DISCUSSION

F. A. Martin[12]

The authors are to be commended for presenting a very interesting paper indicating an optimum bearing shape for a particular dynamic duty cycle involving a constant load on the bearing and an oscillating journal. When considering minimum film thickness, an improvement by a factor of 36 over conventional bearings is really impressive; which prompts one to ask the question—what are the features making the particular geometry ($E_1 = 0$, $E_2 = 1.5$) optimum?

Several factors appear to be involved and the discusser has found that relating performance to a clearance space characterized by the limits to journal movement may prove useful. This technique can only be applied to the circular journal case (i.e. $E_1 = 0$) but will apply to any associated bearing shape defined by E_2 (i.e. δ_2/C). For example, if we examine the authors' example case C (i.e. $\delta_1/C = 0$ and $\delta_2/C = 2.4$) a clearance space can be constructed as shown by the various steps in Fig. D1. In step 4 of this figure the curve abc defines the maximum extent of the possible movement of the journal center. It can be seen, for this particular case, that the journal can never touch the crown of the bearing. This is demonstrated more clearly in Fig. D2.

Fig. D2 has been developed to show geometric characteristics additional to the authors' paper. These include:

(*a*) trend in *clearance shapes* defined by the limits of free journal movement.

(*b*) minimum *film thickness position* as well as magnitude.

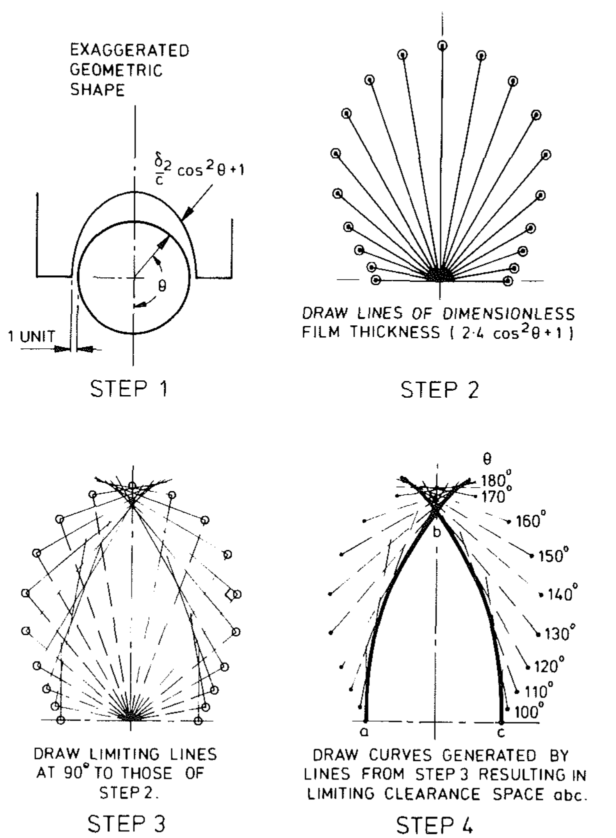

Fig. D1 Construction of limiting clearance shape for a noncircular bearing $\delta_1/c = 0$; $\delta_2/c = 2.4$

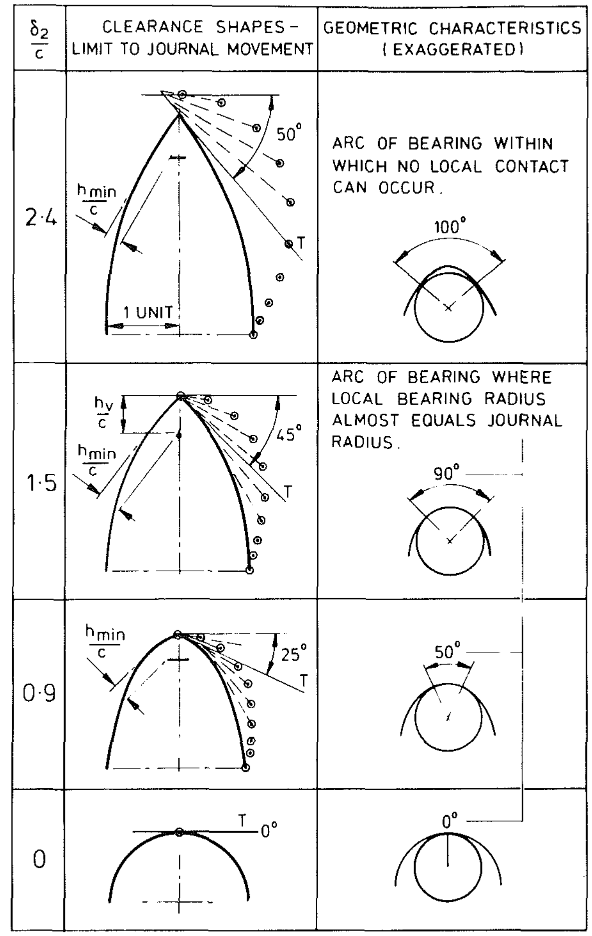

Fig. D2 Limiting clearance shapes, minimum film position and other geometric characteristics $\delta_2/c = 0, 0.9$; 1.5 (optimum shape) and 2.4

