less than one-half of the film thickness at the trailing edge of the bearing.

Operating Curve. Fig. 11 shows the relationship between the dimensionless coefficient of friction factor, $f(R/\delta)$, and the dimensionless load number, S. This curve is for a *centrally pivoted* crowned slider bearing. It was not possible to generate this curve analytically, and consequently it was generated by considerable cross plotting of data for the centrally-pivoted bearing. This curve is similar in form to the well known Sommerfeld-coefficient of friction plot for journal bearings. In either case, the operational frictional characteristics of similar bearings are brought into a single curve. In the present case, it is possible to obtain one such curve for each pivot location.

The results of several tests of crowned slider bearings are shown in Fig. 11. In this experimental work, the bearing loads, crown heights, and speeds were varied. The experimental results are seen to correlate very well with the theory. Each bearing test contributed to a portion of the operating curve, and at the lower end of each portion there was a break in the curve and an increase in the friction factor. The envelope of all test results yielded a close fit to the theoretical curve. This result is similar to that found in plotting test data for journal bearings. It is felt that an even closer correlation could be obtained with a more refined technique for measuring the temperature of the lubricant as it passes through the bearing. The temperature used in computing the lubricant viscosity (required in obtaining the Sommerfeld number) was the average temperature obtained from six recordings around the circumference of the slider. Thus the experimental values of the Sommerfeld number were obtained on the basis of a single, average temperature, whereas the real slider bearing is influenced by a two-dimensional temperature field. These are two different situations, and perfect correlation between theory and experiment should not be expected.

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DISCUSSION

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In their calculations of bearing performance the authors of this very interesting paper have treated the pivot as an ideal hinge, able to move freely in all directions. In reality, however, the spherical bearing, acting as the pivot, could conceivably act as a restraint to the free movement of the Michell-type crowned bearing.

Relatively simple considerations show that such a pivot-torque would considerably alter the performance of the bearing as a whole. It seems very likely that the type of lubrication of the pivot at the very high operating loads is not always fully hydrodynamic during a complete revolution of the swash-plate. In that case a great effect on the performance of the crowned bearings should be found, particularly at conditions of high loads and low speeds acting simultaneously.

I wonder if the authors observed any such effect in their experiments. If not, could they offer any suggestion with respect to the mechanism of pressure formation necessary for pure hydrodynamic lubrication between the presumably perfectly fitting, spherical surfaces of the pivot.

Authors' Closure

This investigation did not explore the mechanism of pressure formation on the spherical pivot. Since the applied load was unidirectional it is doubtful that the pivot was hydrodynamically lubricated during these experiments. It was possible though, to observe the slider bearing change its attitude at different loads and speeds via the thermocouples embedded beneath the slider surface. These attitude changes were observed over the range of test parameters investigated and were found to occur even during conditions of simultaneous high load and low speed.

The performance of crowned circular slider bearings in swash-plate drive mechanisms has been very satisfactory.[3,4] In most swash-plate drive applications it is possible to design the pivot using hydrodynamic squeeze film principles because the applied load normally reverses and monentarily unloads the slider bearing and pivot. Under certain operating conditions this load reversal may not occur and it is necessary to make allowances for boundary lubrication of the pivot. In a swash-plate drive this absence of load reversal will usually take place at low speed and high load. Rocking of the pivot under the action of a swash plate may also contribute to the formation of a measurable hydrodynamic film. This effect was not investigated as part of our work.

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