrication Systems for Vertical Generators," by S. Gynt, *ASEA Journal*, vol. 20, 1947, pp. 72–87.

Authors' Closure

The authors wish to thank the discussers for their interesting and valuable comments. The curves shown by Mr. Baudry are especially interesting and illustrate that a centrally pivoted pad thrust bearing is capable of operating with a relatively large film thickness. This is not surprising since elastic and thermal distortions are sufficient to cause a crown of the order of that necessary to produce appreciable load capacity.

We agree with Professors Saibel and Osterle that an accurate investigation of the effect of viscosity-variation assumptions on the center-of-pressure location would be enlightening. With regard to the thermal wedge, however, it is felt that the approximate treatment used illustrates the relative order of magnitude of this effect and that any refinements in the treatment would not substantially alter the results.

It was not the authors' intention in presenting this paper to imply that all centrally pivoted pad bearings should be crowned. As Mr. Thicke points out, this may not be feasible in some applications. The beneficial effects of variable viscosity and variable density will in many cases be sufficient to permit successful operation. In applications, however, where very little or no benefit can be derived from these effects, such as in water or air-lubricated bearings, it will be necessary to employ crowned pads in order to obtain appreciable load capacity.