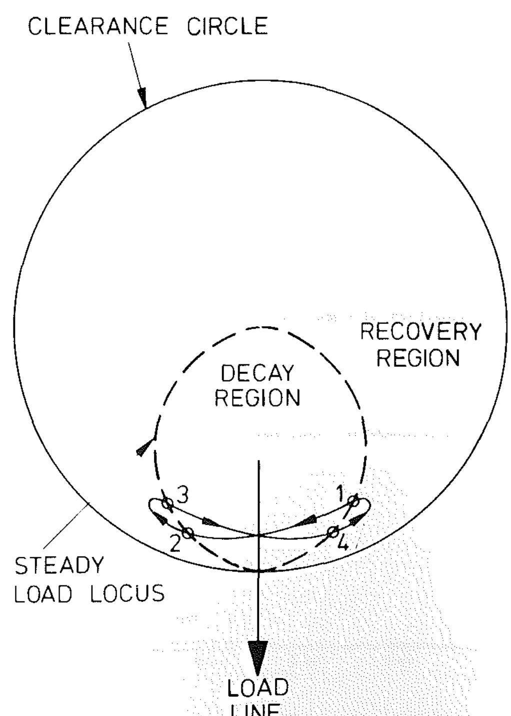

Fig. D3 Large decay region in circular bearings

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(*c*) *angular extent* of bearing *where local contact is impossible.*

(*d*) *angular extent* of bearing *where the radius of the journal and bearing are the same.*

The cases considered in Fig. D2 relate to the line of E_1 (or δ_1/C) equal to zero (circular journal) which passes through the optimum case shown in the authors' Fig. 7 and includes the authors' example cases A, B, and C with δ_2/C equal to 0.9, 1.5, and 2.4, respectively.

When δ_2/C equals 0.9 the journal and bearing radius are approximately the same over an angular extent of 50 deg. This is shown by limit lines which pass through the apex of the "clearance space" and span 25 deg (from the horizontal line to tangent line marked *T*). When δ_2/C is increased to 2.4 the bearing has the characteristic of a relieved surface in the crown over an arc of 100 deg (shown where the limit lines do *not* pass through the apex and are outside the limiting clearance space). For the optimum case (δ_2/C equal to 1.5) the bearing and journal radii are approximately the same over an angle of 90 deg. When the journal and bearing are separated by a lubricant film this should result in a film thickness at the crown larger than $h_{\min}$. The relative values of h_v/C and $h_{\min}/C$ of 0.55 and 0.36 confirm that the two radii are practically the same. The functional part of the film under load will be thicker on the load line and the discusser wonders if the authors have any ideas on how this area can be supplied with lubricant without derating the bearing due to grooving.

Finally, it is noticeable that the journal locus for the optimum case is "constrained" to almost a dot in the clearance space. In a cylindrical bearing however, the journal dynamic locus will span the steady load equilibrium locus as shown by the characteristic shape in Fig. D3 (not to scale). An important feature is the decay (decreasing film thickness) and recovery (increasing film thickness) regions inside and outside the steady state locus. Do the authors' believe that this concept will also apply to non circular bearings where the decay region could be substantially reduced? Could the authors predict the steady state locus for the optimum geometry case and comment on this feature?

P. K. Das[13]

I find the implications of the paper quite remarkable. In my industry of diesel engines, the regular cylindrical bearings are in extensive use. The suggestion that their load capacity may be greatly increased by slightly altering the shape of the bearing or journal has great practical value.

It is, however, not clear if the authors considered the physical constraint to rotation that exists for irregular bearing-journal combination. Without this constraint, interference between the bearing and journal may occur during rotation. The constraint is particularly important when the rotations are complete and both F^x and F^y loads are present as in the case of diesel engine bearings. The derivation of equation [7] from [6] would also help since some further approximations are involved.

Only elliptic type irregular shapes have been optimized. Will it be prohibitively difficult to assume a general second order equation for the irregular shape and then determine what would be optimum?

The authors chose the one dimensional long bearing approximation for the Reynolds equation while the short bearing assumption is true for a variety of situation. Are the conclusions likely to be much different if the latter theory is used? Was the finite element method necessary or did it offer any advantage for this one dimensional equation? Are there significant difficulties in extending the analysis to 2 and 3 dimensional fields?

A designer also considers the dimensional proportions (length, diameter, clearnace) and other performance parameters such as power loss and oil flow requirements in optimizing a bearing design. Is it practical to include these factors in the present analysis? Moreover, many bearings operate over a wide range of speed and load. What strategy would be used for optimizing these bearings?

I congratulate the authors for this very significant work. I hope that the analysis will be extended to make it a truly practical tool for a variety of applications. The potential benefits are enormous.

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Authors' Closure

The present analysis of elliptical bearings suggests that a small derivation in bearing geometry *can* produce a satisfactory film thickness in applications where a circular bearing *cannot* (because of insufficient oscillation and/or frequency). An alternate approach in such applications is a rocking journal bearing with multiple offset segments, the analysis for which is being published simultaneously [11]. Work done by Berthe et al. [12] is also pertinent to the present analysis.

The present analysis shows a tremendous performance gain for the elliptical bearings. Intuitively it is not obvious why it should be so. In this regard the discussion of Mr. Martin is very pertinent and appreciated. He has suggested ways that might help in understanding the mechanism of elliptical bearings. We have generated clearance space plots for circular journal and various sleeve ellipticities, similar to one shown by Mr. Martin in Fig. D2. Unfortunately, these clearance space plots, though interesting to observe, do not help in understanding the performance of the elliptical bearings, in general, and the optimality of $E_2 = 1.5$, in particular.

It is possible to predict steady-state loci for elliptical bearings (with circular journals) just like those for circular bearings. Such a locus for the optimum geometry case is shown in Fig. D4 along with the clearance space and the equilibrium locus for the duty cycle considered in this paper. There are several differences from the circular bearing case,

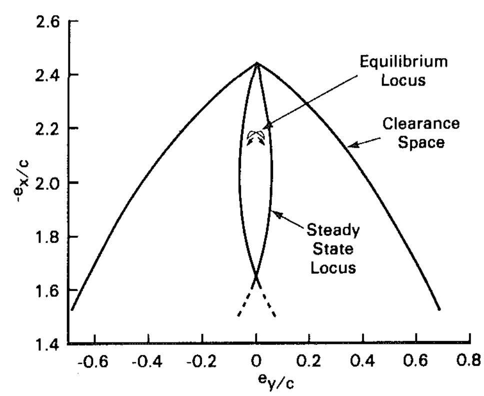

Fig. D4 Steady state and equlibrium loci for the optimum elliptical bearing

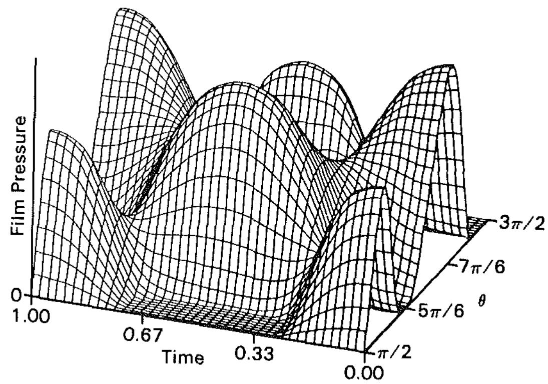

Fig. D5 Film pressure distribution around the bearing versus time for the optimum elliptical bearing

most important of which are that for the elliptical bearing the steady-state locus does not form a closed loop and that the equilibrium locus does not cross the steady load locus. The later observation would be alarming if the concept of decay and recovery regions as presented by Mr. Martin for the circular bearing is also true for the elliptical bearing. But as the mobility formulation [13] does not apply directly for noncircular geometry, we do not think the concept of decay and recovery regions would apply to the elliptical bearings.

Dr. Das has raised several questions of practical importance. The consideration of kinematic constraint to rotation of the journal is inherent in the analysis. The use of the finite-element method was necessary to be able to consider any aribtrary variation in the shape of the journal or the sleeve. After the presentation of this paper a general two-dimensional finite element analysis has been completed; the publication of which is forthcoming. The results presented in Fig. D4 are based on this two-dimensional analysis which predicts a minimum film thickness ratio of 0.181 for the optimum case. The two-dimensional analysis seems to validate the conclusions drawn in this paper.

The present analysis can be easily extended to include the power loss and oil flow calculations. The analysis has no restrictions on considering other non-circular shapes or the dimensional properties of the bearing in the optimization process. The only constraint in considering a wide range of speed is time and cost. In this regard the same strategy should be used as for designing regular circular bearings.

Finally, as a matter of interest, an extension of Fig. 9 is presented in Fig. D5, where the bimodal pressure distribution around the bearing for the optimum case is shown as a function of time. The bimodal pressure distribution is a characteristic of the elliptical bearings and is expected because of the presence of two converging-diverging regions in such bearings.

We are currently doing some experimental investigation to validate the conclusions of this analysis.

Additional References

11 Booker, J. F., Goenka, P. K., and VanLeeuwen, H. J., "Dynamic Analysis of Rocking Journal Bearings with Multiple Offset Segments," ASLE-ASME Lubrication Conference, New Orleans, Louisiana, October 5-7, 1981, Paper No. 81-Lub-34, ASME Journal of Lubrication Technology (in press).

12 Berthe, D., Fantino, B., Frene, J., and Godet, M., "Influence of the Shape Defects and Surface Roughness on the Hydrodynamics of Lubricated Systems," *Journal of Mechanical Engineering Sciences,* Institution of Mechanical Engineers, Vol. 16, No. 3, 1974, pp. 156-159.

13 Booker, J. F., "Dynamically Loaded Journal Bearings: Numerical Application of the Mobility Method," ASME Journal of Lubrication Technology, Vol. 93, No. 1, Jan. 1971, pp. 168-176.